HEAT RECOVERY REFRIGERATION

IN

NEW ZEALAND DAIRY SHEDS

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ABSTRACT

Increased energy costs initiated an investigation into refrigeration heat recovery as one conservation alternative available for reducing water heating costs on farm dairies. A theoretical energy balance was conducted, from which the potential of recovering refrigeration condenser heat was estimated at up to 60% of the water heating energy requirements.

Preliminary tests with heat exchangers lead to the use of a tube-in-tube, counter flow, heat exchanger with fins on the refrigerant side, and cores on the water side, to improve the heat transfer characteristics. The exchanger, designed to provide 300 litres of 60°C water from a 2.25 kW refrigeration system cooling 2000 litres of milk per day, had an area of 0.84 m^2, and an overall thermal conductance of 100 $W.m^{-2}.^{o}C^{-1}$. This heat exchanger was inserted between the compressor and condenser of the refrigeration plant and tested with two condenser systems (air and water), four condenser pressures (6.5 bar, 7.5 bar, 10 bar and 12 bar), two milk inlet temperatures (23°C and 18°C), and two milk final temperatures (4°C and 7°C). In addition, tests on receiver pressure and suction superheat were performed to determine overall system performance.

Increasing condenser pressure increased cooling times from 2 hours 32 minutes to 3 hours 17 minutes, after the completion of the 1200 litre morning milking (thus failing to comply with the 3 hour cooling regulation at high condenser pressures.) Also, C.O.P. decreased from 3.05 to 2.35 for the water cooled condenser system (2.70 to 2.00 for the air cooled condenser system due to fan power consumption). Gross heat recovery rose from 4.2 $kWh.day^{-1}.m^{-3}$ to 8.1 $kWh.day^{-1}.m^{-3}$ for the water cooled system, giving water outlet temperatures of 45°C to 64°C as condenser pressure rose. The corresponding ranges for air cooled condensers were 3.8 $kWh.day^{-1}.m^{-3}$ to 6.6 $kWh.day^{-1}.m^{-3}$, and 38°C to 55°C. Changing milk inlet and final temperatures gave a proportional change in cooling times and total heat recovery, but had no effect on C.O.P. or heat recovery rates. Suction superheating increased total heat recovery by 15%, and water outlet temperatures by 9%.

Increases in gross heat recovery with increasing condenser pressure were partially offset by additional compressor power, and yielded nett heat recoveries of 4.0 $kWh.day^{-1}.m^{-3}$ to 6.0 $kWh.day^{-1}.m^{-3}$ for water cooled, and 3.6 $kWh.day^{-1}.m^{-3}$ to 4.3 $kWh.day^{-1}.m^{-3}$ for air cooled, condenser systems.

The maximum gross and nett heat recoveries (at 12 bar condenser pressure) were applied to the energy requirements of a monitored 220 cow town supply dairy. This analysis showed that the gross heat recovery was 51% of the water heating requirements, but the nett heat recovery dropped to 17% of the total heating and refrigeration demand. Based on current electricity and equipment prices, it is estimated that the payback period for this level of recovery would be 16-17 years. Changing the electricity pricing structure, to reflect up to a 1:3 differential in favour of water heating power costs, results in the 6.5 bar condenser pressure giving optimum results, but the nett returns are significantly lower than those reported.

The potential for improved savings is greater from larger capacity systems as the capital investment is not proportionally increased with an increase in scale.

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CHAPTER 1

INTRODUCTION

1:1 NEW ZEALAND DAIRY INDUSTRY

The New Zealand Dairy Industry exported dairy products worth \$685.9 million in 1980, representing 14% of the Nation's total export earnings (N.Z. Monthly Statistics Abstract, 1981).

Milk for processing into export and domestic products is collected from 1.87 million cows on 14,962 seasonal supply farms. In addition, 167,600 cows, from 1,544 town supply herds, provide the daily fresh milk requirements of the domestic market, with the excess supplementing the processed milk industry (N.Z. Dairy Board, 1980).

Stringent hygiene regulations and increased herd sizes (N.Z. Dairy Board, 1980) have required the development of sophisticated milking plants and cleaning systems, both resulting in a substantial increase in energy inputs on a per farm basis. The dairy industry's present electricity requirements are estimated at 3.20×10^{8} kWh or \$19.54 million per year (Chapter 2:2.1).

Inflation, intensification and increased herd sizes will continue to increase both the total electrical demand and the potential for savings by providing the incentive to use equipment which will improve electricity utilization on a small number of large farms.

1:2 PRESENT ELECTRICITY USAGE

It has been estimated (Section 2:2) that, on average, 40-45% of the total power consumption is used for water heating and 18-25% for refrigeration. Based on the total usage and cost (Section 1:1), this represents approximately \$4.22 million (at 3.1 ¢/unit) for heating and \$6.19 million (at 9.0 ¢/unit) for refrigeration with the remaining \$9.13 million (at 9.0 ¢/unit) being spent on milking plant, yard washdown and other miscellaneous functions, such as lighting.

1:3 SOURCES OF ENERGY

The single major cost component of on-farm electricity is heating water for plant cleaning. Prior to the fuel shortages of the early 1970's electricity was considered the most suitable means of heating water, but rapidly increasing prices have increased the interest in alternative sources of energy for this operation. Prominent among these have been solar heating and refrigeration heat recovery.

Solar heating, although successful, has the disadvantages of requiring a significant capital outlay and is of variable capacity. There are also several siting and operating requirements which may not always be suitable to the farmer (Studman, 1980).

Refrigeration heat recovery recovers the heat normally rejected to the atmosphere during the milk cooling operation. The heat is transferred to the water via a heat exchanger installed as part of the refrigeration circuit. This system has the advantages of; low increased capital cost (compared to solar systems), readily available and reliable heat all year round, no major siting limitations, and providing hot water at temperatures up to 60^{o}C. In addition, average water heating savings of 60% (Section 2:6) can be achieved compared to 35% for solar heating. Savings of 60% in water heating costs represents $2.53 million per year. Government policy of increasing electricity usage, while at the same time increasing revenue to pay for power development, means that heat recovery has the potential for greater savings.

Heat recovery systems are only available, at present, as add-on units for air cooled refrigeration systems. The potential for improved performance in heat recovery, using a water cooled condenser to preheat the water for a primary heat recovery heat exchanger, has not been investigated for N.Z. conditions.

In addition, only limited information is available on:-

1) alternative refrigeration heat recovery systems ,

2) the effect of heat recovery on milk cooling temperatures and rates,

3) the effect of changing operating conditions on system performance,

4) the economics of heat recovery.

1:4 RESEARCH OBJECTIVES

The lack of information has made guidelines for the industry difficult to formulate. Consequently, the objectives of this project were:

1) To design and construct a primary heat exchanger which would efficiently recover waste heat from a refrigeration system.

2) To determine the effect of changing operating conditions on system performance for a dairy shed refrigeration system modified to harness waste heat.

3) To establish operating conditions for the most promising systems of heat recovery which will produce the required volume of hot water to regulation temperature for twice per day plant cleaning.

4) To assess the economic viability of a correctly designed and operated system.

CHAPTER 2

REVIEW OF LITERATURE

2:1 DAIRY FARM MILK QUALITY

High levels of hygiene and quality control are required at all levels of milk processing to ensure that the resulting products meet export standards. Since the achievable standards are limited by the quality of the collected milk, on-farm processing must minimise the level of deterioration.

To meet these quality requirements, standards for milk plant cleaning and milk cooling have been set by the New Zealand Government, in conjunction with the Dairy Division (D.D.) of the Ministry of Agriculture and Fisheries (M.A.F.), and these are outlined below.

2:1.1 Milk Quality Standards

All milk received by a processing plant or milk treatment station is graded on the basis of bacterial contamination, the level of sediment and the presence of inhibitory substances such as antibiotics.

The milk is graded into three classes, namely; finest, first and second grades based on the results of standard tests (Dairy Industry Regulations, 1977). Table 2:1 presents the mandatory and advisory tests to be conducted and the limits for each grade (D.D. Circular 80/29, 1980).

To encourage farmers to maintain the standards set by the regulations a differential payment scheme operates as part of the grading system. The financial penalty involved is such that the level of payment for first grade milk will be 3% below that for finest grade milk and for second grade milk, at least 10% below finest grade (Dairy Industry Regulations, 1977).

2:1.2 Milk Quality Control

The biggest factor in milk quality is the level of bacterial contamination which, if not kept within satisfactory limits, reduces the processing potential of the milk and hence incurs

TABLE 2:1

Quality standards, test frequency and penalty levels for raw milk

	MANDATORY TESTS AS AT 1 JUNE 1981				ADVISORY TESTS
Test	Standard Plate Count (SPC)	Inhibitory Substances	Sediment	Organoleptic	Thermoduric & Coliform
Standard	Finest: less than 100,000 colonies/ml* First: between 100,00 and 200,000 colonies/ml Second: over 200,000 colonies/ml	Finest: less than 0.003 I.U. penicillin/ml	Finest: 7.5 mg First: 7.5, 15 mg Second: 15 mg	Finest: No defects determined in fresh samples examined at 40°C after heating	Thermoduric Finest: less than 5,000 colonies/ml Coliform Finest: less than 100 colonies/ml
Test Frequency	Three tests per month on a random basis with at least one test per 10 day period. On a consignment that has failed to obtain finest on a random test, continuous daily testing is conducted until 3 consecutive Finest results are obtained.			Per consignment	At company discretion. These tests are helpful on SPC failures.
Penalties	Penalties are applied to any consignments that fail. The minimum penalties that must be applied are found in the Dairy Industry Regulations 1977, regulation 36. For samples that fail to obtain finest grade with the inhibitory substances, sediment or organoleptic tests, penalties may be applied at company discretion, but not less than the minimum specified for the first grade.				Penalties may be applied by individual companies.

* To be reviewed after first year's operation.

financial penalty not only to the supplier but also to the processor.

The control of bacteria levels is achieved through plant cleaning, which removes potential milk contaminating bacteria from the milk plant, and milk cooling, which inhibits bacterial growth in the milk (Currier, 1976).

2:1.2.1 Plant Cleaning Procedures

Government regulations (Milk Production and Supply Regulations, 1973) specify that:-

1) Milking plants will be cleaned after each milking.

2) An adequate supply of clean water, suitable for plant cleaning, will be provided.

3) An approved plant for heating of sufficient water for cleaning purposes will be installed. The volume and temperature settings of the thermostat will be as prescribed by D.D.

4) Only detergents, chemicals or chemical compounds approved by D.D. shall be used for cleaning purposes.

In conjunction with these regulations, D.D. has specified (N.Z. Gazette Notice, 1973; D.D. Circular 80/16, 1980)

1) Standards for hot water cylinder construction.

2) The volumes of hot water required based on the size of the milking plant, milk storage facilities and cleaning system.

Further discussion on item 2) will be presented in Section 2:2.2.

The responsibility for approving cleaning systems has been delegated to the National Dairy Laboratory (N.D.L.) by D.D. If, after testing, a product and cleaning system satisfies the necessary criteria, N.D.L. makes the appropriate recommendation to the Director of D.D. (Arnott, 1982).

As of January 1982, four cleaning systems at three temperature ranges have been approved. These have been reported by Heyes et al. (1980) and are summarised in Table 2:2. The approval of any cleaning system does not require the farmer to comply with it but merely defines the procedure by which the approved cleaning system will give the required level of hygiene.

2:1.2.2 Milk Cooling

Milk bacteria have a range of ideal growth conditions. Currier (1976) stated that milk bacteria grow best between $10^{\circ}C$ and $37^{\circ}C$, but below $15^{\circ}C$ the rate is reduced (Figure 2:1). Currier also noted that associated with any change in conditions there was a time delay or lag phase before growth was resumed. The duration of the lag phase varied depending upon bacteria type and temperature. For each bacteria type, Currier found that below $5^{\circ}C$ the bacteria went into hibernation regardless of the stage of development. If the temperature was to rise again, above $10^{\circ}C$, growth would resume at whatever stage the bacteria had reached (Figure 2:2). Currier concluded that in order to inhibit the deterioration of the milk it is necessary to cool it rapidly to below $7^{\circ}C$.

The resulting regulations (Milk Industry Regulations, 1977) specify that the temperature of the milk supplied to a processing plant shall be reduced to $7^{\circ}C$ within 3 hours of the completion of milking and maintained at that temperature until the milk is accepted by the factory. (For town milk suppliers the requirement is $5^{\circ}C$ within 3.5 hours of the start of milking.) Problems of milk collection before the 3 hour cooling time and rising

TABLE 2.2

Categories of approved cleaning systems
and temperature ranges

	Cleaning Temperature		
Cleaning System	Above 65°C	50-55°C	20°C *
Triple System	0	X	X
Quatenary Ammonium Compound (Q.A.C.) System	0	0	0
Alkali System	X	0	0
Strong Acid System	X	0	0

0 = Suitable system
X = Not recommended

* Cold systems require a hot wash once per week.

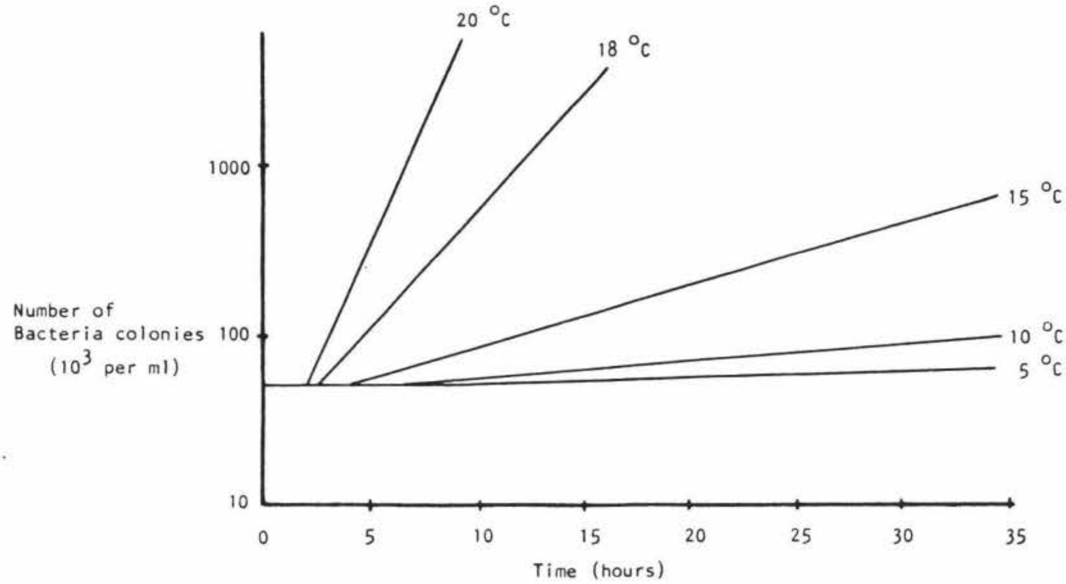


FIGURE 2:1

Growth rates of milk bacteria for various temperature conditions (after Currier, 1976)

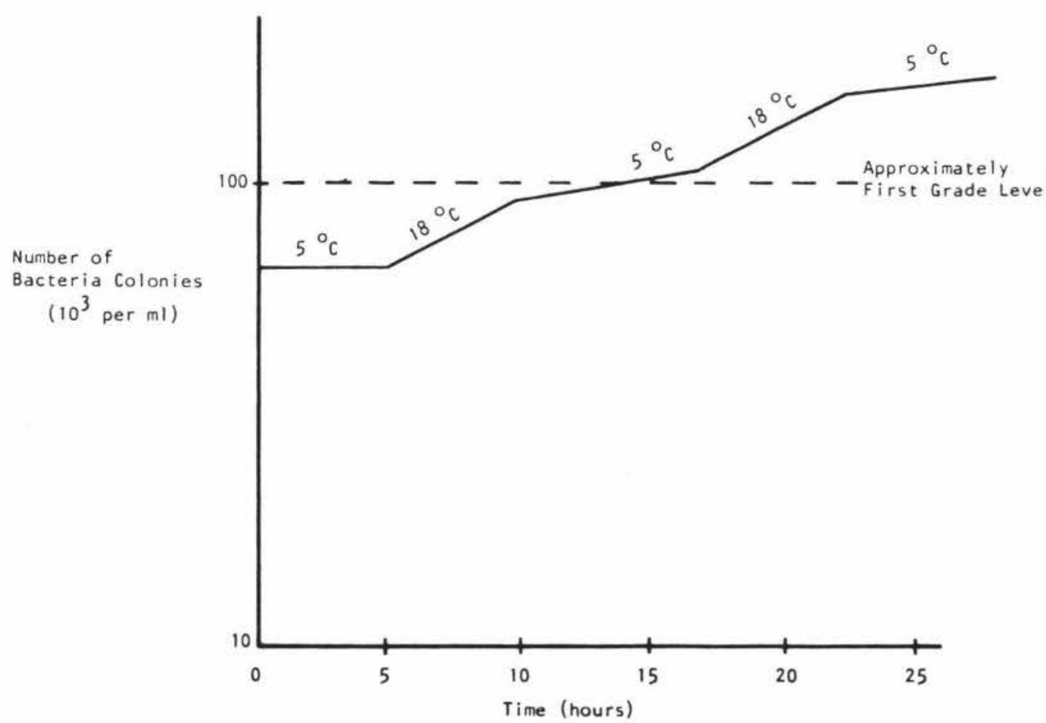


FIGURE 2:2

Growth response of milk bacteria (after Currier, 1976) temperature in bulk tankers ($3^{o}$to $4^{o}C$ over long hauls) have not been resolved. At present, rapid processing (within 2 hours of arrival) or additional factory refrigeration capacity maintains milk quality. However, there are suggestions of on-farm cooling to $4^{o}C$ which would eliminate the factory problems but increase on-farm refrigeration requirements (Vickers, 1980).

Internationally, the basic standard is cooling to $4^{o}C$ - $5^{o}C$ with the time limits varying from 0.5 - 4 hours.

In practice, milk cooling in N.Z. is a two stage process of:-

1) Reduction in milk temperature from $37^{o}C$ to approximately $18^{o}C$, or within $2^{o}C$ of the water inlet temperature, in an in-line counter flow water/milk plate heat exchanger connected between the milk pump and storage vat (Carter and Fisher, 1982). This reduces the refrigeration requirement by 50%.

2) A refrigeration system (as described in Section 2:4.2) which has sufficient capacity to meet the regulations.

The use of in-line plate coolers overseas is not as wide spread as in New Zealand, either because the technology has not become accepted (Roller et al., 1977; Wiersma et al., 1980; and Ewell et al., 1980) or because water temperatures or availability makes milk precooling uneconomic (Wiersma et al., 1980; Parkinson, 1982). Parkinson suggested that nocturnal cooling of stored water, for use in a plate cooler circuit, was a viable alternative, based on results from a two farm study where nocturnal water cooling to 10^{o}-$12^{o}C$ was achieved.

2:2 DAIRY SHED RESOURCE REQUIREMENTS

2:2.1 Total Electricity Usage

Total electrical usage in the New Zealand farming sector for the year ending 31 March 1981 was approximately 4.272×10^8 kWh (N.Z.E.D., 1981) of which 3.20×10^8 kWh (75%) (Carter and Fisher, 1982) was used by dairy farmers.

2:2.2 Water Heating Requirements

Rennie (1979) estimated, from results of a 9 farm survey, that 49% of total electricity requirements was used for water heating.

Estimates for overseas dairy operations range from 16% (U.S.D.A., 1977) to 50% (Ubbels and Bouman, 1979; Cox, 1979) with the average being 40%-45% (Peterson and Kolesch, 1979; British National Agricultural Centre, 1977; and Newell, 1980).

The amount of electricity used for water heating is a function of the water temperature and volume required.

2:2.2.1 Water Temperature

Water temperatures required for N.Z. conditions are dependent upon the specifications of the cleaning system, as approved by N.D.L., and vary considerably from product to product within the categories listed in Table 2:2. In general, the triple system requires an 80°C hot rinse, the triple and hot Q.A.C. systems require a 65° - 70°C wash, and the medium temperature detergents require a 50° - 55°C wash (D.D. Circular 80/16). Water temperatures for cow preparation are of the order of 20° - 25°C, depending on the discharge temperature from the plate heat exchanger (Massey University, 1972).

North American requirements are 70° - 75°C for plant cleaning and 35° - 45°C for cow preparation (Roller et al., 1977;

Wiersma and Armstrong, 1979; Thompson and Fairbank, 1979; and Peterson, 1979). British and European requirements are similar:- 80°C for plant cleaning and 35-45°C for cow preparation (Fleming and O'Keeffe, 1977; Ubbels and Bouman, 1979).

2:2.2.2 Water Volumes

The volume of water required by N.Z. dairy farmers is dependent upon the cleaning product and system selected. In general, they follow the Regulations (N.Z. Gazette, 1973) and are:-

10 litres/set of cups (14 litres for reverse flow)

50 litres for ancillary equipment

50 litres for the first vat up to 2500 litres

30 litres for subsequent vats up to 2500 litres

5 litres/500 litres capacity for each vat above 2500 litres.

For example, a 200 cow herd with 10 sets of cups and 2 vats, one of which has a capacity of 3000 litres, would require per day:-

2 milkings X 10 litres/set X 10 sets
+ 50 litres for ancillary equipment
+ 50 litres Vat 1
+ 30 litres Vat 2
+ 2 increments of 500 litres X 5 litres/increment

= 200 + 50 + 50 + 30 + 10

= 340 litres/day or 1.7 litres/cow.day

When plate cooler discharge water is used for cow preparation, no heating costs are involved. However, calculation of an average figure will make comparisons between researchers easier. For example, the 200 cow herd above, would produce, on average, 2200 litres of milk per day. A milk to water ratio of 1:2 would require 4400 litres of water, or 21 litres/cow.day.

Overseas water requirements vary considerably, with figures ranging from 5 to 24.5 litres/cow.day being quoted for both cow preparation and plant cleaning (Wiersma and Armstrong, 1979; Wiersma et al., 1980; and Peterson, 1979). Wiersma and Armstrong did point out that there was a wide variation in the quantity of hot water used by farmers and that most farmers were using more than was required.

2:2.3 Refrigeration Power Requirements

Refrigeration electricity requirements are based on cooling loads as discussed in Section 2:5.1. For New Zealand conditions this means cooling from approximately 18°C to 7° C or 4°C, and for overseas conditions, not using in-line milk precoolers, from 33°C to 7° C or 4°C.

Rennie (1979), in his survey, found that 18% of the total power requirements were used for refrigeration. Carter and Fisher (1982) suggest a figure of 18%-25%.

Figures quoted by overseas workers for cooling the total load by refrigeration range from 33% to 75%, with the average being about 40% (Wiersma et al, 1980; Peterson and Kolesch, 1979; Hoards Dairyman, 1979; Cox, 1979; and Newell, 1980). Correcting for the effect of a milk precooler would result in a range of 20% to 45%, with an average of 25%.

The range of 20%-45% compares well with the 18%-25% range suggested for N.Z., especially when management and climatic variations are taken into consideration.

2:3 PRINCIPLES OF REFRIGERATION

The basic principles of refrigeration are assumed or can be found in standard texts (Dossat, 1981).

2:3.1 Summary of a Refrigeration Cycle

A typical refrigeration cycle for Freon 12 is presented as a pressure-enthalpy diagram (Figure 2:3). Pressure-enthalpy diagrams allow the determination of thermodynamic properties and the calculation of heat flows. The cycle is described in stages starting with the refrigerant at the compressor inlet port.

2:3.1.1 The Compression Process (State 1 to State 2)

The compressor compresses the low pressure low temperature superheated suction vapour at State 1 to State 2 under approximately constant entropy conditions (isentropic compression). The work done is expressed by:-

$$W = h_2 - h_1 \quad (kJ.kg^{-1}) \qquad .....\text{Eqn } 2:1$$

where W = work done $(kJ.kg^{-1})$

h_2 & h_1 = Specific Enthalpy $(kJ.kg^{-1})$

NOTE: By convention $h=0$ at a saturation temperature of $-40^{o}C$.

The work done on the refrigerant results in an increase in both pressure and delivery superheat temperature.

2:3.1.2 Desuperheating (State 2 to State 2a)

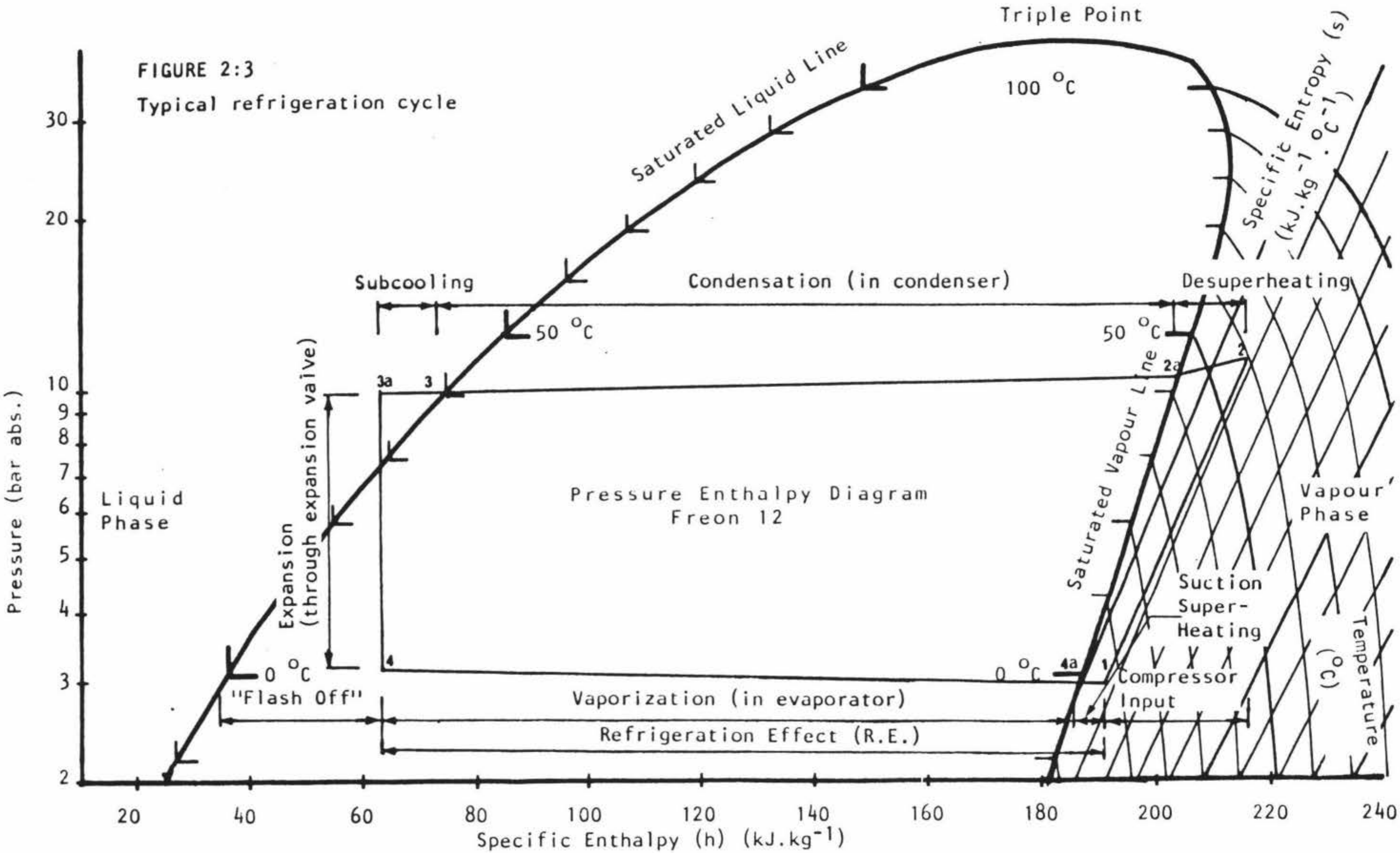
The condenser initially removes the sensible heat from the superheat vapour and changes the refrigerant from State 2 to State 2a. The vapour at State 2a is saturated and the pressure drop is due to losses in the discharge piping. The quantity of heat is expressed as follows:-

$$DSH = h_2 - h_{2a} \quad (kJ.kg^{-1}) \qquad .....\text{Eqn } 2:2$$

Although the quantity involved is small it is significant in that its removal in heat recovery situations provides

FIGURE 2:3

Typical refrigeration cycle



higher water temperatures than can be obtained from the condensation stage.

2:3.1.3 Condensation (State 2a to State 3)

Once the superheated vapour is cooled to saturation temperature (State 2a), further heat removal results in the condensation of the refrigerant at constant temperature. State 3 represents the state at which all the refrigerant has condensed, but it is still at saturation temperature. The heat removed is expressed as follows:-

$$\text{COND}\ (kJ.kg^{-1}) = h_{2a} - h_3\ (kJ.kg^{-1}) \text{.....Eqn 2:3}$$

2:3.1.4 Subcooling (State 3 to State 3a)

Removal of additional heat after condensation by either surplus condenser capacity or a liquid/vapour heat exchanger reduces the temperature of the liquid further. However, the quantity of heat is small. It is expressed by:-

$$SC = h_3 - h_{3a}\ (kJ.kg^{-1}) \text{.....Eqn 2:4}$$

Subcooling is desirable in that it reduces the degree of "flashing off" at the expansion valve and as a consequence increases the refrigeration effect (R.E.)

2:3.1.5 Expansion Process (State 3a to State 4)

The expansion of the refrigerant is approximately equal to an irreversible throttling process occurring without loss of enthalpy. The degree of "flashing off" is dependent upon the subcooled temperature and the pressure drop across the expansion valve.

2:3.1.6 Evaporation and Suction Superheating (State 4 to State 1)

Since the refrigerant is at saturation conditions at State 4, further addition of heat will result in vaporisation of the remaining liquid until State 4a is reached. Additional heating results in the superheating of the suction vapour to State 1. Heating of the suction superheated vapour is achieved by heating in the evaporator, the suction pipework or, if fitted, in the liquid/vapour heat exchanger.

If suction superheating produces useful cooling, then the R.E. is expressed by:-

$$\text{R.E.} = h_1 - h_4 \quad (\text{kJ.kg}^{-1}) \qquad \text{.....Eqn 2:5}$$
$$= h_1 - h_{3b}$$

If it is not, then

$$\text{R.E.} = h_{4a} - h_4 \quad (\text{kJ.kg}^{-1}) \qquad \text{.....Eqn 2:6}$$
$$= h_{4a} - h_{3b}$$

The conversion of these heat capacities (kJ.kg^{-1}) to heat flows (kJ.s^{-1}) is by including the flow rate of the refrigerant $(\text{kg}\cdot\text{s}^{-1})$.

2:3.2 The Effect of Changing Operating Conditions

To maximise heat recovery and water outlet temperatures it is desirable to increase the quantity and quality of the delivery superheat. Increasing delivery superheat can only be achieved by increasing condenser pressure or increasing suction superheating. However, changing either of these two parameters will have an effect on cycle efficiency and system performance.

2:3.2.1 The Effect of Changing Condenser Pressure

The effect of increasing condenser pressure is to reduce the capacity of the system. This occurs in three ways.

1) An increase in pressure differential across the expansion valve reduces R.E. (Figure 2:4).

2) Increase in Compression Ratio (C.R.) (ratio of compressor discharge pressure to suction pressure) requires more work to be done by the compressor.

3) Increase in C.R. decreases volumetric efficiency (Figure 2:5) (ratio of discharge capacity to swept volume) reducing refrigerant flow.

Increasing condenser pressure does, however, increase both the quantity of superheat and, more importantly, increases the temperature of the superheated vapour. This is an important aspect when considering the water outlet temperature of any heat recovery device. It must be remembered, however, that all the increase in delivery superheat comes from the compressor motor and, therefore, for every unit of additional energy supplied to the motor, only a percentage of it will be added to the vapour due to thermal and mechanical losses in the motor and compressor.

2:3.2.2 The Effect of Increasing Suction Superheat

Since the compressor is a vapour pump, a certain amount of superheating is required in practice to ensure that liquid does not enter the compressor. Increasing suction superheat forces the compression process to operate at a higher rate of specific entropy and, as a consequence, produces more delivery superheat at a higher temperature (Figure 2:6).

Since the entropy lines "fan out" with increasing condenser pressure, the amount of power required by the

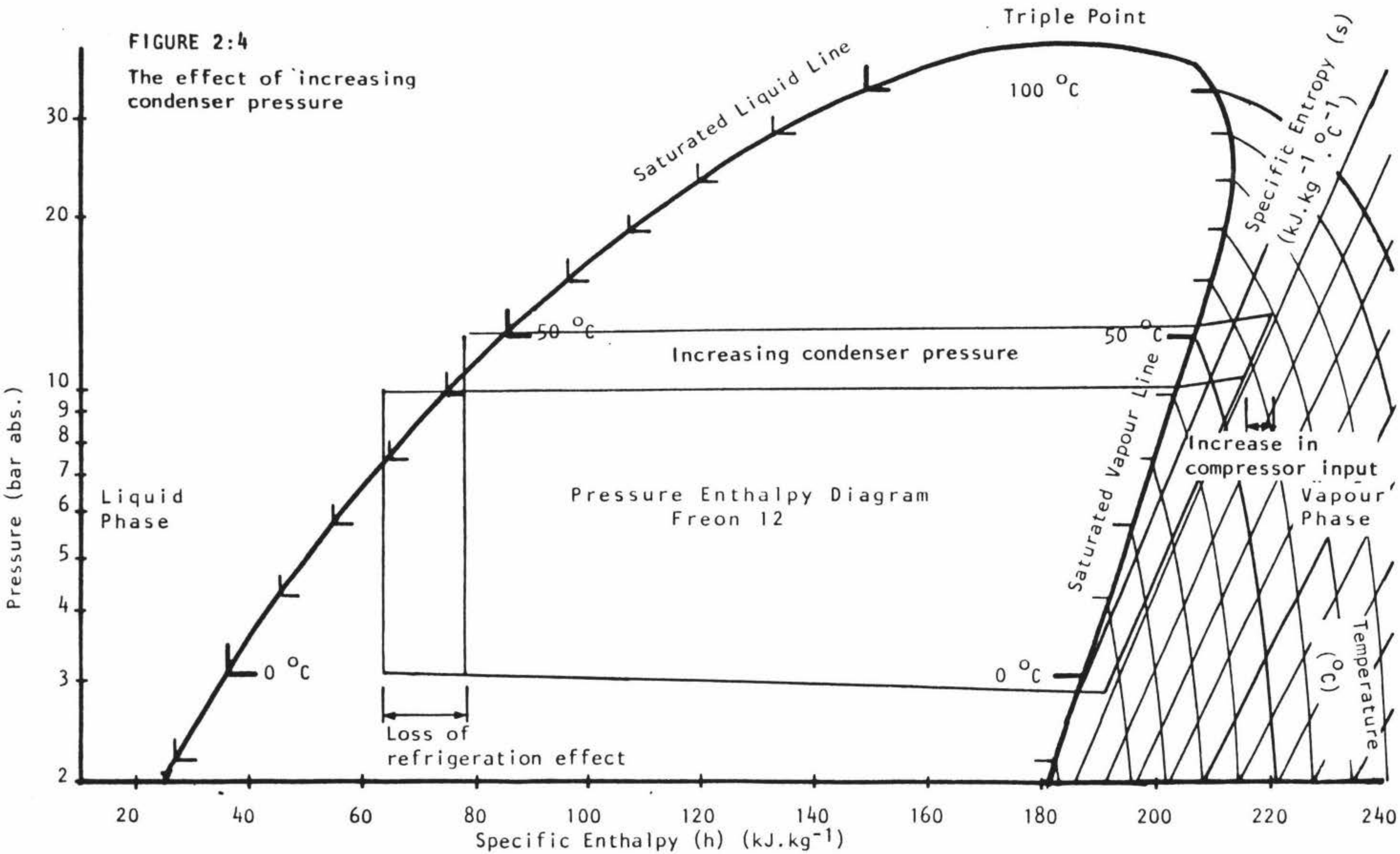


FIGURE 2:4

The effect of increasing condenser pressure

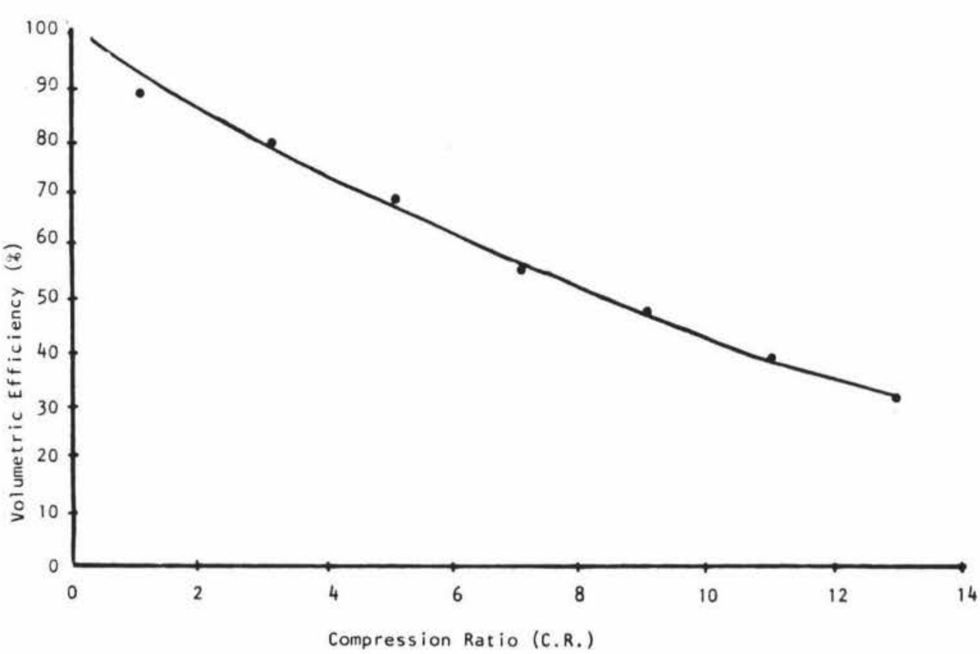


FIGURE 2:5

The effect of compression ratio on volumetric efficiency

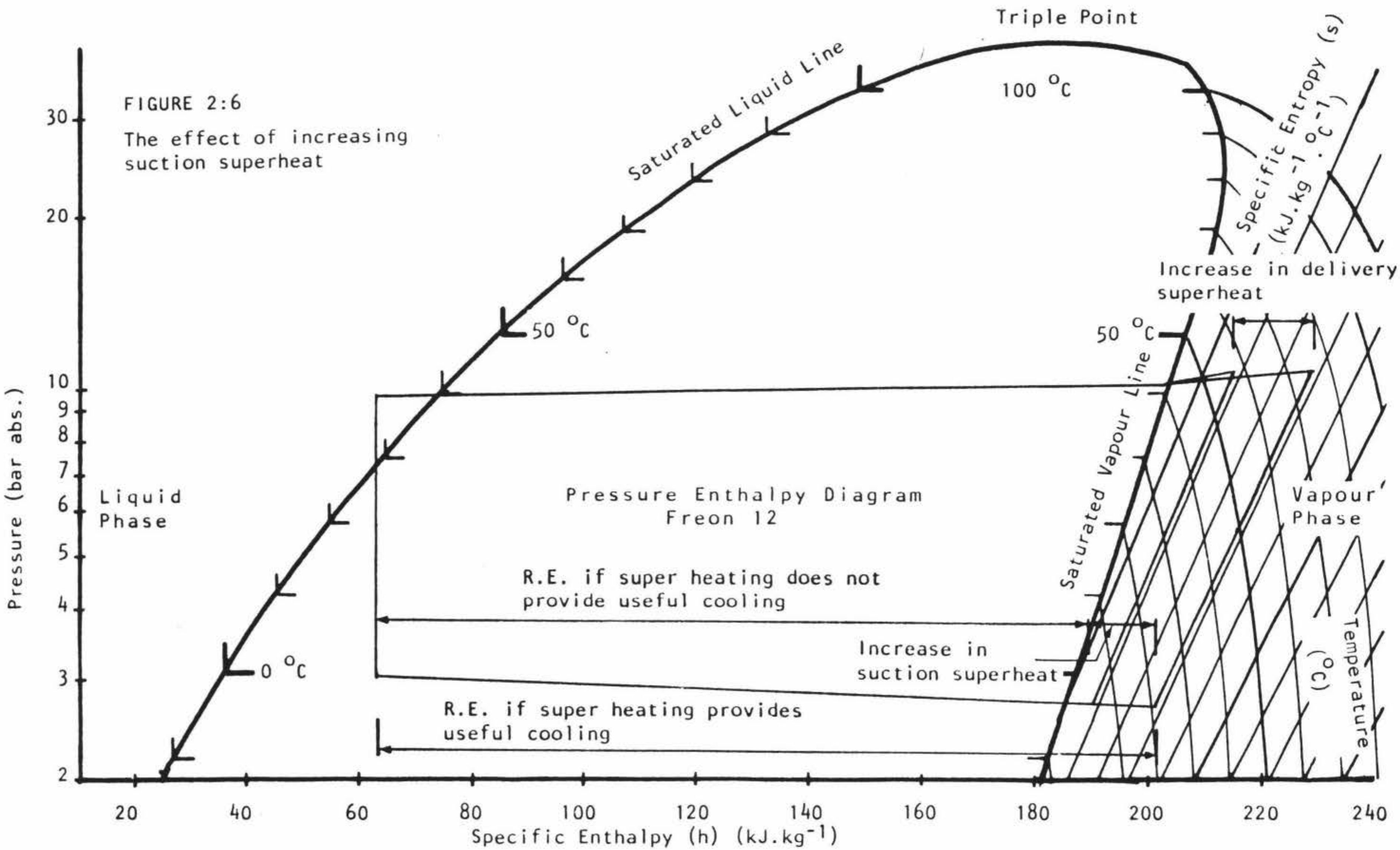


FIGURE 2:6
The effect of increasing suction superheat

compressor increases, but this increase is less than the amount of superheat gained. In addition, the temperature increase achieved is higher than the temperature rise in the suction line.

The way in which suction superheat is generated dictates the effect on cycle efficiency.

If superheating of the suction vapour with heat from a source outside of the evaporator occurs then the superheating does not provide useful cooling and must be excluded from the R.E. calculation. If, on the other hand, suction superheating provides useful cooling then the efficiency of the cycle is increased in spite of increased power consumption. Large scale superheating in the evaporator should be avoided because using evaporator area for heating vapour removes less heat from the load than does the boiling of the liquid. As a result, most thermostatic expansion valves are set to produce approximately $5^{o}C$ of superheat.

As previously discussed, subcooling of the liquid prior to the expansion valve decreases the amount of vapour "flash off". In conjunction with this, limited suction superheating is desirable and, therefore, it is usual to find a liquid/vapour heat exchanger installed in the evaporator refrigerant lines, as illustrated in Figure 2:7. The effect of this heat exchanger is shown in Figure 2:8.

Suction superheating has the additional effect of increasing the specific volume of the vapour. Since the compressor is a constant volume vapour pump , the increase in specific volume of the vapour decreases the mass capacity of the compressor. To maintain evaporator conditions, the expansion valve reduces the flow of refrigerant with the effect of reducing system capacity and efficiency.

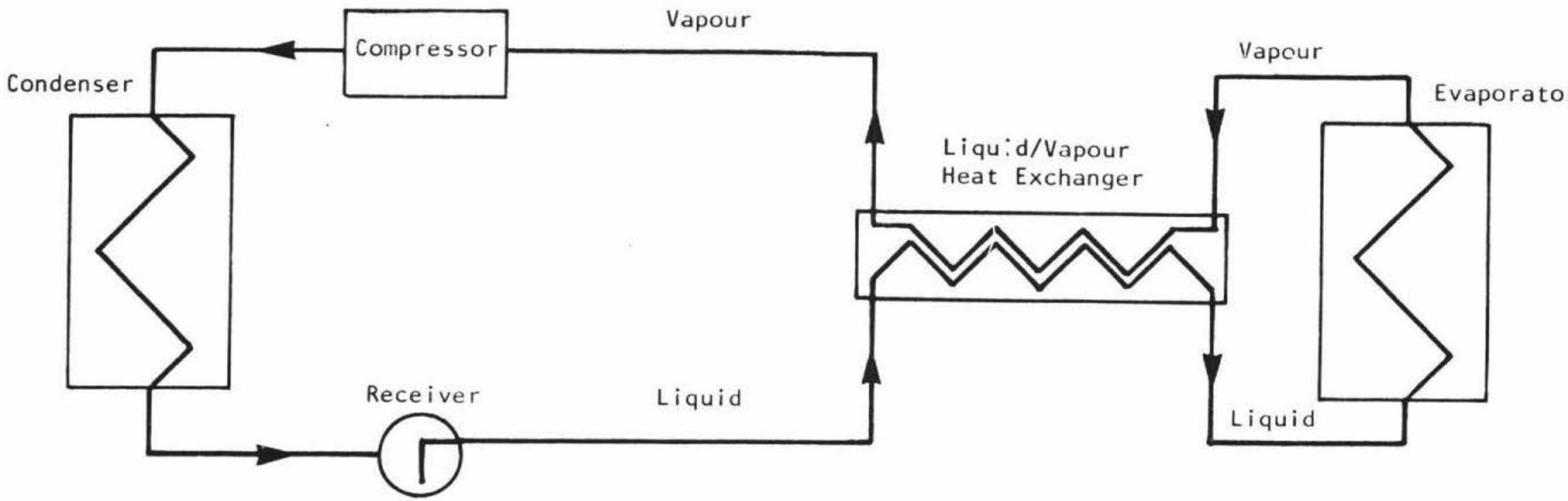


FIGURE 2:7

Schematic diagram of a refrigeration system
with liquid / vapour heat exchanger

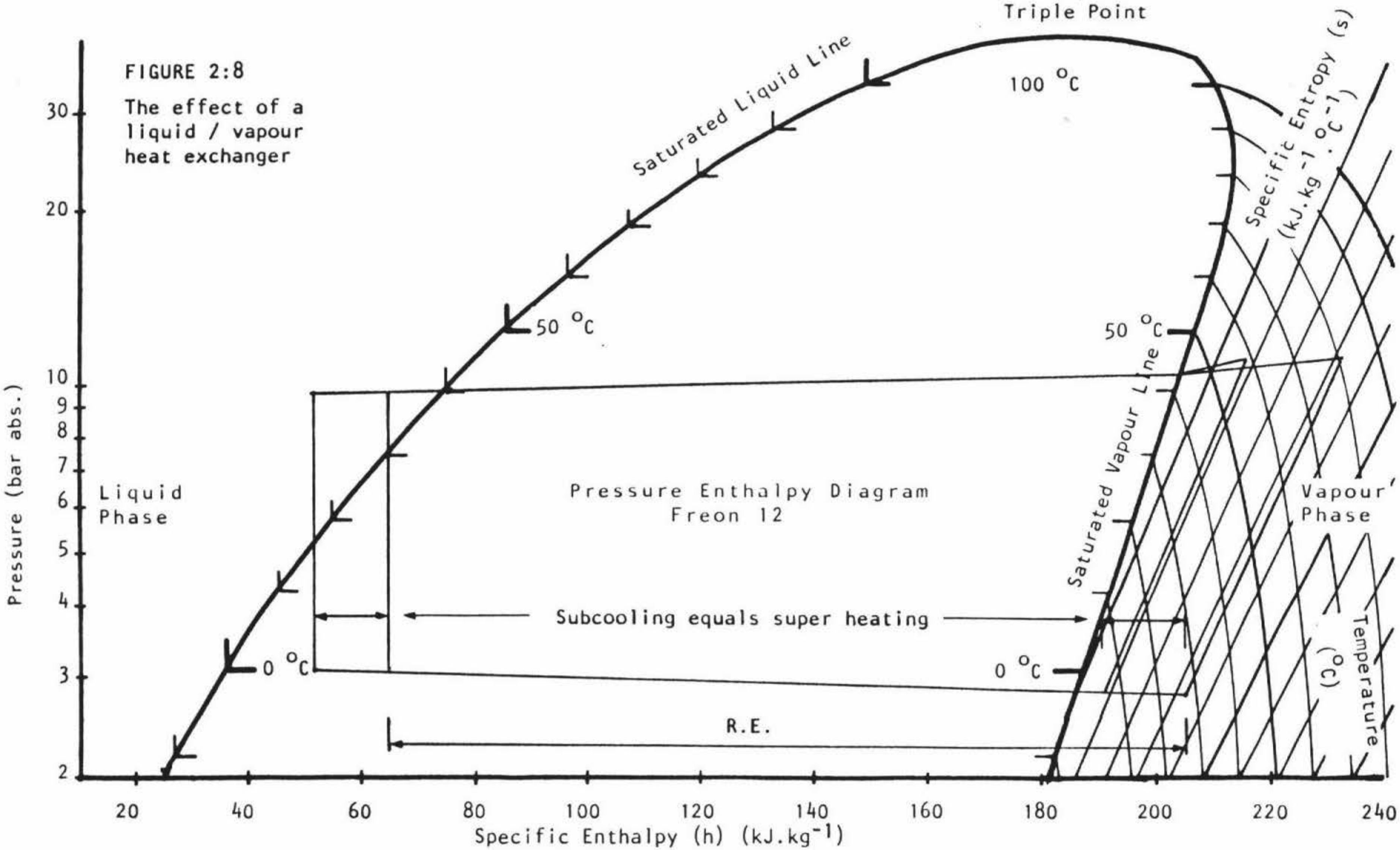


FIGURE 2:8
The effect of a liquid / vapour heat exchanger

2:3.3 Conclusion

Changing refrigeration conditions, as a means of increasing the delivery superheat for heat recovery purposes, can have detrimental effects on system efficiency and capacity, the magnitude of which can only be determined experimentally.

2:4 APPLICATION OF REFRIGERATION TO THE DAIRY INDUSTRY

2:4.1 Refrigeration Heat Loads

The total refrigerant load for the milk cooling system is comprised of the heat to be removed from the milk and the environmental load.

2:4.1.1 Milk Loads

Milk is collected from the cows twice per day at a temperature of 37°C, reducing to 33°C before entering the cooling system, due to temperature losses in the milk plant pipework.

Cooling milk from 33°C to 7°C (4°C)* requires that 109 $kJ.m^{-3}$ (121 $kJ.m^{-3}$) be removed. Since only 50% of this load is removed by refrigeration (Section 2:1.2.2), and the morning and night milkings produce 60% and 40% of the daily production respectively (Lovell-Smith, 1982; Belcher, 1979), refrigeration capacities of 14 (15) $MJ.m^{-3}.h^{-1}$ are required to comply with the regulations. Assuming a C.O.P. of 2.2 (Carter and Fisher, 1982), a compressor motor capacity of 1.8 (1.9) $kW.m^{-3}$ is required.

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* Throughout this section the first figure refers to cooling to 7°C, while the figure in brackets refers to cooling to 4°C.

2:4.1.2 Environmental Loads

Since milk is stored and cooled in vats for 1 to 2 days, heat transfer from the environment into the cooled milk is possible. The degree of environmental heating increases when the vat is uninsulated. Vickers (1980) found that only 0.38 $MJ.h^{-1}$ of additional electricity was consumed by the refrigeration system per m^3 of milk stored in an uninsulated vat cooled to 7^oC compared to a vat insulated with 50 mm of foam rubber. An additional 0.48 $MJ.m^{-3}.h^{-1}$ was required for a 4^oC final temperature.

Currier (1977) and Vickers (1980) both suggest that the environmental load is small and that changes in vat size do not significantly alter the general conclusion that vat insulation was not justified. Typical environmental heat loads are estimated to be 1.4 (1.5) $MJ.m^{-3}.h^{-1}$, resulting in a total refrigerant load of 15.4 (16.5) $MJ.m^{-3}.h^{-1}$.

2:4.2 Milk Refrigeration Systems

2:4.2.1 Ice Bank System

The ice bank system requires a spiral copper refrigerant coil to be immersed in an insulated storage tank such that ice, to a depth of 100 mm, can be built up between milkings (Figure 2:9). During milking the chilled water, at approximately 2^oC, is circulated at two to three times the milk flow through a second section of a plate heat exchanger so that it effectively chills the milk to approximately 4^oC. The cooled milk is stored in the vat until tanker pickup.

Ice bank systems are not popular in New Zealand but are relatively common overseas (Belcher, 1977). Patchett and Vickers (1973) and Currier (1977) did not favour the ice bank system because of milk temperature rises between cooling and tanker pickup. These workers also concluded

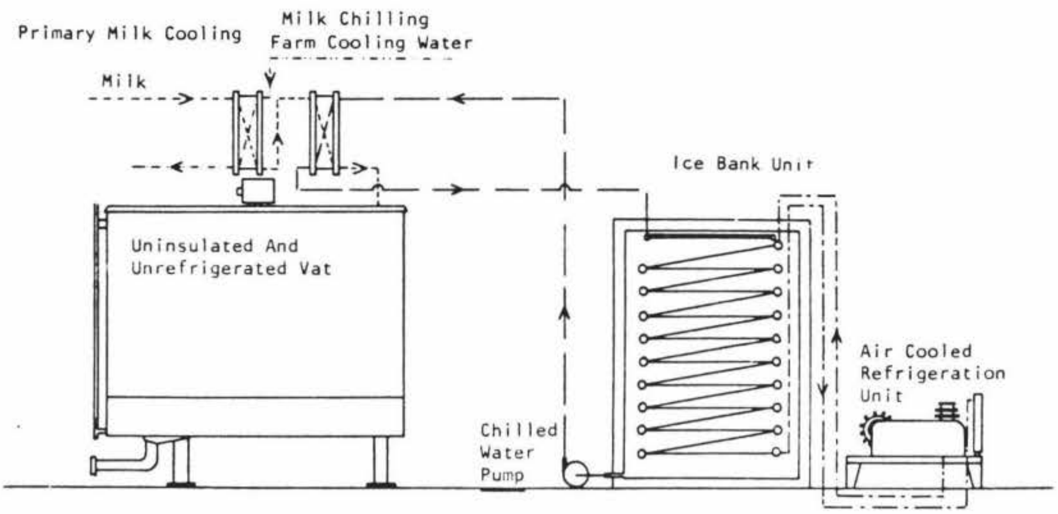


FIGURE 2:9

Schematic diagram of an ice bank chiller sytem

(after Patchett and Vickers, 1973)

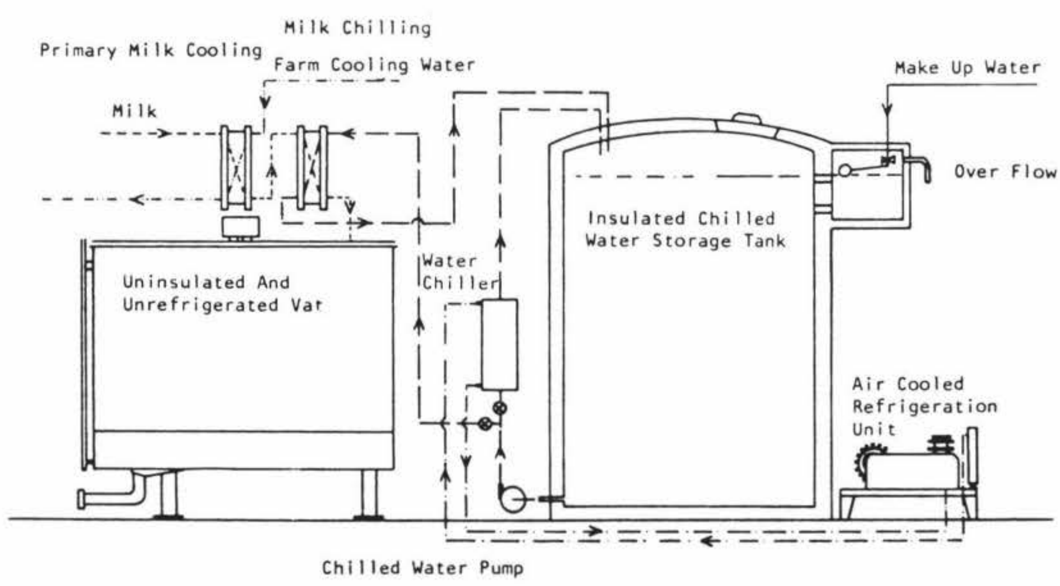


FIGURE 2:10

Schematic diagram of a water chiller system

(after Patchett and Vickers, 1973)

that high initial and maintenance costs made this alternative less attractive than other available systems.

Although less efficient (based on R.E.) than direct expansion units, Patchett and Vickers (1973) did concede that the use of off-peak electricity tariffs and the potential to use plants with a lower peak refrigeration capacity may offset much of the disadvantage of lowered efficiency.

Reported cooling capacities for ice bank systems ranged from 35 to 67 litres of milk cooled per kWh, with temperature changes of 33°C to 4.5°C (Belcher, 1979; Fleming and O'Keeffe, 1977).

Under New Zealand conditions Currier (1977) found that ice banks had a cooling capacity of 58.8 litres per kWh for milk cooled from 20°C to 7°C, and 22.2 litres per kWh for milk cooled from 37°C to 7°C.

2:4.2.2 Chilled Water Refrigeration System

The chilled water system (Figure 2:10) is similar to the ice bank system and both are forms of indirect expansion refrigeration.

Patchett and Vickers (1973) and Currier (1977) analysed this system and drew the same conclusions as obtained for the ice bank system.

2:4.2.2 Direct Expansion Refrigeration System

Direct expansion refrigeration systems for milk cooling require that the evaporator is bonded directly to the bottom of the vat and surrounded by insulation (Figure 2:11).

The advantages of this system are:-

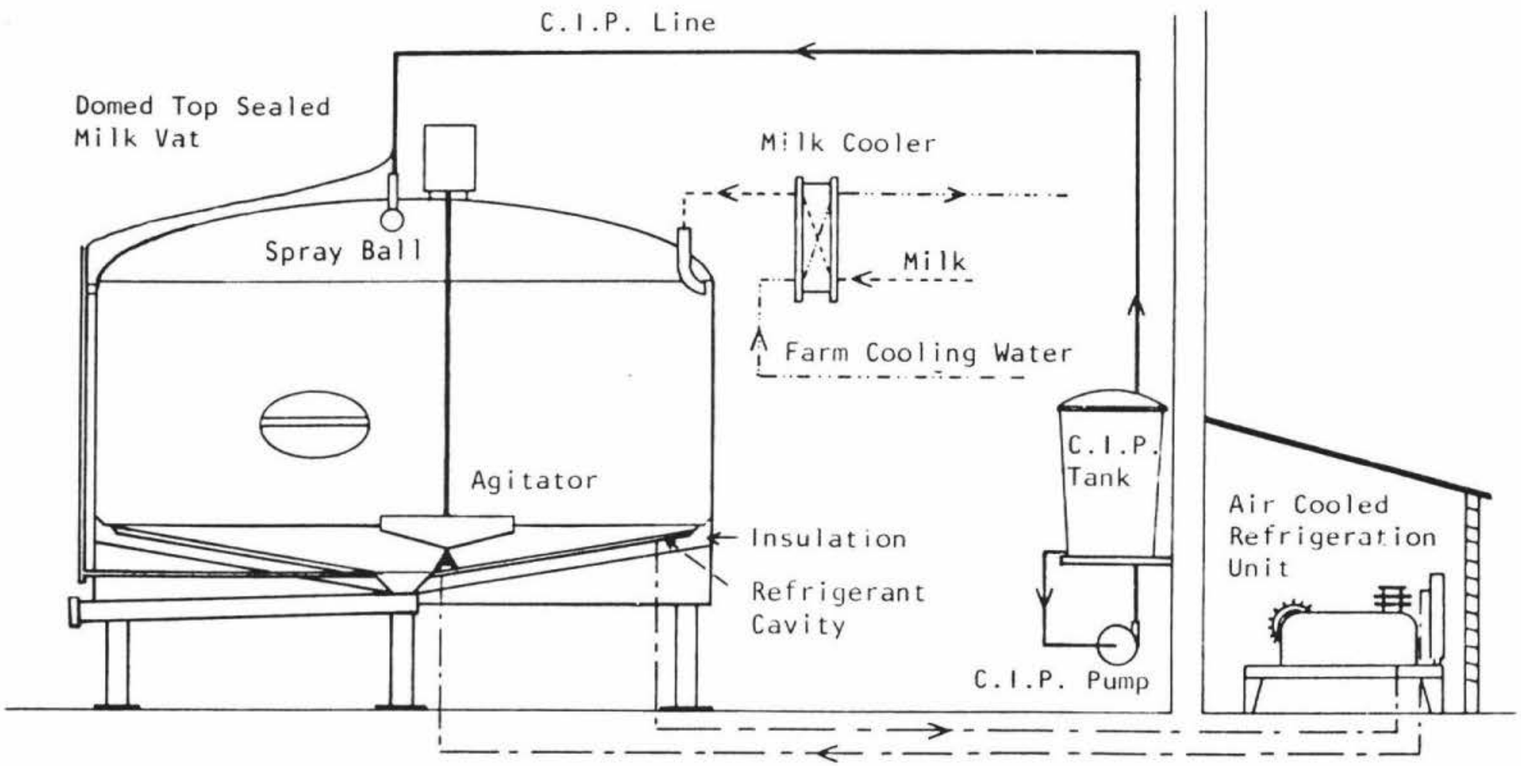


FIGURE 2:11

Schematic diagram of a direct expansion refrigeration system

(after Patchett and Vickers, 1973)

1) No ice, chilled water tanks or pumping systems are required.

2) No additional in-line heat exchanger is required.

3) In-vat refrigeration allows temperature control between milkings and before tanker pickup.

4) A more compact arrangement of equipment is possible.

5) Improved cooling efficiencies are obtained.

The disadvantages are:-

1) Increased compressor size.

2) Higher running costs if the system does not use off-peak power. (Common practice in New Zealand does not favour off-peak power.)

3) Condensers and compressors must be outside (for air cooled units), which may require long evaporator lines and loss of efficiency, due to refrigerant boiling before reaching the expansion valve.

Cooling capacities for direct expansion systems range from 54-90 litres per kWh for a temperature range of 37°C to 4.5°C (Belcher, 1977; Fleming and O'Keeffe, 1973; Currier, 1977). Currier also reported that cooling capacity for the 20°c to 7°C range was 200 litres per kWh. Direct comparison with ice bank systems is difficult, due to variation in operating conditions and system arrangements but, on average, direct expansion systems are 34% - 54% more efficient than ice banks (Belcher, 1979; Fleming and O'Keeffe, 1977) with extremes of 240% (Currier, 1977).

Of the three systems, Patchett and Vickers (1973) concluded that direct expansion vat refrigeration is the most satisfactory system, especially where milk precooling reduces the temperature to 18° - $20^{\circ}C$.

2:5 HEAT RECOVERY SYSTEMS

Early work on dairy shed heat recovery investigated both desuperheaters (Sturges, 1961; Turner, 1961) and complete condensing units (Aherns, 1959). In addition, two types of heat exchanger were developed, namely; a remote heat exchanger with water circulating in conjunction with a storage tank (Aherns, 1959) and a coil in the storage tank (coil-in-tank) (Sturges, 1961; Turner, 1961). More recently, once-through systems have been studied (Fleming and O'Keeffe, 1977; Carter and Fisher, 1979).

Comparison of the two major systems, i.e., desuperheaters or complete condensing, and the three heat exchanger systems ('once-through', circulating and 'coil-in-tank'), is difficult. Kolesch (1979) emphasised the difficulties when trying to compare results of the different systems. The major areas of difficulty were:-

1) Insufficient information, with regards to heat transfer area, did not allow comparisons to be made between heat exchangers on the basis of overall thermal conductance.

2) Insufficient information with respect to heat recovery heat exchanger mode of operation, i.e., 'once-through', circulating or coil-in-tank.

3) Variation in compressor head pressures and refrigerants, both affecting temperatures and available heat.

4) The variation in the ratio of water volumes heated to the volume of milk cooled.

For a scientific comparison to be made, experiments would have to be conducted under conditions of:-

1) Constant heat exchanger area

2) Constant water to milk volume ratio

3) Constant refrigeration operating conditions.

Since no work of this nature has been reported, accurate comparisons of the various systems is difficult and only a discussion of general trends will be presented.

2:5.1 Heat Exchanger Design and Performance

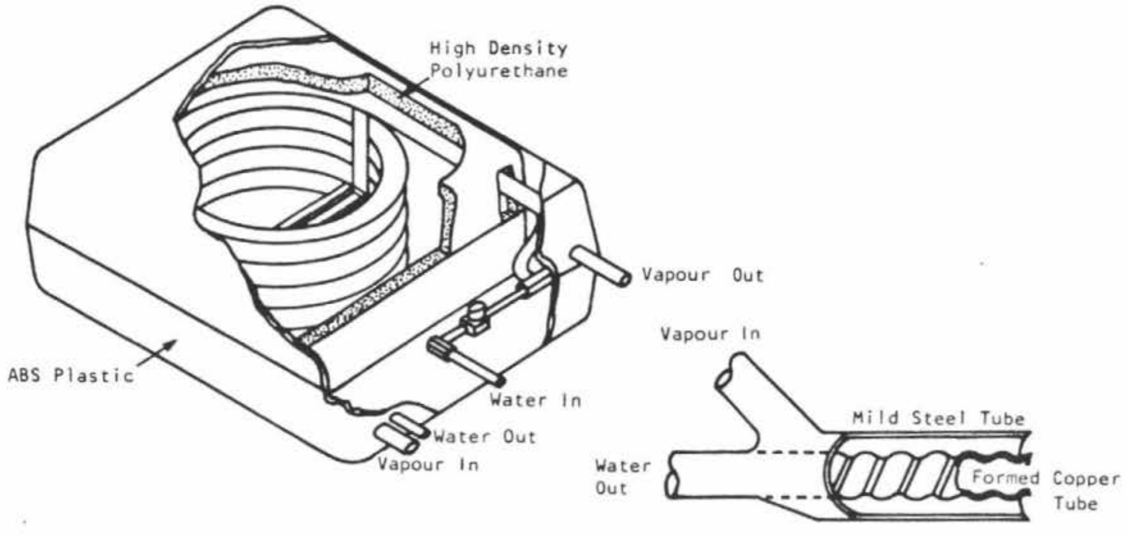
The three basic heat recovery heat exchanger designs are; tube-in-tube, shell and tube, and coil-in-tank (Figure 2:12).

2:5.1.1 Tube-in-Tube

The tube-in-tube heat exchanger is normally counter flow, with the refrigerant vapour in the outer tube or annulus and the water in the inner tube. This type of exchanger is popular because of its simple construction and relatively high heat transfer characteristics.

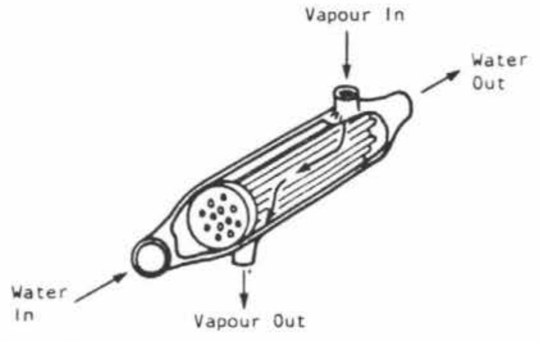
Performance characteristics of this design have been studied in New Zealand by Carter and Fisher (1979) using a commercially available heat exchanger with an area of 0.2617 m^2. Overall thermal conductance (U) was determined for a number of water flow rates and refrigerant head pressures, and for water passing 'once-through' and circulating. This data is presented in Table 2:3 and Table 2:4.

The results show that circulating the water gives considerably less heat transfer than the 'once-through' system does. This is to be expected from heat exchanger theory.

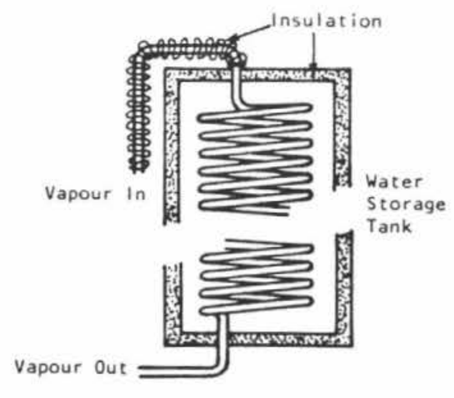


a) Tube-in-tube Heat Exchanger
(after Carter and Fisher, 1982)

Tube-in-tube End Construction



b) Shell and Tube Heat Exchanger



c) Coil-in-tank Heat Exchanger

FIGURE 2:12

Schematic diagram of three heat exchanger types

TABLE 2:3

Summary of U values and water temperatures for a tube-in-tube heat exchanger - water 'once-through' (Carter and Fisher, 1979)

Head Pressure kPa		Water Flow Rate $l.h^{-1}$	Water Temperature ^oC	U $W.m^{-2}.^oC^{-1}$
896	min.	35	58	180
	max.	220	51	950
830	min.	35	55	180
	max.	195	50	780
725	min.	55	43	175
	max.	180	33	600

TABLE 2:4

Summary of U values and water temperatures for a tube-in-tube heat exchanger recirculating 180 litres (Carter and Fisher, 1979)

Time h	Water Flow Rate $l.h^{-1}$: 220		550	
	U $W.m^{-2}.^oC^{-1}$	Water Temperature ^oC	U $W.m^{-2}.^oC^{-1}$	Water Temperature ^oC
0	1260	15	1700	15
1	480	35	650	38
2	380	41	500	43
3	300	44	-	-

2:5.1.2 Coil-in-Tank

Using a 270 litre storage tank and three copper coils, each with an area of 1.372 m^2, Cromarty (1968) tested five coil arrangements as illustrated in Figure 2:13. He concluded that arrangement E was the most efficient since it gave the greatest percentage of heat recovered. However, on the basis of U, arrangement C was the most efficient heat exchanger with a value of U = 148 $W.m^{-2}.^{o}C^{-1}$ compared with 66 $W.m^{-2}.^{o}C^{-1}$ for arrangement E. This is understandable in that the maximum refrigerant film coefficient will be for the arrangements which are not in parallel flow (i.e., D and E). The value of U, of 148 $W.m^{-2}.^{o}C^{-1}$, is 15% lower than the minimum value achieved in Table 2:3 (175 $W.m^{-2}.^{o}C^{-1}$). This difference is expected in that water film coefficients under conditions of laminar flow are relatively low in coil-in-tank heat exchangers.

2:5.1.3 Shell and Tube

Based on limited data (heat exchanger design texts) it is anticipated that a 'once-through' shell and tube heat exchanger would perform better than the coil-in-tank and recirculating tube-in-tube heat exchanger, but worse than the tube-in-tube 'once-through' type. The reason for this is that water flows would be split between tubes thus reducing film coefficients. The refrigerant film coefficient may also be less than the tube-in-tube type, depending on the intertube area of flow.

2:5.2 Desuperheater Heat Recovery System Performance

2:5.2.1 In-line Heat Exchanger - 'Once-Through'

The in-line heat exchanger is mounted in the refrigerant line between the compressor and the condenser. The water is passed 'once-through' on its way to the storage cylinder,

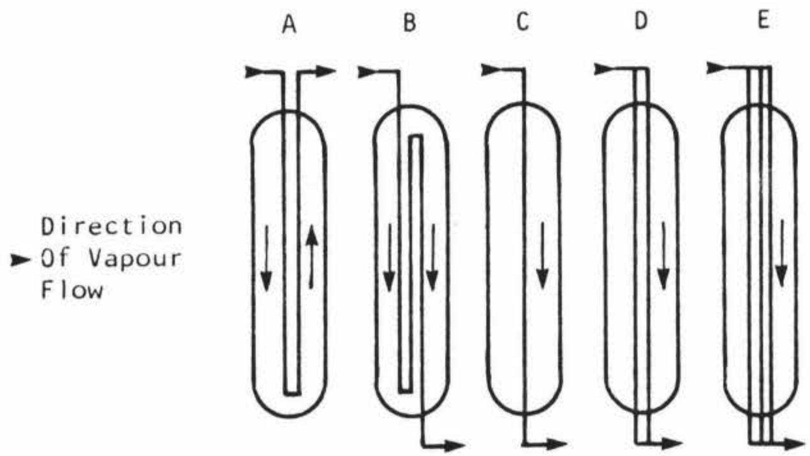


FIGURE 2:13

Schematic diagram of five coil arrangements for a coil-in-tank heat exchanger
(after Cromarty,1968)

at a flow rate that will give the required volume at the end of the cooling time (Figure 2:14).

Performance testing of this type of heat recovery system has been conducted in New Zealand by Carter and Fisher (1979, 1982). Results on water flow rates and temperatures are presented in Table 2:3. In terms of heat recovery efficiencies*, their system was only 20%-30% efficient.

C.O.P.'s for the tests conducted ranged from 2.6 to 1.8 depending on vat temperature, condenser pressure and ambient temperature. In general, C.O.P.'s decreased with decreasing vat temperature, increasing condenser pressure and increasing ambient temperatures.

However, the tests did not simulate normal milk loading operations (temperature and flow rates). Therefore, the results do not give a true indication of the effects of changing conditions on total heat recovery, cooling times and power consumption.

Vickers (1980) found that heat recovery, using and in-line heat exchanger, produced warm water at 41°C for a flow rate of 50 $l.h^{-1}$. The flow rate was designed to give 300 $l.day^{-1}$ for 6 hours cooling time. A total of 1650 litres was cooled from 18°C to 4°C, which resulted in a heat recovery efficiency of 33%.

2:5.2.2 In-line Heat Exchanger - Circulating

Carter and Fisher (1979, 1982) found that the circulating system tested was only 24% efficient in heat recovery. Water temperatures achieved at hourly intervals are presented in Table 2:4.

Kolesch (1979) found that heat recovery efficiencies of 25% were obtained with a similar system when heating

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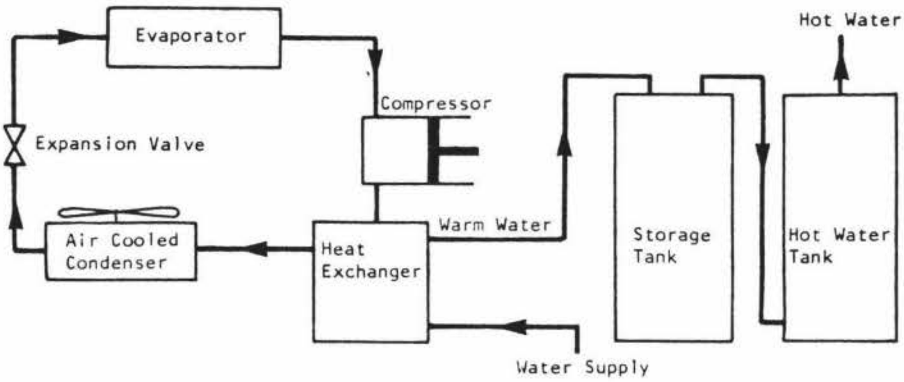
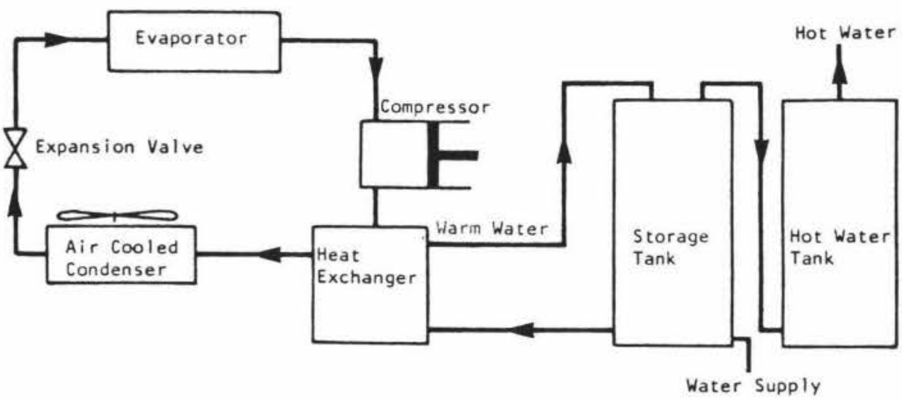
* Heat recovery efficiency is defined as the heat recovered expressed as a percentage of the heat removed from the milk.

a) "Once Through" System



b) Circulating System



c) "Coil-in-tank" System

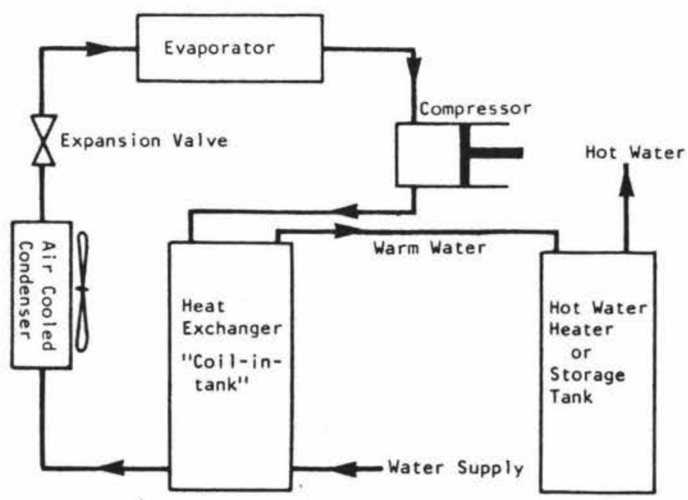


FIGURE 2:14

Desuperheating heat recovery systems retaining air cooled condenser water from $15^{\circ}C$ to $43^{\circ}C$ at condenser pressures ranging from 630 kPa gauge to 790 kPa gauge.

Kolesch also found that a shell and tube system was 18% efficient and was only capable of raising tap water to $38^{\circ}C$.

2:5.3 <u>Complete Condensing Heat Recovery Systems</u>

Complete condensing systems are similar to the desuperheating systems illustrated in Figure 2:14, except that the air cooled condenser is omitted in some installations.

2:5.3.1 Complete Condensing - 'Once-Through''

Fleming and O'Keeffe (1977) investigated a 'once-through' full condensing system and found that, at condensing pressures ranging from 1100 kPa to 1520 kPa, the amount of heat recovery ranged from 500 litres at $35^{\circ}C$ to 170 litres at $60^{\circ}C$.

Under normal operating conditions, in a milk cooling situation, the system tested produced water at $56^{\circ}C$. It was calculated to be 80% efficient in heat recovery. It was also found that 20% more power was required for the heat recovery system than that required for a similar sized unmodified air cooled system.

2:5.3.2 Complete Condensing - Circulating

Kolesch (1979) found that systems condensing the refrigerant by circulating water through a heat exchanger were able to recover 50-60% of the available heat. Efficiencies higher than this are difficult to achieve because increasing hot water temperatures caused the condenser pressure to increase. This is in agreement with results from the work of Carter and Fisher (1979), showing that heat transfer coefficients decrease with rising water temperatures.

Peterson (1979) also tested two examples of this system and found that 60°C water could be produced. However, there is insufficient information to calculate the efficiency of the system.

2:5.3.3 Complete Condensing - 'Coil-in-Tank'

Cromarty (1968) conducted tests on various coil arrangements (Figure 2:13) in a 270 litre cylinder under environmental temperatures of 10°C, 21°C and 32°C. The conclusions reached were that:-

1) A triple pass parallel flow heat exchanger (arrangement E) was the most efficient with 71% of the available heat being recovered as 270 litres of 59°C water.

2) Heat recovery increased compressor power consumption but reduced air condenser fan operation. (Fan switching was used as a means of controlling condenser pressures.)

3) Increasing condenser pressure to 1100 kPa allowed vapour pressures to reach 100°C. Under these conditions fans only operated for 30% of the time and satisfactory milk cooling rates were obtained.

4) Increasing environmental temperatures increased cooling times and power consumption. This is consistant with results obtained by Carter and Fisher (1979, 1980).

Ubbels and Bouman (1979) found that a complete condensing coil-in-tank system could economically produce 58°C water. This conclusion was based on the principle that heat recovery should only continue as long as it recovers more energy than the extra required by the refrigeration system. To determine this point, Ubbels and Bouman developed the following equation, which is derived in Appendix A2.

$$Ke = \frac{1 - \frac{COPRc}{COPRca}}{COPH} \qquad .....\text{Eqn } 2:7$$

where COPRc = COP refrigeration with heat recovery

COPRca = COP refrigeration without heat recovery

COPH = COP of heat recovery

Heat recovery is only economical for values of $Ke \leq 1$

Timmons et al. (1977) studied a simulated coil-in-tank system with and without a suction cooled compressor. The results obtained showed that 57% and 61% of the heat was recovered at 57°C and 61°C for non-suction and suction cooled compressors respectively.

The simulation indicated that total power consumption increased by approximately 31% , probably due to the higher head pressures and longer running times,as a result of the heat recovery exchanger.

2:5.4 Condenser Pressure Control

The need for maintaining minimum and maximum condenser pressures has led to the use of a number of condenser pressure control systems.

2:5.4.1 Minimum Condenser Pressure Control

The increase in condenser circuit area due to the addition of the desuperheater heat exchanger results in reduced condenser pressures, especially under low ambient temperature conditions. Condenser pressures below a certain point (the value varies with the refrigeration system) results in either low heat recovery temperatures or malfunctioning of the expansion valve, due to refrigerant boiling in the evaporator liquid line.

Control of condenser pressure is either by switching of electric fans (Kolesch, 1979) or by capacity regulating valves (Carter and Fisher, 1979). Fan switching has the disadvantage of insufficient motor cooling in non-suction cooled compressors, whereas capacity regulating valves, due to their high head loss characteristics, require a compressor/receiver by pass line and pressure regulating valve to maintain receiver pressure (Figure 2:15).

2:5.4.2 Maximum Condenser Pressure Control

All circulating and coil-in-tank systems either dump heat through an air cooled condenser (Cromarty, 1968; Ubbels and Bouman, 1979; Kolesch, 1979; Timmons et al., 1977) or dump hot water (Kolesch, 1979) as a means of controlling the maximum condenser pressure.

It is important to note that it is inefficient to recover high grade heat (water above 50^{0}C) to later dump it as a means of controlling condenser pressure.

2:6 EFFECT OF HEAT RECOVERY ON WATER HEATING

The savings in hot water heating costs gained from heat recovery have been reported for several systems.

Peterson (1979) found that heat recovery saved 74% for a 55 cow herd, 71% for a 200 cow herd, 71% for a 90 cow herd and 85% for a 250 cow herd. Kolesch (1979) gave a range of 24% to 85% savings from heat recovery for herds of 50 to 188 cows.

There is no New Zealand data to confirm if similar savings can be achieved under local management and climatic conditions. However, assuming similar conditions of:-

1) system capacity
2) water volumes and temperatures
3) heat loads
4) exchanger efficiency

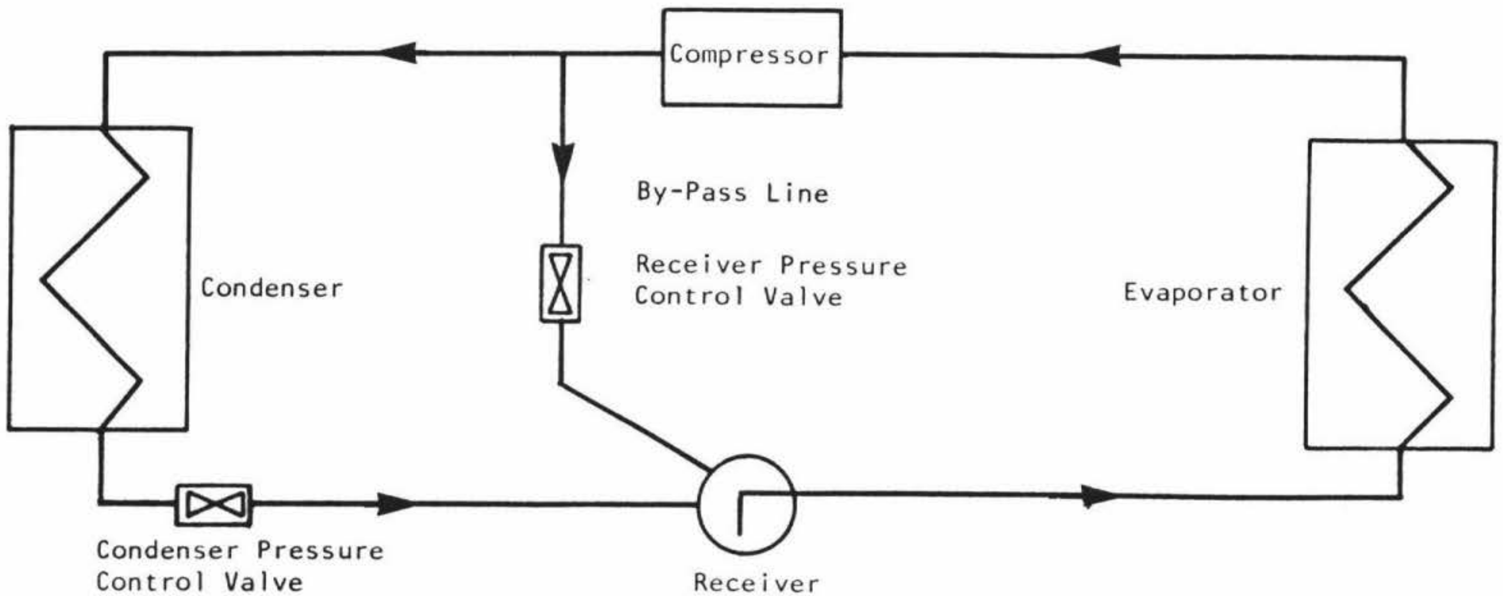


FIGURE 2:15

Schematic diagram of a refrigeration system with a refrigerant vapour by-pass line and receiver pressure control valve a theoretical estimate of 50%-60% may be made. Confirmation of this estimate would have to be obtained experimentally.

2:7 DISCUSSION AND CONCLUSIONS

To meet the major milk quality standard of less than 100,000 bacteria colonies/ml, water heating and milk cooling have become an integral part of dairy shed operations.

Water temperatures ranging from $50^{\circ}C$ to $90^{\circ}C$ are required for the majority of approved plant cleaning systems while, under New Zealand conditions, water volumes of approximately 1.7 litres/cow.day are required, depending on plant size. Electricity requirements for this operation have been estimated to be, on average, between 40%-55% of the total farm requirements.

Milk cooling, to meet Government regulations, is most efficiently achieved by removing 50% of the 109 (121) $MJ.m^{-3}$ heat load through an in-line plate heat exchanger with the remainder being removed by a direct expansion refrigeration system at a rate of 14 (15) $MJ.m^{-3}.h^{-1}$. The electricity requirement is reported to be in the range of 15%-25% of the total farm requirements.

Two basic heat recovery systems have been used with the desuperheater systems achieving efficiencies in the range of 24% to 35% and 50% to 80% for complete condensing systems. The overall effect of heat recovery has been to reduce water heating electricity requirements by 24% to 85%.

To offset this saving, heat recovery systems have increased refrigeration power consumption by between 20% and 31%. In general, heat recovery has been achieved on dairy farms by adding a heat exchanger to an existing air cooled condenser system. There is only one reported instance of a water cooled condenser being used for a complete condensing system. The use of a water cooled condenser as a means of preheating water for a desuperheater has not been reported.

Three heat exchanger designs have been investigated with counter flow, as opposed to coil-in-tank, systems being the most efficient in terms of overall thermal conductances, with values of 175 to 950 $W.m^{-2}.^{o}C^{-1}$ for a tube-in-tube counter flow 'once-through' design and 66 to 148 $W.m^{-2}.^{o}C^{-1}$ for coil-in-tank designs. Recirculating systems gave results between those of tube-in-tube and coil-in-tank. Water temperatures of 38^{o} - $65^{o}C$ have been recorded, with one coil-in-tank system becoming uneconomical over $58^{o}C$. The use of extended surface area on the refrigerant side, for heat recovery heat exchangers, has not been reported.

The level of heat recovery and the temperature of the hot water produced was reported to be affected by condenser pressures ranging from 630 kPa to 1520 kPa. Some form of condenser pressure control required the use of regulatory valves or the dumping of heat through air cooled heat exchangers or as hot water.

Variations in heat recovery heat exchanger area, refrigeration capacity, and the relationship between milk and water volumes have made it difficult to conclude which overall system is best. It is apparent that the selection of any heat recovery system will be largely dependent upon the relationship between the volume of water to be heated and the volume of milk cooled, with high volumes of hot water being provided by complete condensing systems and low volumes by desuperheating systems.

CHAPTER 3

EXPERIMENTAL DESIGN

A study of the areas requiring further investigation, discussed in Section 2:7, suggested that three experiments should be designed.

1) To develop a heat recovery exchanger, incorporating secondary refrigerant area.

2) To evaluate the effect of different operating conditions on system performance and heat recovery.

3) To predict the performance of heat recovery systems under field conditions.

3:1 EXPERIMENTAL BASIS

The experiments were modelled around a 210 cow herd. A 2.25 kW direct expansion Freon 12 refrigeration system was used to cool a 2250 litre uninsulated vat, as this enabled a comparison with actual operating conditions at a nearby town supply farm (Massey University No.1 Dairy Unit).

A cooling load of 2000 $l.day^{-1}$ was selected with a 40/60 split between evening and morning milkings. An average flow into the vat of 12 $l.min^{-1}$ at 23^{o}C was measured at the dairy and was used in the refrigeration trials.

To comply with the regulations (Section 2:2.2), 300 $l.day^{-1}$ of hot water at temperatures up to 95^{o}C was required.

For the milk and hot water volumes selected, theoretical calculations (Appendix A3) showed that a complete condensing 'once-through' system was not a viable alternative. This was supported by Fleming and O'Keeffe (1977). Therefore, a desuperheater system was selected.

3:2 EXPERIMENT I

A tube-in-tube heat exchanger was selected as the basic design for the desuperheater unit. To improve heat transfer, a spiral finned tube was used with the refrigerant flowing over the fins. Increased turbulence in the water tube was achieved with cores.

To establish performance characteristics of the selected design, U values were determined for:-

1) Refrigerant flows of 2.0, 1.5, 1.2 and 0.7 $l.min^{-1}$
   (The lowest flow rate was chosen to evaluate the effect of parallel flow.)

2) A water flow rate of 0.63 $l.min^{-1}$ to obtain 300 litres in 8 hours of cooling
   (To estimate the performance under turbulent and laminar flow conditions, flowr of 1.4, 0.63, 0.4 and 0.31 $l.min^{-1}$ were selected. The 0.31 $l.min^{-1}$ flow was half the required flow of 0.63 $l.min^{-1}$. It was chosen to match the refrigerant parallel flow configuration.)

3) Water inlet temperatures of 15^{o}C and 30^{o}C
   (These temperatures were selected as the 30^{o}C could be achieved in a water cooled condenser, whilst 15^{o}C corresponded to tap water.)

The various combinations of these variables used to obtain the design information are presented in Table 3:1.

3:3 EXPERIMENT II

3:3.1 Condenser System

The condenser system consisted of a primary heat exchanger for heat recovery (in which some condensation could occur) and a secondary heat exchanger for condensation. The secondary heat exchanger was either air or water cooled.

3:3.2 Condenser Pressures

Four condenser pressures were selected to evaluate the effect of condenser pressure on system performance. The maximum was

TABLE 3:1

Sequence of settings for Experiment I

Run	Regrigerant Flow ($1\ min^{-1}$)	Water Temperature (^{o}C)	Water Flow ($1\ min^{-1}$)	Cores
1	2.0	30	1.4	X
2	2.0	30	0.4	X
3	2.0	15	1.4	X
4	2.0	15	0.4	X
5	1.5	30	1.4	X
6	1.5	30	0.4	X
7	1.5	15	1.4	X
8	1.5	15	0.4	X
9	1.2	30	1.4	X
10	1.2	30	0.4	X
11	1.2	15	1.4	X
12	1.2	15	0.4	X
13	2.0	30	0.4	0
14	2.0	30	0.6	0
15	2.0	30	1.4	0
16	2.0	15	0.4	0
17	2.0	15	0.6	0
18	2.0	15	1.4	0
19	0.7	30	0.3	0
20	0.7	15	0.3	0
21	1.9	30	0.3	0
22	1.9	15	0.3	0

(X denotes without cores)

(0 denotes with cores)

set at 1200 kPa, since higher pressures were expected to be detrimental to the compressor's service life and require excessive increases in compressor power consumption. To avoid the risk of the refrigerant boiling in the liquid line, the minimum condenser pressure setting used was 650 kPa to accommodate the headloss through the secondary heat exchanger, receiver and the evaporator liquid line. Intermediate values of 750 kPa and 1000 kPa were selected to complete the range.

3:3.3 Milk Temperatures

Based on field requirements and the literature (Section 2:1.2.2), milk inlet temperatures of 18°C and 23°C, and final temperatures of 7°C and 4°C, were selected.

3:3.4 Suction Superheating

The effect on system performance of increasing suction superheat, by using discharge water from the secondary water cooled heat exchanger, was investigated. (This technique has received little attention in other work on heat recovery on dairy farms.)

3:3.5 Primary Heat Exchanger

The effect of the primary heat exchanger on system efficiency, in particular compressor power consumption, was investigated for the air cooled condenser system by testing the unit with the primary heat exchanger removed. The water cooled condenser system was not tested in this way, as the primary and secondary heat exchangers were considered to be an integral unit, i.e., the secondary always preheated the water for the primary.

3:3.6 Receiver Pressure

One run was conducted with a receiver pressure of 700 kPa to evaluate the effect of changing receiver pressure. For other runs the receiver pressure was 600 kPa.

To obtain the required data, 18 runs were conducted as listed in Table 3:2.

3:4 EXPERIMENT III

Concurrent with Experiments I and II, a monitoring programme was conducted on a nearby town supply farm (Massey University No.1 dairy farm). The objective of this experiment was to collect data on

1) Milk and water volumes and temperatures

2) Refrigeration, water heating and total power consumption

3) Refrigeration and water heating operating times

such that the laboratory results could be related to the field situation.

From relationships established, the effect of the final heat recovery system (developed in Experiment II) on dairy shed energy requirements, under field conditions, was estimated. This analysis enabled the impact of refrigeration heat recovery, on the New Zealand Dairy Industry, to be predicted.

TABLE 3:2

Summary of test runs for Experiment II*

Run No.	Condenser System	Condenser Pressure (kPa)	Milk Inlet Temperature (°C)	Milk Final Temperature (°C)	Primary
W12	Water	1200	23	4	Yes
W10	Water	1000	23	4	Yes
W10/18	Water	1000	18	4	Yes
W7	Water	750	23	4	Yes
W7/7	Water	750	23	7	Yes
W7/7/18	Water	750	18	7	Yes
W7/18	Water	750	18	4	Yes
W6	Water	650	23	4	Yes
A12	Air	1200	23	4	Yes
A12/NP	Air	1200	23	4	No
A10	Air	1000	23	4	Yes
A10/NP	Air	1000	23	4	No
A10/18	Air	1000	18	4	Yes
A7	Air	750	23	4	Yes
A7/NP	Air	750	23	4	No
A7/7B**	Air	750	23	4	Yes
A7/18	Air	750	18	4	Yes
A6	Air	650	23	4	Yes

* Note that the runs were not necessarily conducted in the sequence shown

** Receiver pressure of 700 kPa (7 Bar)

CHAPTER 4

EXPERIMENTAL EQUIPMENT, METHODS AND DATA ANALYSIS

4:1 LABORATORY PLANT

4:1.1 Mechanical Equipment

4:1.1.1 Refrigeration System

A schematic diagram of the refrigeration system used is presented in Figure 4:1. This may be compared with the photographs in Figures 4:2 to 4:5. A wiring diagram and arrangement drawing are presented in Appendix A4:1.

From the 2.25 kW direct drive sealed compressor unit (Figure 4:1 and 4:5), refrigerant passed through the primary heat exchanger, one of the secondary heat exchangers and the refrigerant flow meter to the receiver. A solenoid shut-off valve, connected to the vat thermostat, allowed the refrigerant to pass to the 7 kW capacity evaporator via a second flow meter, liquid/vapour heat exchanger, drier and thermostatically controlled expansion valve (Figure 4:5). The vapour from the evaporator then passed through a suction superheater before entering the compressor.

The system was designed to allow flexibility in the routing of the refrigerant through the condenser circuit such that varying proportions of refrigerant flow could by-pass the test primary heat exchanger and refrigerant flow meter for Experiment I. and allow the complete isolation of the primary heat exchanger in Experiment II.

Condenser pressure for the air cooled secondary heat exchanger was controlled by a 'Danfoss CPR' valve (Figure 4:3). The CPR valave restricted the flow of refrigerant from the heat exchanger until the heat exchange area had been sufficiently reduced to cause the pressure to rise to the preset level required to open the valve against the adjustable spring.

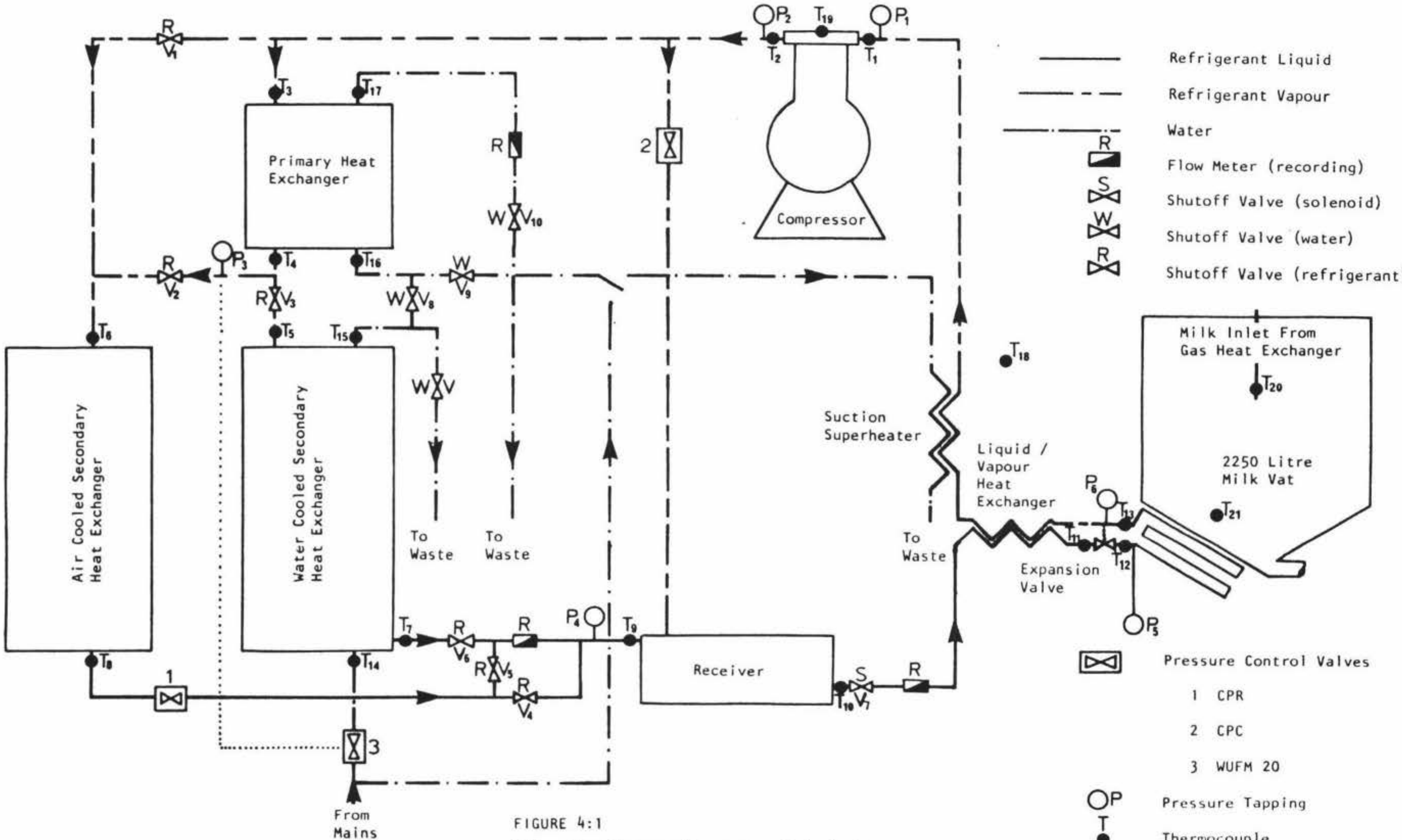


FIGURE 4:1

Schematic diagram of experimental plant

FIGURE 4:2
Plate showing A) experimental plant and
B) instrumentation

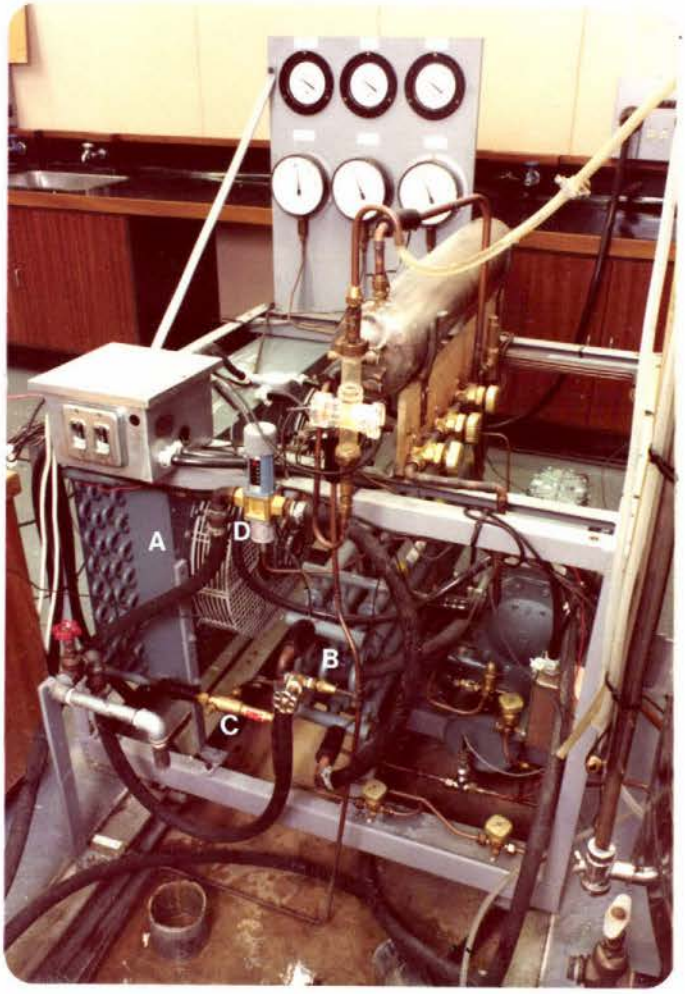


FIGURE 4:3

Plate showing A) air and B) water cooled condensers and C) refrigerant control valve CPR and D) WUFM 20

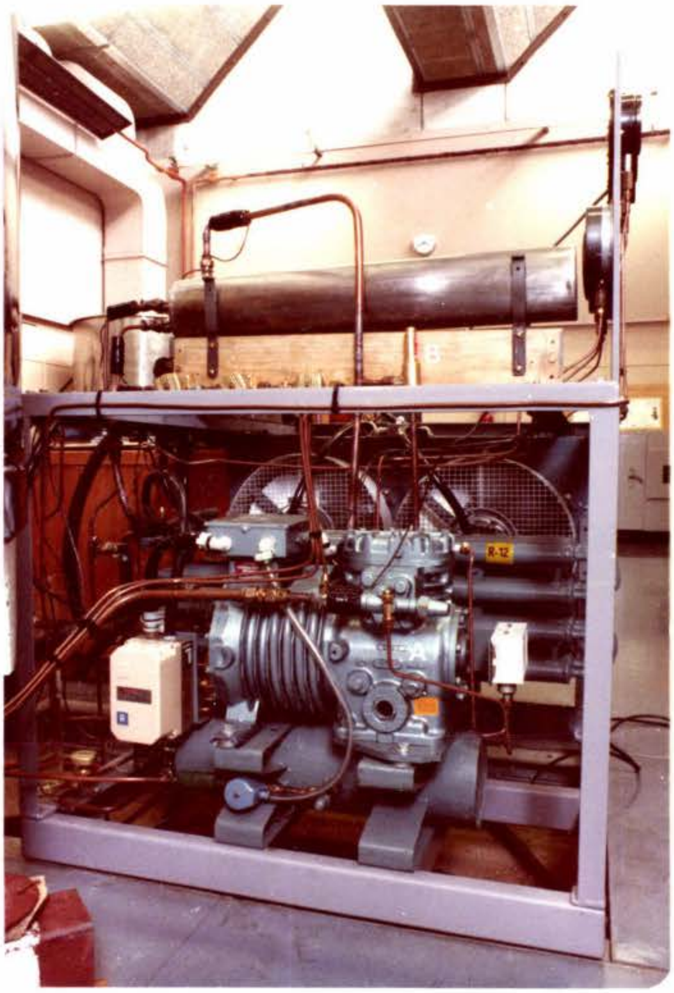


FIGURE 4:4

Plate showing A) compressor and B) refrigerant control valve (CPC)

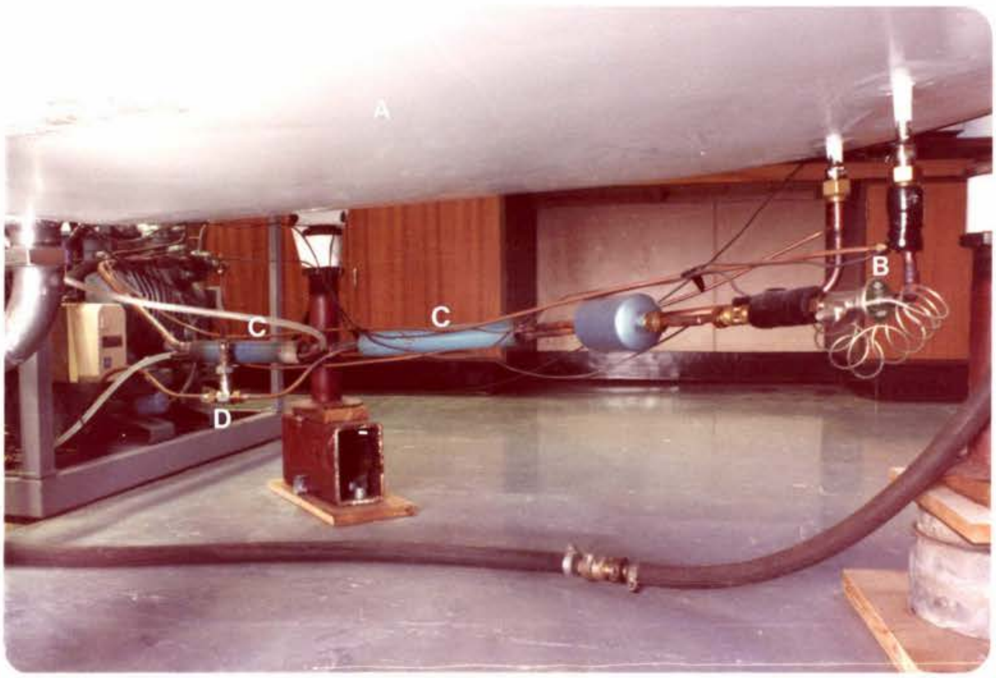


FIGURE 4:5

Plate showing A) evaporator, B) expansion valve, C) heat exchangers and D) evaporator refrigerant flow meter.

Condenser pressure control, for the water cooled secondary heat exchanger, was by a 'Danfoss WUFM20' valve (Figure 4:3). This valve varied the water flow rate through the secondary heat exchanger such that the vapour pressure acting against a diaphragm was in equilibrium with the pressure exerted by an adjustable spring.

Receiver pressure was maintained by a 'Danfoss CPC' valve (Figure 4:3) in the by-pass line (Figure 4:1) in the case of the air cooled condenser system, and by valve V6 for the water cooled system. The CPC valve operated in a similar manner to the CPR valve by varying the refrigerant vapour flow so as to maintain a constant downstream (Receiver) pressure.

4:1.1.2 Heat Recovery Exchangers

Stainless steel and copper finned tubing were selected as the basis for the heat exchanger design as they were readily available, corrosion resistant and complied with the requirements of Section 3:2.

The design claculations (Appendix A4:2) resulted in the heat exchanger consisting of:-

1) 9 mm OD stainless steel copper finned tube (158 fins per meter)

2) 4.7 mm OD stainless steel core

3) 31.75 mm OD stainless steel outer tube

4) 20 mm 'armaflex' foam tube insulation.

This arrangement gave a primary water heat transfer area of 0.0216 m^2, a secondary fin area of 0.264 m^2 and a refrigerant heat exchange area of 0.28 m^2 (fins plus bare tube).

The design of the full size primary heat exchanger for Experiment II is discussed in Section 5:5.

4:1.2 Instrumentation

Monitoring points are shown in Figure 4:1.

Pressures were measured at six points with Bourdon gauges (Figure 4:3); three for the condenser circuit and the other three in the evaporator circuit. Temperatures were read at 18 points with copper constantan thermocouples, using a digital multimeter and an ice/water cold junction (Figure 4:2).

Total power consumption for the compressor and the fans was measured with single phase kWh meters, while instantaneous power was measured with a 'Cambridge' load analyser. In addition, the voltage current per phase, and power factor (P.F.), were measured, from which the instantaneous compressor power consumption could also be calculated.

Two 6.25 mm turbine flow meters were used to measure the refrigerant flow entering the receiver and the evaporator (Figures 4:3 and 4:5). Primary water flow was measured with a low head turbine flow meter, developed in the Agricultural Engineering Department, Massey University (Studman and Compton, 1982, Figure 4:3). Pulses from these three meters were counted by 'Racal' counters. Secondary water flow rate was measured by collecting a weighed volume of water over 30 seconds. Water flow rates through the suction superheater were measured with a measuring cylinder and stop watch.

Calibration data and error calculations are presented in Appendix A4:3.

4:2 FIELD PLANT

A schematic view of the field plant is presented in Figure 4:6 and a wiring diagram is given in Appendix A4:1.2.

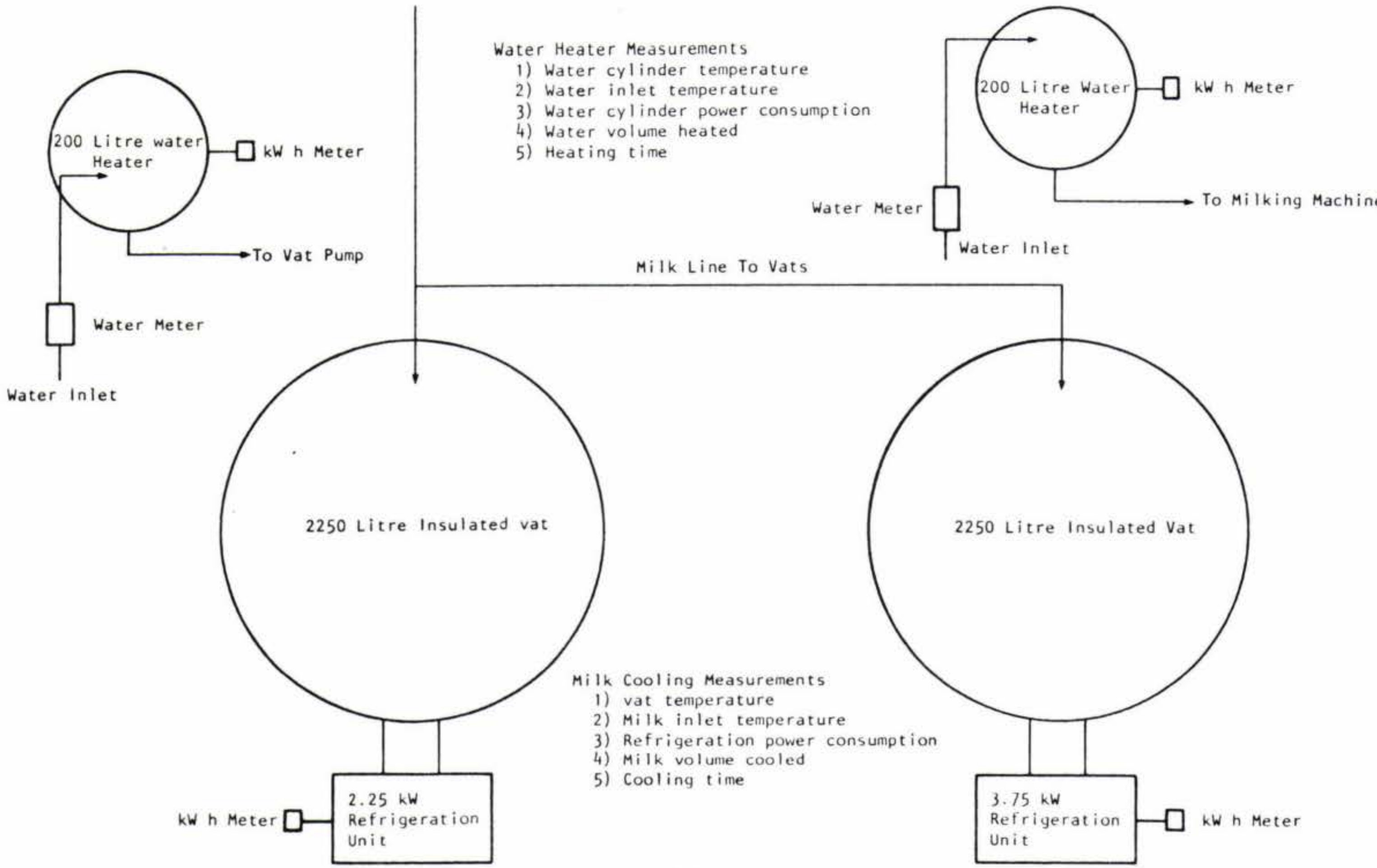


FIGURE 4:6

Field plant measurements

4:2.1 Instrumentation

Temperatures at six points were measured with copper constantan thermocouples and recorded on a multichannel chart recorder. Calibration was by ice and boiling water. Water volumes for the two water heating cylinders were measured with 'Kent' water meters.

Power consumption for milk cooling, water heating and total farm power was measured with kWh meters. Cooling and heating times were determined from data recorded on a multichannel event recorder.

4:3 EXPERIMENTAL METHODS

4:3.1 Experiment I

For each of the twenty two tests in Table 3:1, the following procedure was adopted.

1) A constant refrigerant flow in the system was obtained by establishing a constant water flow rate over the evaporator, i.e., constant refrigerant loading.

2) A constant condensing pressure of 800 kPA was set up in each of the two secondary heat exchanger circuits.

3) Water flow rates were adjusted using valve V8 in Figure 4:1 (with valve V9 closed) for 30°C water, or valve V9 (with valve V8 closed) for 15°C water.

4) Refrigerant flows were controlled with valve V1, while valves V3, V4 and V6 were open, and valves V2 and V5 were closed.

5) After the system stabilised (approximately 60 minutes after start up and 10 minutes between settings) readings of temperature, pressure and flow were taken.

Although each run in Table 3:1 was independent, the experiment was not conducted on a random basis due to operating difficulties and the time required for stabilisation. Therefore, once the refrigerant flow was adjusted to the required level, all runs at that setting were carried out. The procedure was than repeated for a new refrigerant flow.

4:3.2 Experiment II

The test runs conducted in this experiment are presented in Table 3:2. The following procedure was adopted for each run.

1) The selection of system conditions (Table 3:2), (system loading, condenser pressure and primary water flow rates) were made by adjusting the required control valves and hand shut-off valves (V1 to V9). Milk loading temperature was achieved by adjusting the gas flow at the heat exchanger (Figure 4:1).

2) The refrigeration system was started at the same time as the milk flow (system loading) commenced. System loading was 800 litres followed by 1200 litres (simulating evening and morning milkings).

3) Final adjustment of condensing pressure and primary water flow rate was made within 20 minutes of initial start-up.

4) Readings at all data collection points were made at 30 minute intervals until the final temperature was reached, at which point only totals of power consumption, time and refrigerant volume were recorded.

An additional run was conducted under the same conditions as Run W10 (Table 3:2) but with the suction superheater operating. Modifications to valve V9, and the throttling of the primary heat exchanger flow (valve V10), were required to force the water through the suction superheater.

4:3.3 Experiment III

Data on a continuous basis was collected by the chart recorders for the temperatures and events specified in Section 3:3. Each chart was marked with the date and time daily.

Power meters were read daily between milk pick up and water cylinder filling (8.00 a.m. - 8.30 a.m.). This time was selected for convenience and because it was the only period in the day when both the refrigeration and water heating systems were off.

Milk and water volumes were measured after each milking and cylinder filling respectively.

Significant events, such as cylinder over filling, were noted.

The recording period was over 50 days from 1 November 1980 to 22 December 1980.

This experiment was in two parts, the first of which is presented in this thesis, while the second, monitoring of heat recovery systems under field conditions, is continuing.

4:4 DATA ANALYSIS

4:4.1 Experiment I

The performance of the test primary heat exchanger was measured in terms of the overall thermal conductance (U) as expressed by the equation;

$$U\ (kW.m^{-2}.^{o}C^{-1}) = \frac{Q}{\Delta t_m\ A} \qquad \text{..... Eqn 4:1}$$

where Q = Heat flow into the water (kW)

$= M_w \times Sp\ Ht \times \Delta t$ ..... Eqn 4:2

= Mass flow of water X specific heat X temperature rise

Δt_m = Log mean temperature difference (^{o}C)

A = Area of the refrigerant side of the primary heat exchanger (m^2)

= 10.72 X Area of the water tube

A comparison of the U values for the conditions tested (Table 3:1), in conjunction with the headlosses across the heat exchangers, were used for the design of the final primary heat exchanger.

4:4.2 Experiment II

A computer was used to process the large volume of data. The list of variables processed, together with computer programs, sample data sheets and calculations are presented in Appendix A4:4.

Analysis of the refrigeration system and the primary heat exchanger performance required the determination of certain key variables from the raw data. These variables were;

1) The COP (refrigeration)
2) The COP (heat recovery)
3) The refrigerant mass flow
4) The refrigeration effect (R.E.)
5) The primary heat exchanger heat flow
6) The instantaneous compressor power consumption.

Calculations of 1) to 5) above, required, at some stage, the determination of either refrigerant specific enthalpies, specific heats and/or liquid density. Since the large number of data points made the use of refrigerant tables impracticable, equations in terms of pressure (bars abs.) and temperature (^{o}K) were used. Sources of these equations and error calculations are listed in Appendix A4:5.

4:4.2.1 Equations of Refrigerant Properties

The following equations were used:-

1) Saturation pressure (P_{SAT})

$$P_{SAT} = \left(\frac{T_{SAT} - 157.564}{85.69}\right)^{3.8226} \qquad .....\text{Eqn } 4:3$$

where T_{SAT} = Saturation temperature (^{o}K)

2) Specific heat of the superheated vapour (C_{VAP})

$$C_{VAP} = 0.61244 \times 10^{(0.008849 \times P_{SAT})} \qquad .....\text{Eqn } 4:4$$

3) Specific heat of the liquid (C_f)

$$C_f = 0.819566 - 0.894247 \times 10^{-3}\, T_{SAT} + 4.9062 \times 10^{-6}\, T_{SAT}^{\,2} \qquad .....\text{Eqn } 4:5$$

4) Specific volume of the liquid (V_f)

$$V_f = {}^1/\rho_f = {}^1/(558.025 + 0.777K + 17.943K^{0.5} + 117.4362K^{0.333} - 3.40204 \times 10^{-4}K^2) \quad \text{.....Eqn 4:6}$$

where K = Triple point temperature - T_{SAT}

$= 388.3^{o}K - T_{SAT}$

5) Liquid enthalpy (h_f)

$$h_f = (-187.36287 + 0.819566\ T_{SAT} + 0.4471 \times 10^{-3}\ T_{SAT}{}^2 + 2.4531 \times 10^{-6}\ T_{SAT}{}^3) + \frac{(V_f T_{SAT} + 6.556 \times 10^{-4})}{2} \times (P_{SAT} - 0.6417) \quad \text{.....Eqn 4:7}$$

6) Heat of vaporization (h_{fg})

$$h_{fg} = 165.063 \times 10^{(-0.011 \times P_{SAT})} \quad \text{.....Eqn 4:8}$$

7) Superheat enthalpy (h_{SH})

$$h_{SH} = h_f + h_{fg} + (T_{SH} - T_{SAT})\ C_{VAP} \quad \text{.....Eqn 4:9}$$

4:4.2.2 Calculation of System Variables

Equations for basic system variables are summarised below together with derived error calculations in Appendix A4:5.2.

1) Refrigeration Effect (R.E.)

$$RE = h_1 - h_{3b} \quad (kJ.kg^{-1}) \qquad \text{(Ref. Eqn 2:5)}$$

2) Refrigerant Mass Flow (M_r)

$$M_r = \frac{\text{Refrigerant flow } (l.min^{-1})}{60000 \times V_f} \quad (kg.s^{-1})$$

.....Eqn 4:10

3) COP refrigeration (COP_r)

$$COP_r = \frac{RE \times M_r}{\text{Instantaneous Compressor Power}}$$

.....Eqn 4:11

4) Heat Recovery Rate (Q)

$$Q = M_w \times Sp\ Ht_w \times \Delta t \quad (kW)$$ .....Eqn 4:2

where M_w = the primary water flow ($kg.s^{-1}$)

$Sp\ Ht_w$ = the specific heat of the water

Δt = the temperature rise

5) COP heat recovery (COP_h)

$$COP_h = \frac{Q}{\text{Instantaneous Power Consumption}}$$

.....Eqn 4:12

The results of these analyses were used to select the final settings of the heat recovery system which would achieve satisfactory cooling while recovering the greatest quantity of heat as hot water. This selection was made using the equation for Ke in Section 2:5.3.3 (Eqn 2:7).

4:4.3 Experiment III

4:4.3.1 Water Heating

Total water heating power consumption figures were allocated to heating and standings loads according to data from the event recorder. An estimate of theoretical power consumption for the measured water volumes was also made.

From this data, regression relationships for these variables were determined to give predictions of water heating power requirements and the effect of heat recovery.

4:4.3.2 Refrigeration

Raw data was processed in the same manner as for water heating. However, instead of estimating theoretical power consumption, an estimate of the COP of refrigeration was made, where:-

$$COP = \frac{Vol_{evening} \text{ X Sp Ht X } 19^{o}C + Vol_{morning} \text{ X Sp Ht X } \Delta t}{\text{Total Power X 3600}}$$

.....Eqn 4:13

where Sp Ht = Specific heat ($kJ.kg^{-1}.^{o}C^{-1}$)

Δt = Mix temperature - Pick up temperature

$$\text{Mix temperature} = \frac{\text{Vol}_{\text{evening}} \times 4^{\circ}\text{C} + \text{Vol}_{\text{morning}} \times 23^{\circ}\text{C}}{\text{Total Volume}}$$

.....Eqn 4:14

Regression relationships between milk volume cooled and power consumption were determined to predict the impact of heat recovery.

4:4.3.3 Total Power Consumption

Refrigeration and water heating power, as percentages of total power consumption, were calculated in order to compare them with the literature .

CHAPTER 5

EXPERIMENT I RESULTS AND DISCUSSION

5:1 TEST PRIMARY HEAT EXCHANGER WITHOUT CORES

Calculated results of overall thermal conductance (U) and Reynolds Number (Re) for water, at the refrigerant flow rates tested (Table 3:1), are presented in Table 5:1 and Figure 5:1.

The results show that there were no significant changes in U for varying refrigerant flow rates over the range tested.

Increasing water flow rates (M_w) increased U values significantly. This was expected since the low flow produced laminar conditions (Re < 2100) while the two high flows (1.30 and 1.40 $l.min^{-1}$) produced turbulent conditions (Re > 4000).

Increasing water inlet temperatures had no significant effect on U for the low water flow rates, as the flow regime remained laminar for both inlet conditions, i.e., Re < 2100. For the high water flow rates, the effect of inlet temperature could not be determined as the two water flow rates differed. The effect of these two flow rates, in conjunction with the temperature effect on water viscosity, was to increase U by approximately 0.3, i.e., a small significant difference. Results from Section 5:2 indicate that the effect of water inlet temperature was small and therefore a large proportion of the difference in U values can be attributed to the differences in water flow rate.

5:2 TEST PRIMARY HEAT EXCHANGER WITH CORES

Results of U values for the four water flows and two inlet temperatures tested are presented in Table 5:2 and Figure 5:2. (Corresponding values from Table 5:1 are also presented in Figure 5:2 for comparison.) These tests were conducted at the high refrigerant flow on the assumption that the magnitude of the effect would be the same for the other refrigerant flow rates.

There was a significant increase in U due to the action of the cores. This was particularly true for the low flow rates as the flow regime moved from laminar to turbulent flow on the insertion of the cores.

TABLE 5:1

Overall thermal conductance ($kW.m^{-2}.^{o}C^{-1}$) values for the test primary heat exchanger without cores

Refrigerant Flow ($l.min^{-1}$)	Low Water Flow Inlet Temperature 15°C			Low Water Flow Inlet Temperature 30°C			High Water Flow Inlet Temperature 15°C			High Water Flow Inlet Temperature 30°C		
	U	Re	M_w ($l.min^{-1}$)	U	Re	M_w ($l.min^{-1}$)	U	Re	M_w ($l.min^{-1}$)	U	Re	M_w ($l.min^{-1}$)
2.0	0.098 $\pm$0.008	1777	0.44	0.096 $\pm$0.008	2029	0.39	0.139 $\pm$0.011	4689	1.31	0.169 $\pm$0.013	6417	1.38
1.5	0.098 $\pm$0.008	1715	0.39	0.099 $\pm$0.008	2033	0.40	0.137 $\pm$0.011	4590	1.30	0.170 $\pm$0.013	6098	1.40
1.2	0.098 $\pm$0.008	1740	0.43	0.096 $\pm$0.008	1877	0.39	0.118 $\pm$0.009	4423	1.30	0.157 $\pm$0.013	5885	1.43

Note: U is based on the refrigerant surface area of 0.28 m^2

U for water side = 10.7 X U for the refrigerant side

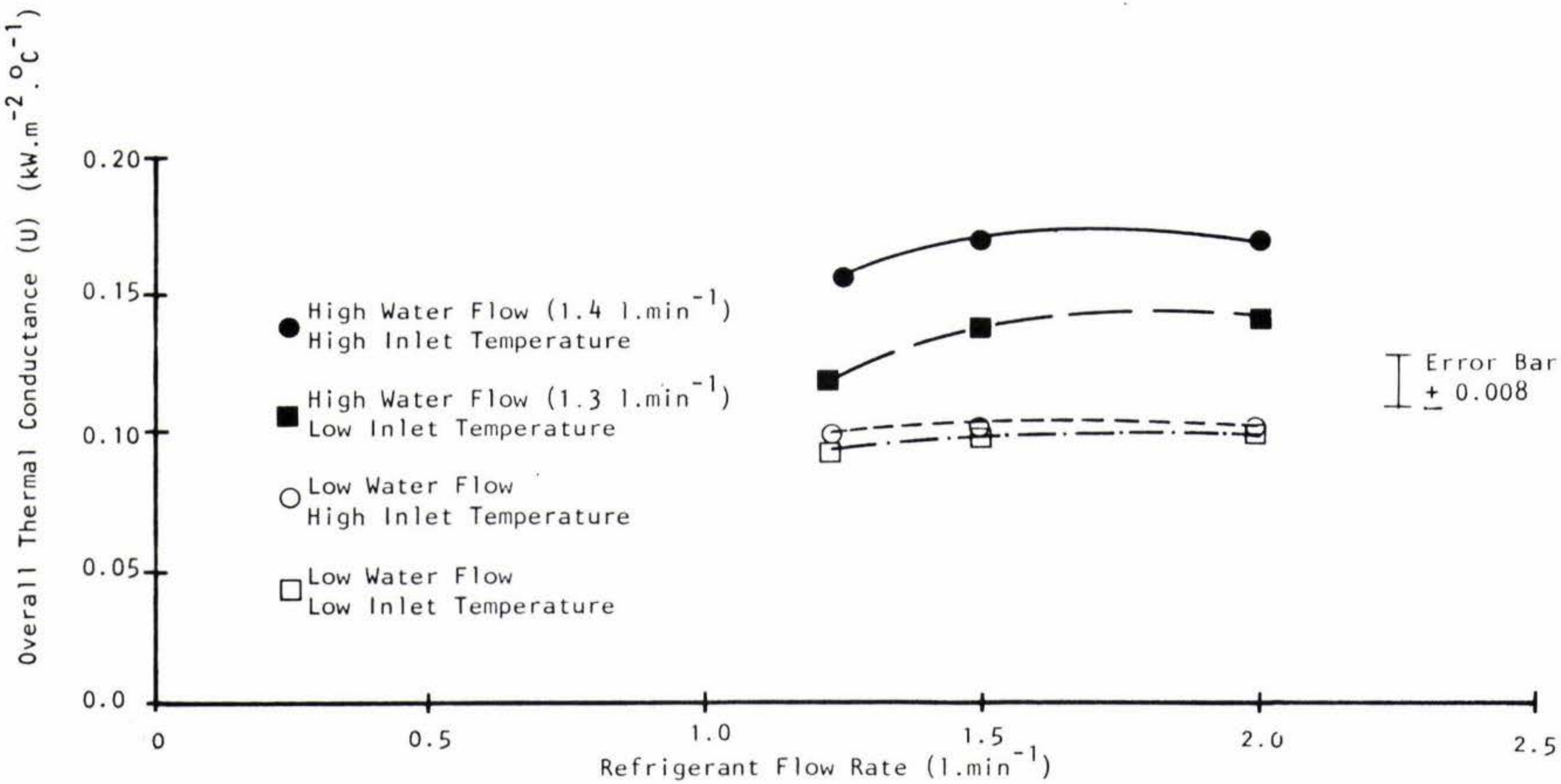


FIGURE 5:1

The effect of refrigerant and water flow rates on overall thermal conductance of the test primary heat exchanger

TABLE 5:2

Overall thermal conductance values (kW. $m^{-2}.^{o}C^{-1}$) for the test primary heat exchanger with cores at a refrigerant flow of 2.0 $l.min^{-1}$

Water Flow Rates ($l.min^{-1}$)	Water Inlet Temperature 15^{o}C			Water Inlet Temperature 30^{o}C		
	U	Re	M_w ($l.min^{-1}$)	U	Re	M_w ($l.min^{-1}$)
0.40	0.145 $\pm$0.012	1269	0.41	0.157 $\pm$0.013	1466	0.39
0.60	0.134 $\pm$0.011	1554	0.60	0.160 $\pm$0.013	1933	0.61
1.40	0.179 $\pm$0.014	2995	1.40	0.189 $\pm$0.015	3596	1.36

Note: 1) The critical region for annuli was suggested by Prengles and Rothfus (1955) to have a range of 700 to 2200.

2) U for water side = 10.7 X U for refrigerant side.

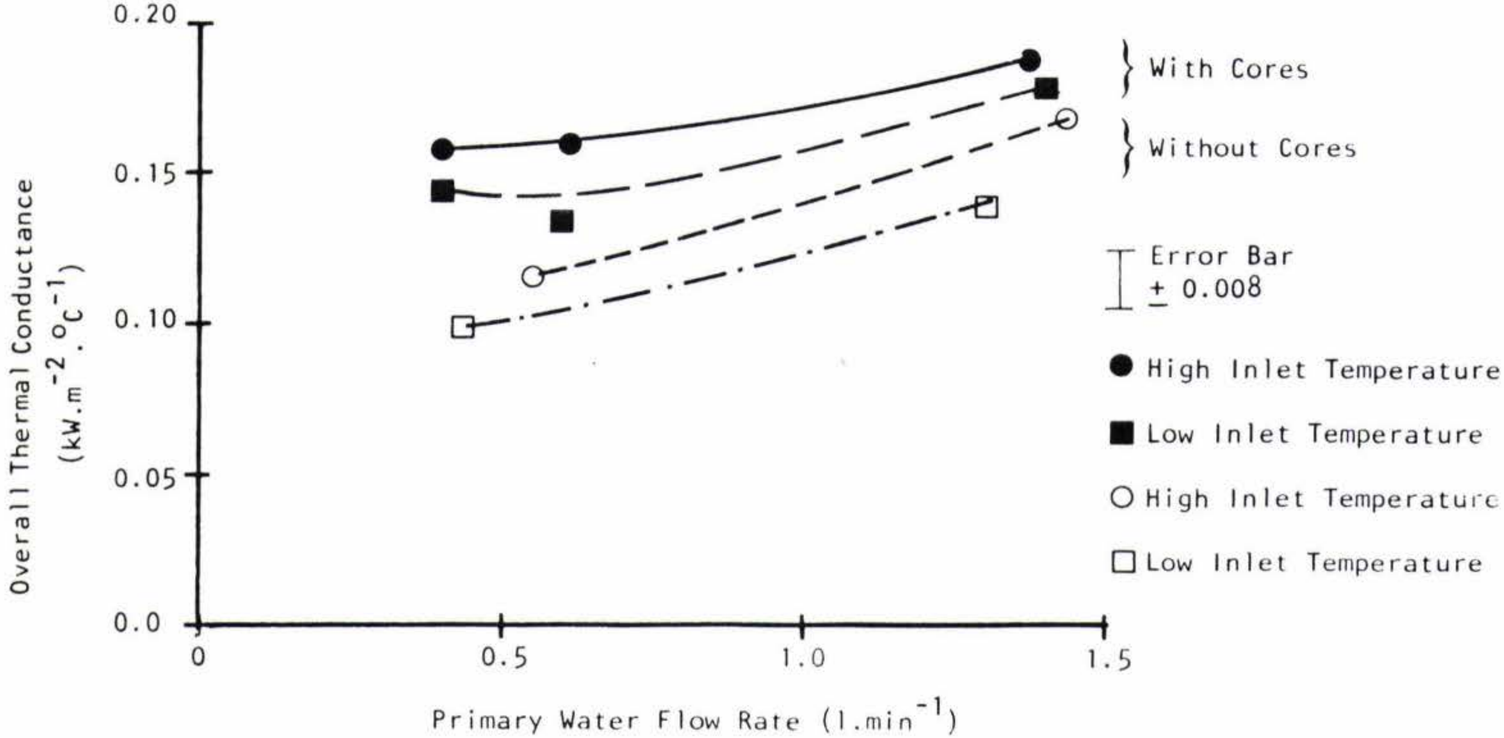


FIGURE 5:2

The effect of cores on the overall thermal conductance of the test primary heat exchanger at high refrigerant flow rates

The effect of increasing water flow rates resulted in only a small increase in U, probably as a result of the flow being turbulent over the whole range and the refrigerant film coefficient being the limiting factor.

The effect of increasing water inlet temperature at a fixed refrigerant flow was to slightly increase U. The effect was due to the reduction in viscosities as temperature increased and is expressed by the differences in Re.

5:3 TEST PRIMARY HEAT EXCHANGER UNDER SIMULATED PARALLEL FLOW ARRANGEMENT

Results for U and Re for a water flow of 0.31 $l.min^{-1}$ and refrigerant flows of 0.70 $l.min^{-1}$ and 1.86 $l.min^{-1}$ are presented in Table 5:3 and Figure 5:3 with curves from Figure 5:1 for comparison. Only two refrigerant flows were tested as it was assumed that the curves would have the same shape as those in Figure 5:1. The high flow of 1.86 $l.min^{-1}$ was arbitrarily selected to pin the maximum end point of the curve, not to simulate a total flow of 3.72 $l.min^{-1}$.

The results show that doubling the inlet temperature produced a small, but not significant increase in U. The U values for a flow of 0.31 $l.min^{-1}$ with cores is significantly higher than for flows of 0.4 $l.min^{-1}$ without cores at the higher inlet temperatures. The trend was not as marked at the low inlet temperatures.

The variation in U due to changes in refrigerant flow was greater than those in Table 5:1, due to the greater range tested.

5:4 TEST PRIMARY HEAT EXCHANGER HEADLOSS

Headloss data for various refrigerant flows is presented in Figure 5:4.

TABLE 5:3

Overall thermal conductance values ($kW.m^{-2}.^{o}C^{-1}$) for the test primary heat exchanger with cores

Refrigerant Flow ($l.min^{-1}$)	Water Temperature 15°C U	15°C Re	15°C M_w ($l.min^{-1}$)	30°C U	30°C Re	30°C M_w ($l.min^{-1}$)
1.86	0.114 ±0.009	921	0.295	0.130 ±0.010	1087	0.326
0.70	0.096 ±0.007	784	0.315	0.110 ±0.008	897	0.321

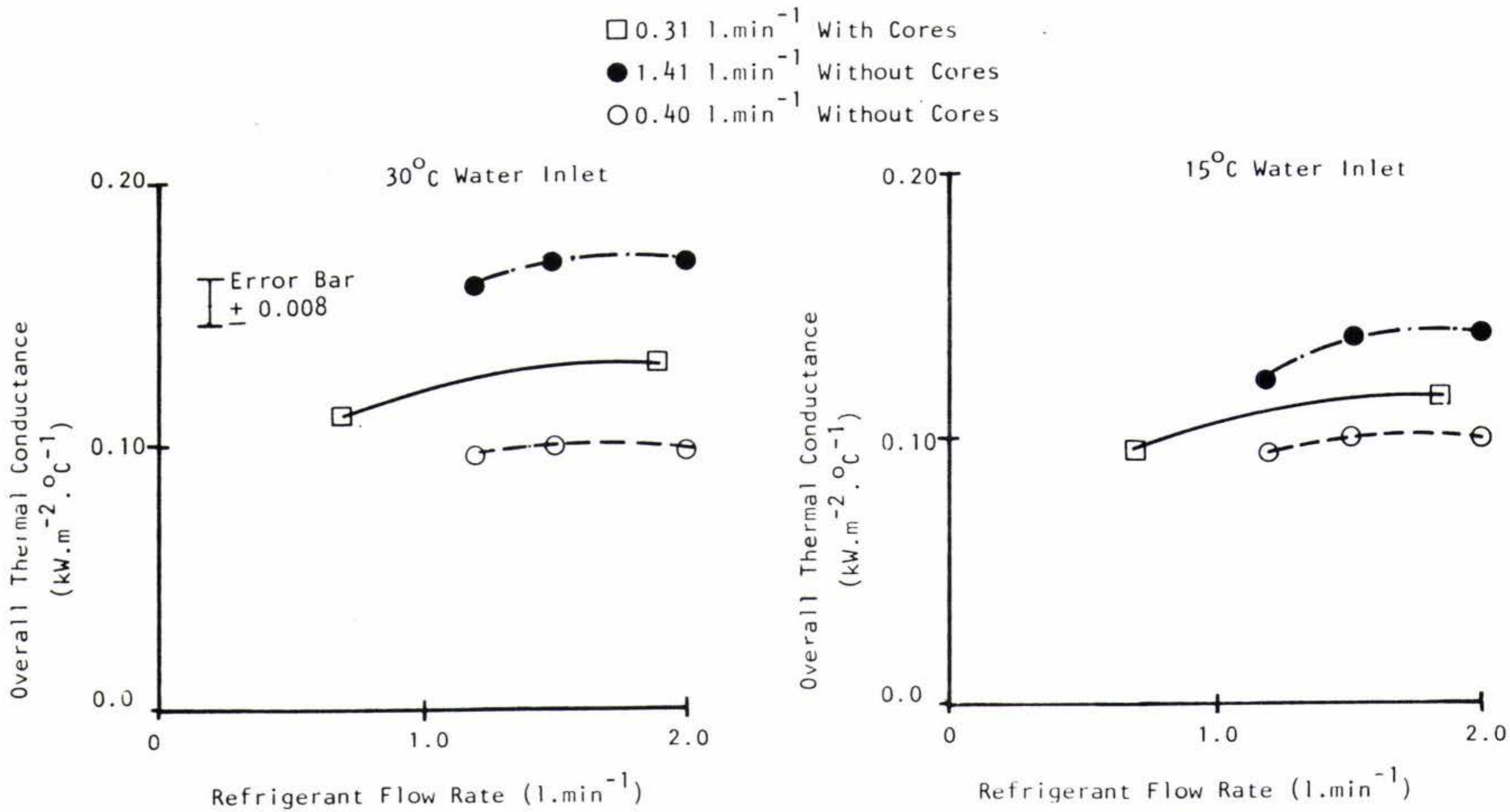


FIGURE 5:3

The effect of low refrigerant and water flow rates on overall thermal conductance of the test primary heat exchanger

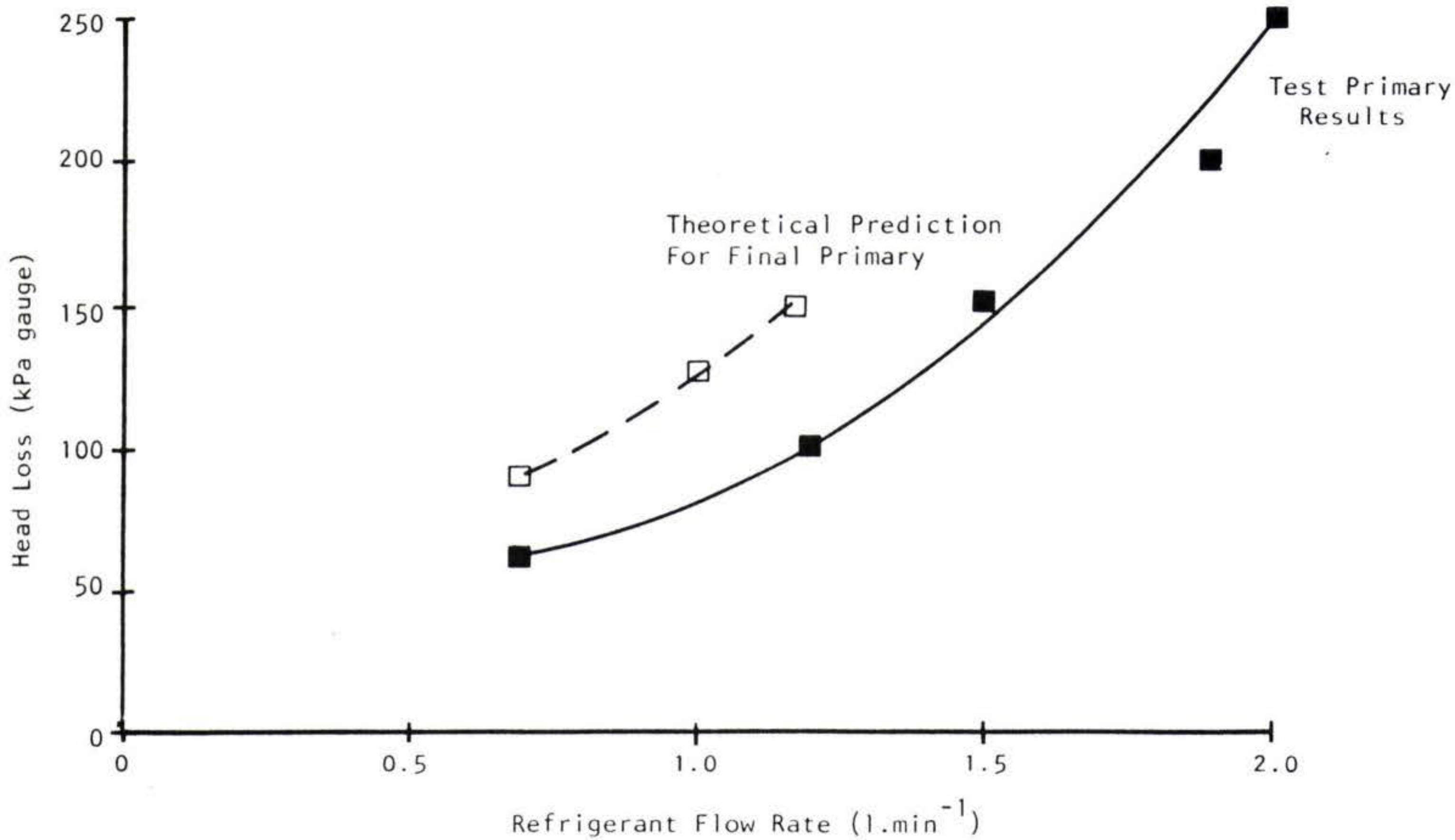


FIGURE 5:4

Relationship between head loss
and refrigerant flow

5:5 PRIMARY HEAT EXCHANGER DESIGN

5:5.1 Basic Arrangement

To avoid excessive head pressures a maximum headloss of 150 kPa was selected for the primary heat exchanger. Based on a refrigerant flow range of 1.7 $l.min^{-1}$ to 2.4 $l.min^{-1}$ obtained from initial tests it was necessary to split the refrigerant flow into two parallel pathways, each with a flow range of 0.85 $l.min^{-1}$ to 1.2 $l.min^{-1}$. The resulting headloss would therefore be in the range of 75 to 100 kPa for a 1 m length.

To maintain counter flow in the heat exchanger the water path was also split.

5:5.2 Heat Exchanger Dimensions

The total length of heat exchanger was calculated under various conditions using the equation

$$L = \frac{A}{a} = \frac{Q}{a.U.\Delta t_m} \qquad \text{.....Eqn 5:1}$$

where L = length of heat exchanger/leg (m)

A = heat exchanger area - refrigerant side (m^2)

a = heat exchanger area per unit length (m^2/m)

Q = heat absorbed by the water (kW)
= Eqn 4:2

U = overall thermal conductance ($kW.m^{-2}.{}^oC^{-1}$)

Δt_m = Log mean temperature difference (^oC)

The conditions used for the calculation were:-

1) Materials and dimensions as in the test heat exchanger

2) Refrigerant inlet temperature of 84^oC
(Based on results from tests for Section 5:3)

3) Refrigerant outlet temperature of $37^{o}C$
(Condensation temperature at 800 kPa)

4) Refrigerant flows of 0.7, 1.0 and 1.2 $l.min^{-1}$

5) Water inlet temperatures of $15^{o}C$ and $30^{o}C$

6) Water outlet temperature of $60^{o}C$
(Based on the discussion in Section 2:7)

7) Total water flow rate of 0.625 $l.min^{-1}$.

U values for the required refrigerant flows were read from Figure 5:3. Values of L, calculated from Eqn 5:1, are presented in Table 5:4.

5.5:3 Final Design

Based on an average refrigerant flow of 1.0 $l.min^{-1}$ for each leg, the final length selected was 1.5 m. To keep the heat exchanger compact the final arrangement was four 0.75 m tubes arranged as two parallel units 0.75 m long in series. The final arrangement is shown in Figures 5:5 to 5:9.

The expected headloss of the arrangement is presented in Figure 5:4 and was inside the required 150 kPa maximum.

Calculations showed that no other combinations of commercially available cores and tubes would produce higher Re and U values. Increasing water flow rates, while improving the theoretical performance of the system, was rejected as additional water would be of little value in a commercial situation.

TABLE 5:4

Determination of the length of heat exchanger per leg
(a = 0.28 m^2/m)

Refrigerant Flow $l.min^{-1}$/leg	$\Delta t = (60-15)^{o}C$, $\Delta t_m = 23.3^{o}C$, $Q = 1.005$ kW			$\Delta t = (60-30)^{o}C$, $\Delta t_m = 14.4^{o}C$, $Q = 0.670$ kW		
	U ($kW.m^{-2}.^{o}C^{-1}$)	A (m^2)	Length (m)	U ($kW.m^{-2}.^{o}C^{-1}$)	A (m^2)	Length (m)
1.2	0.110	0.393	1.40	0.127	0.366	1.26
1.0	0.103	0.418	1.49	0.117	0.396	1.44
0.7	0.960	0.449	1.61	0.110	0.423	1.51

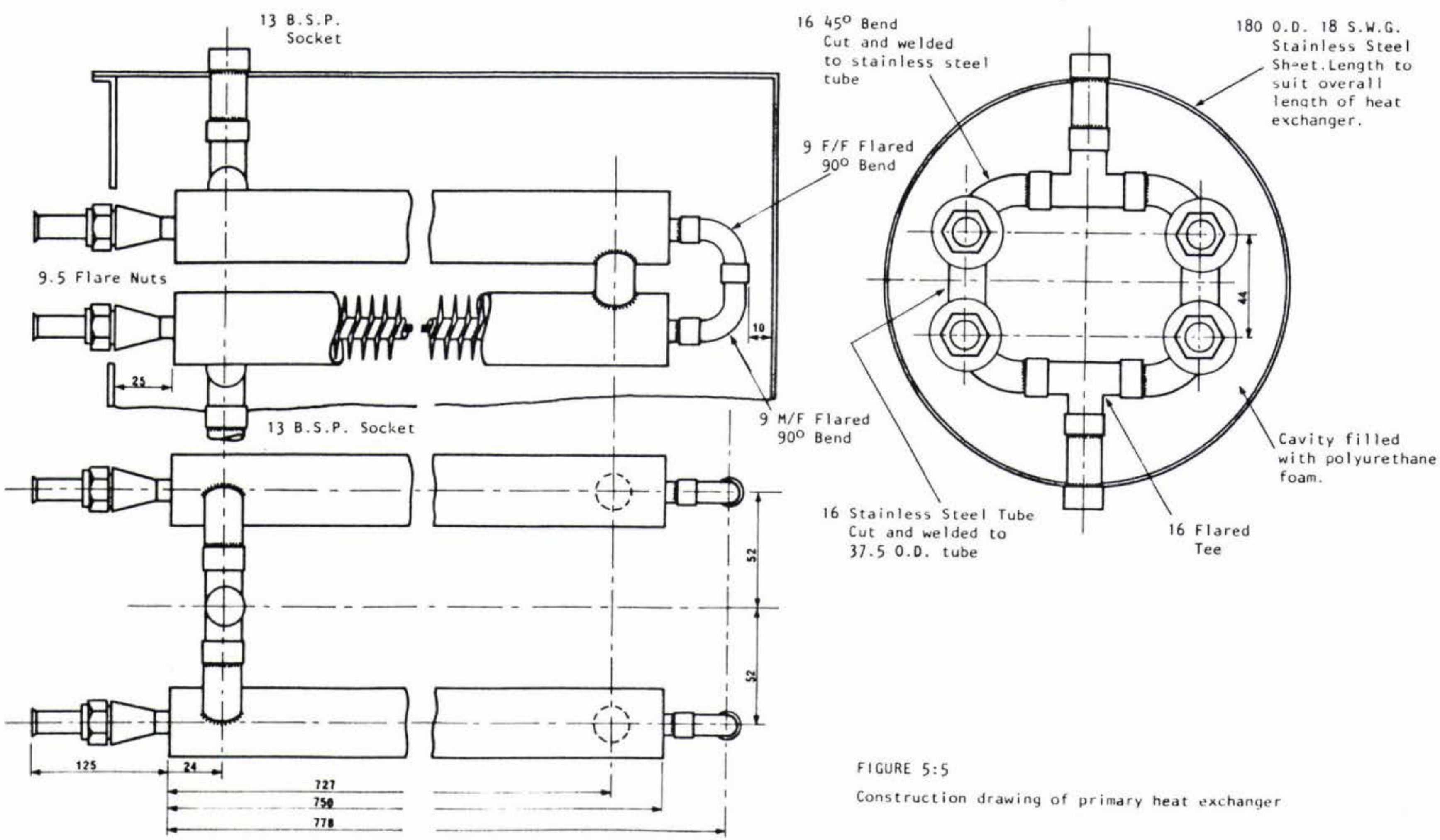


FIGURE 5:5

Construction drawing of primary heat exchanger

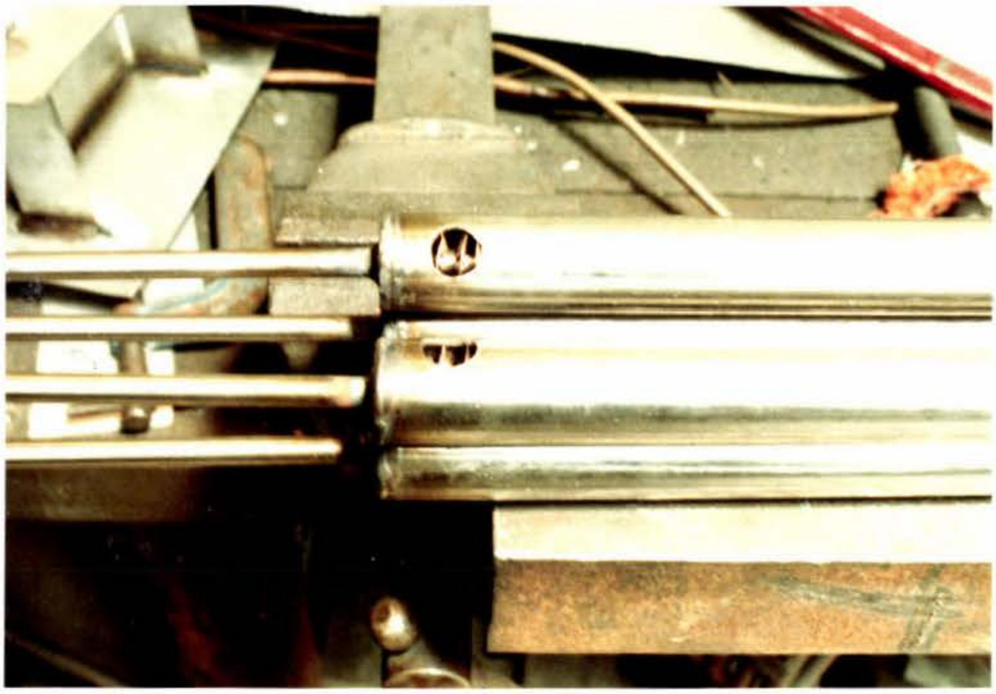


FIGURE 5:6

Plate showing primary heat exchanger tubes

FIGURE 5:7

Plate showing heat exchanger end connection

FIGURE 5:8

Plate showing primary heat exchanger in outer casing prior to filling with insulation

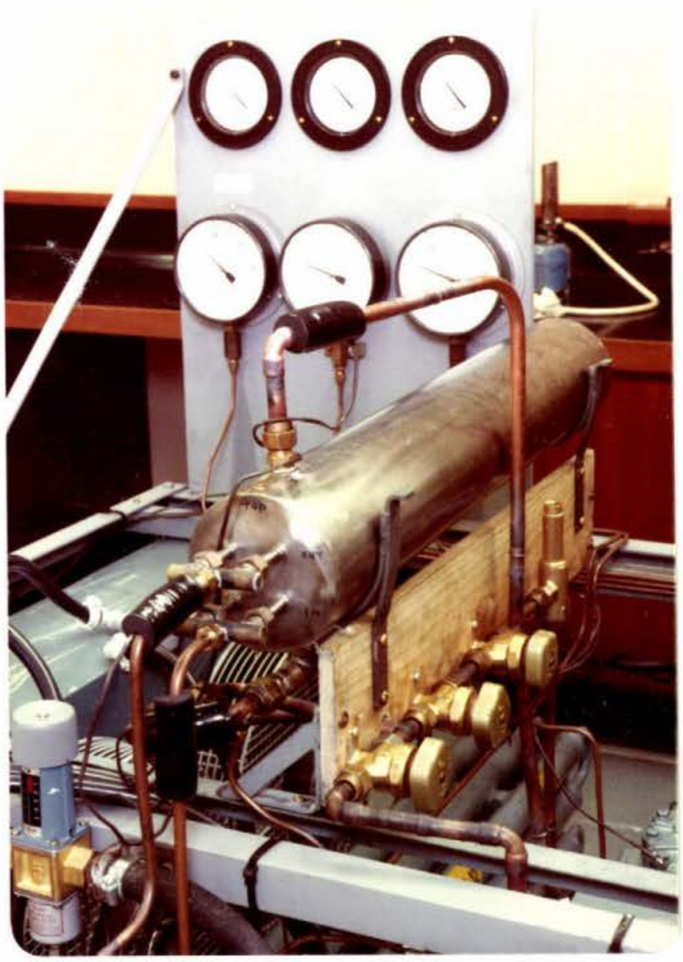


FIGURE 5:9

Plate showing primary heat exchanger installed in experimental plant

CHAPTER 6

EXPERIMENT II RESULTS AND DISCUSSION

6:1 INTRODUCTION

The presentation of results and discussion is in three stages. The effect of condenser system and pressure, the effect of milk temperatures and the effect of the primary heat exchanger on the performance of the system is discussed in Sections 6:2 to 6:8. Results and discussion for the evaluation of the primary heat exchanger, in terms of the quantity of heat recovered and the water outlet temperature, is presented in Sections 6:9 to 6:11. The evaluation of the overall refrigeration/heat recovery system's performance and economic viability is discussed in Sections 6:12 to 6:14.

For Sections 6:2 to 6:8 the results are presented separately for the 800 litre and 1200 litre loadings since each loading resulted in different operating conditions.

The nomenclature for the runs discussed in this chapter is the same as that presented in Table 3:2.

Results in the text are presented in figure form supported by tables of ranges, means and errors in Appendix 6:1. However, tables listing values referred to in the discussion are presented in the text as appropriate.

Significance tables are used to indicate whether runs are significantly different or not in situations where there is no conclusive trend in the data.

6:2 COOLING TIMES

The lack of control over the environmental loading required the correction of cooling times for a constant total loading (milk* plus environmental).

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* Throughout the discussion in this chapter the milk substitute (water) will be referred to as milk.

The results of the correction calculations are summarised in Tables 6:1 and 6:2. Analysis of vat loadings showed that average environmental load was 9% of the total load (i.e., heat extracted). Differences for the corrected cooling times of greater than 10 minutes were considered significant.

6:2.1 The Effect of Condenser Pressure and Condenser System

The effect of condenser system and pressure, and their relationship with environmental loads, can be seen in Figure 6:1.

In general, the effect of condenser system on cooling times was negligible, except for the A6:5 run (Table 6:3) under an 800 litre loading. The significant reduction in cooling time for this run, compared with the corresponding run for the water cooled system, is not understood.

Cooling times for the 800 litre loading (Figure 6:1) showed that there was no significant effect due to small changes in condenser pressure. There was, however, a significant difference between the maximum and minimum condenser pressures, as indicated in Table 6:3. The lack of differences can be attributed to the relatively short cooling times involved.

The position changed under 1200 litre loading conditions, with the effect of condenser pressure on cooling times being significant in all cases except the W6.5 and W7.5 runs. This change is understandable, due to the longer cooling times amplifying the insignificant differences associated with the 800 litre loading rate.

Cooling times for all the 800 litre runs and the two lowest condenser pressures for the 1200 litre runs were less than those specified in the regulations. Condenser pressures of 10 bars and 12 bars failed to meet the requirements by a maximum of 17 minutes (Table 6:2) when cooling 1200 litres of milk from 23^{o}C to 4^{o}C in conjunction with an environmental load of 8500 kJ.

TABLE 6:1

Cooling loads and rates for the refrigeration system - 800 litres

	Run (as per Table 3:2)	Heat Extracted (kJ)	Heat Added (kJ)	Heat Environment (kJ)	Cooling Time (min)	Cooling Rate ($kJ.min^{-1}$)	Cooling Time	
							Without Env. (min)	With Env.* (min)
WATER	W12	73859	63660	10198	175	422	150	166
	W10	72760	63643	9116	170	428	148	164
	W10/18	53466	45660	7805	135	396	118	130
	W10/SSH	71220	64447	6773	175	407	156	172
	W7.5	71767	64430	7336	165	435	146	161
	W7.5/7	51346	54401	2944	130	441	120	144
	W7.5/7/18	41406	37958	3447	100	414	89	98
	W7.5/18	53319	46431	6888	130	410	114	126
	W6.5	76301	65619	10681	165	462	138	152
AIR	A12	69985	63660	6324	175	400	160	175
	A12/NP	69749	63660	6088	180	387	164	181
	A10	69748	62873	6874	165	423	151	166
	A10/NP	65620	64062	1557	160	410	155	171
	A10/18	52632	46046	6586	135	390	120	132
	A7.5	68817	62019	6798	165	417	152	168
	A7.5/NP	64094	61299	2794	150	427	150	164
	A7.5/7B	70538	62488	8049	160	441	145	159
	A7.5/18	52715	46832	5882	135	391	120	132
	A6.5	64929	63258	1670	150	433	147	162

* 6500 kJ for 23-4^{o}C 4750 kJ for 18-4^{o}C } 9% of milk loading
5500 kJ for 23-7^{o}C 3750 kJ for 18-7^{o}C }

TABLE 6:2

Cooling loads and rates for the refrigeration system - 1200 litres

	Run (as per Table 3:2)	Heat Extracted (kJ)	Heat Added (kJ)	Heat Environment (kJ)	Cooling Time (min)	Cooling Rate ($kJ.min^{-1}$)	Cooling Time Without Env. (min)	Cooling Time With Env.* (min)
WATER	W12	105064	94888	10176	300	350	273	297
	W10	101779	94888	6891	275	370	258	281
	W10/18	81950	68683	13267	230	356	197	215
	W10/SSH	108522	96479	12043	305	355	268	292
	W7.5	105082	95290	9792	270	389	245	267
	W7.5/7°C	85586	80622	4964	210	407	197	215
	W7.5/7/18	58783	53999	4784	160	367	151	164
	W7.5/18	78742	70241	8501	215	366	192	209
	W6.5	105896	97249	8647	275	385	248	270
AIR	A12	107703	94888	12815	305	353	270	294
	A12/NP	109310	98421	10889	305	358	266	290
	A10	99653	93732	5921	275	362	263	287
	A10/NP	108289	96093	12196	295	367	260	283
	A10/18	80227	68298	11929	230	349	202	219
	A7.5	104193	93732	10461	280	372	256	279
	A7.5/NP	93906	91757	2149	245	383	250	271
	A7.5/7B	105179	94519	10660	275	382	250	272
	A7.5/18	83973	71815	12158	220	381	185	200
	A6.5	108262	94888	13374	265	408	233	255

* 8500 kJ for 23-4°C　　6250 kJ for 18-4°C
7000 kJ for 23-7°C　　5000 kJ for 18-7°C　　} 9% of milk loading

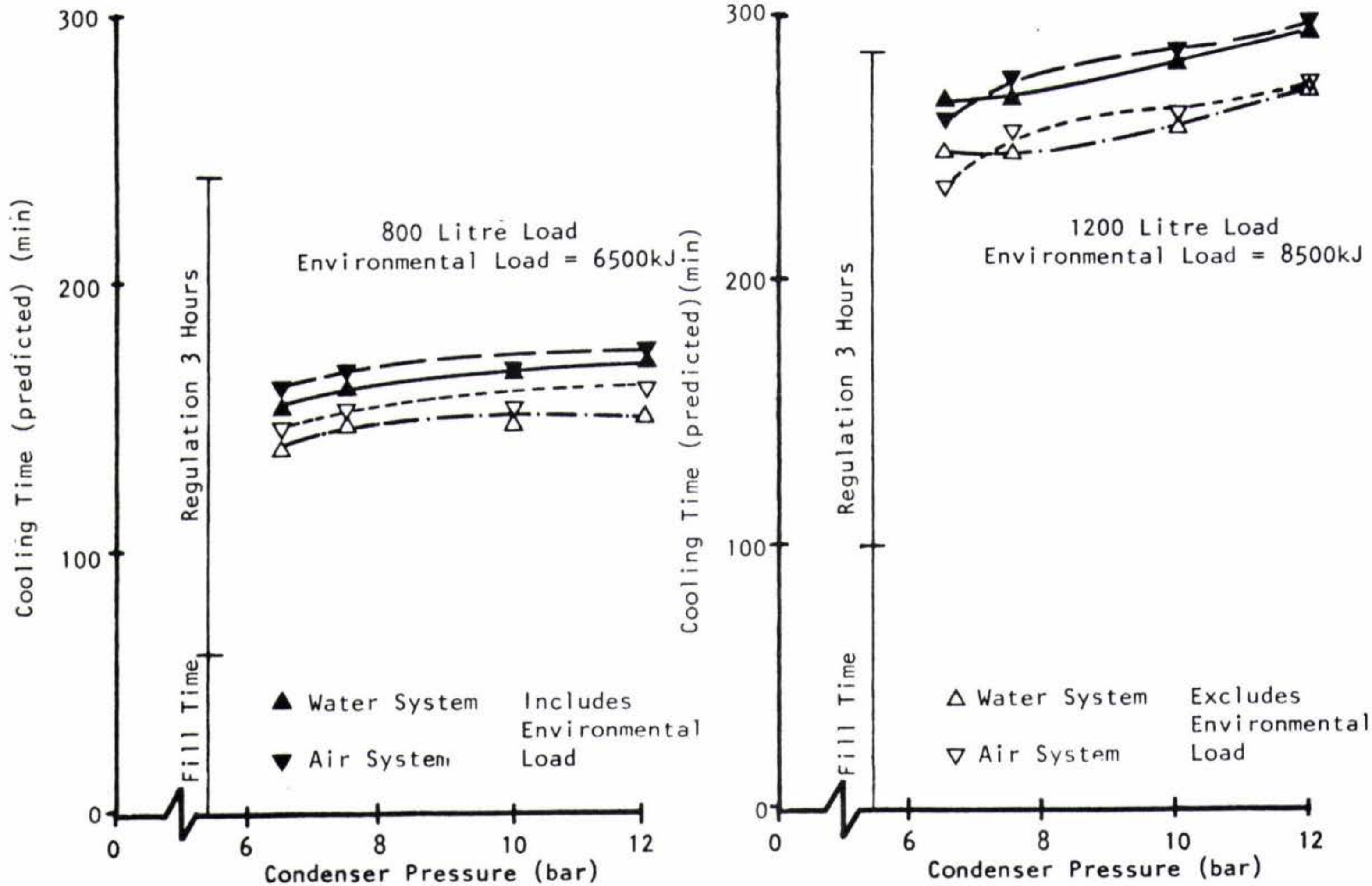


FIGURE 6:1

The effect of condenser pressure on cooling times

TABLE 6:3

Significance tables for 800 and 1200 litre loading
(with environmental effect)

NO denotes no significant difference
YES denotes significant difference

800 LITRE LOADING

Water System

	6.5	7.5	10.0	12.0
6.5		NO	YES	YES
7.5	NO		NO	NO
10.0	YES	NO		NO
12.0	YES	NO	NO	

Air System

	6.5	7.5	10.0	12.0
6.5		NO	NO	YES
7.5	NO		NO	NO
10.0	NO	NO		NO
12.0	YES	NO	NO	

Air System

Water System	6.5	7.5	10.0	12.0
6.5	YES	YES	YES	YES
7.5	NO	NO	NO	YES
10.0	NO	NO	NO	YES
12.0	NO	NO	NO	NO

1200 LITRE LOADING

Water System

	6.5	7.5	10.0	12.0
6.5		NO	YES	YES
7.5	NO		YES	YES
10.0	YES	YES		YES
12.0	YES	YES	YES	

Air System

	6.5	7.5	10.0	12.0
6.5		YES	YES	YES
7.5	YES		NO	YES
10.0	YES	NO		NO
12.0	YES	YES	NO	

Air System

Water System	6.5	7.5	10.0	12.0
6.5	YES	NO	YES	YES
7.5	YES	YES	YES	YES
10.0	YES	NO	NO	YES
12.0	YES	YES	YES	NO

For a zero environmental load applicable to insulated vat systems , these two condenser pressures would have met the cooling regulations.

The exclusion of the primary heat exchanger (runs A12/NP, A10/NP and A7.5/NP) did not have any measurable effect on the total cooling time.

6:2.2 The Effect of Milk Inlet and Final Temperature

The effect of milk inlet temperature and final temperature on cooling times, for the two condenser pressures tested, is shown in Figures 6:2 and 6:3 for the air and water condenser systems respectively.

Analysis of the results shows that there was a significant difference in total cooling times due to the effect of milk inlet and final temperature. The percentage differences in cooling times between different temperature conditions closely followed the expected theoretical differences (Table 6:4). (Theoretical differences were calculated on the basis of the temperature differential tested as a percentage of the 23°C - 4°C temperature differential.)

This means that cooling time was largely dependent on cooling load (i.e., milk temperatures, since volumes were constant) with the small differences between the experimental and theoretical results being due to the lower cooling rates at lower milk inlet temperatures (Figure 6:4).

6:2.3 The Effect of Increasing Receiver Pressure

Figure 6:3 indicates that there was no significant reduction in cooling time due to an increase in receiver pressure from 6.0 bars to 7.0 bars.

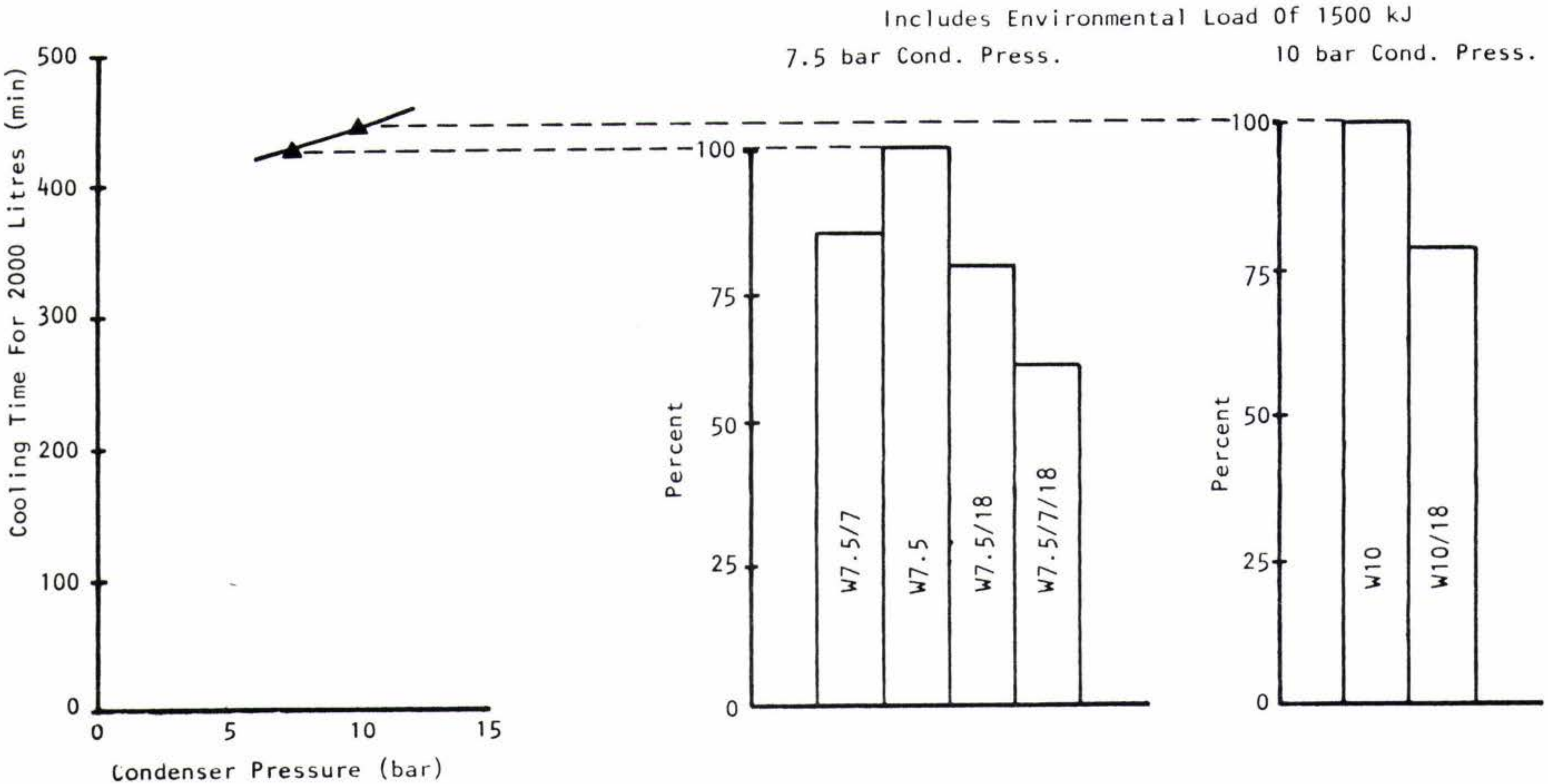


FIGURE 6:2

The effect of milk inlet and final temperature on cooling times - water system

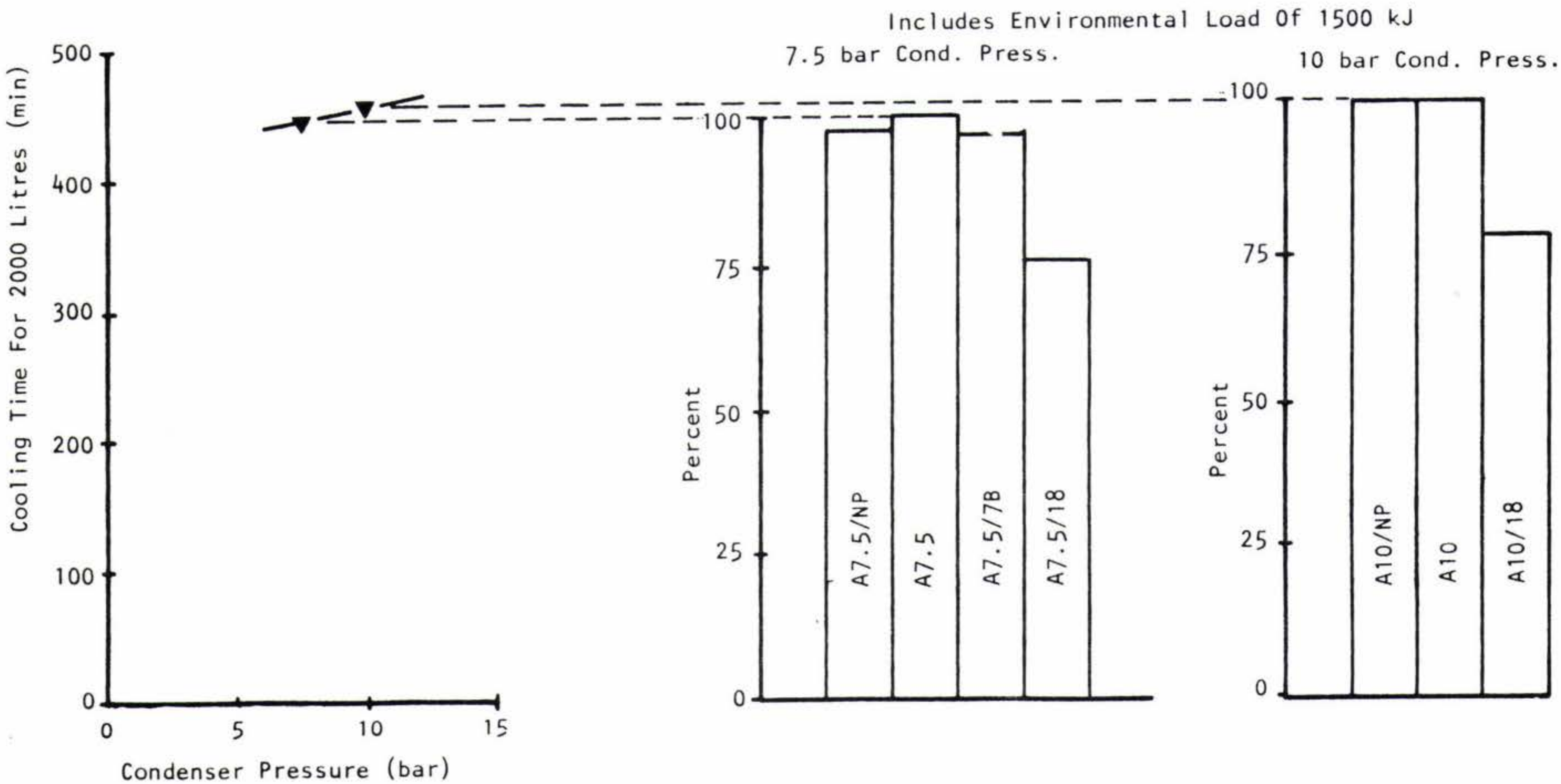


FIGURE 6:3

The effect of milk inlet temperature, primary heat exchanger and receiver pressure on cooling times - air system

TABLE 6:4

Experimental and theoretical cooling time as a percentage of the cooling time required for a $23^{0}C$-$4^{o}C$ differential

Run	$18^{o}C$ - $4^{o}C$		$23^{o}C$ - $7^{o}C$		$18^{o}C$ - $7^{o}C$	
	Exp.	Theo.	Exp.	Theo.	Exp.	Theo.
W7.5	78	74	85	84	61	58
A7.5	75	74				
W10	78	74				
A10	74	74				

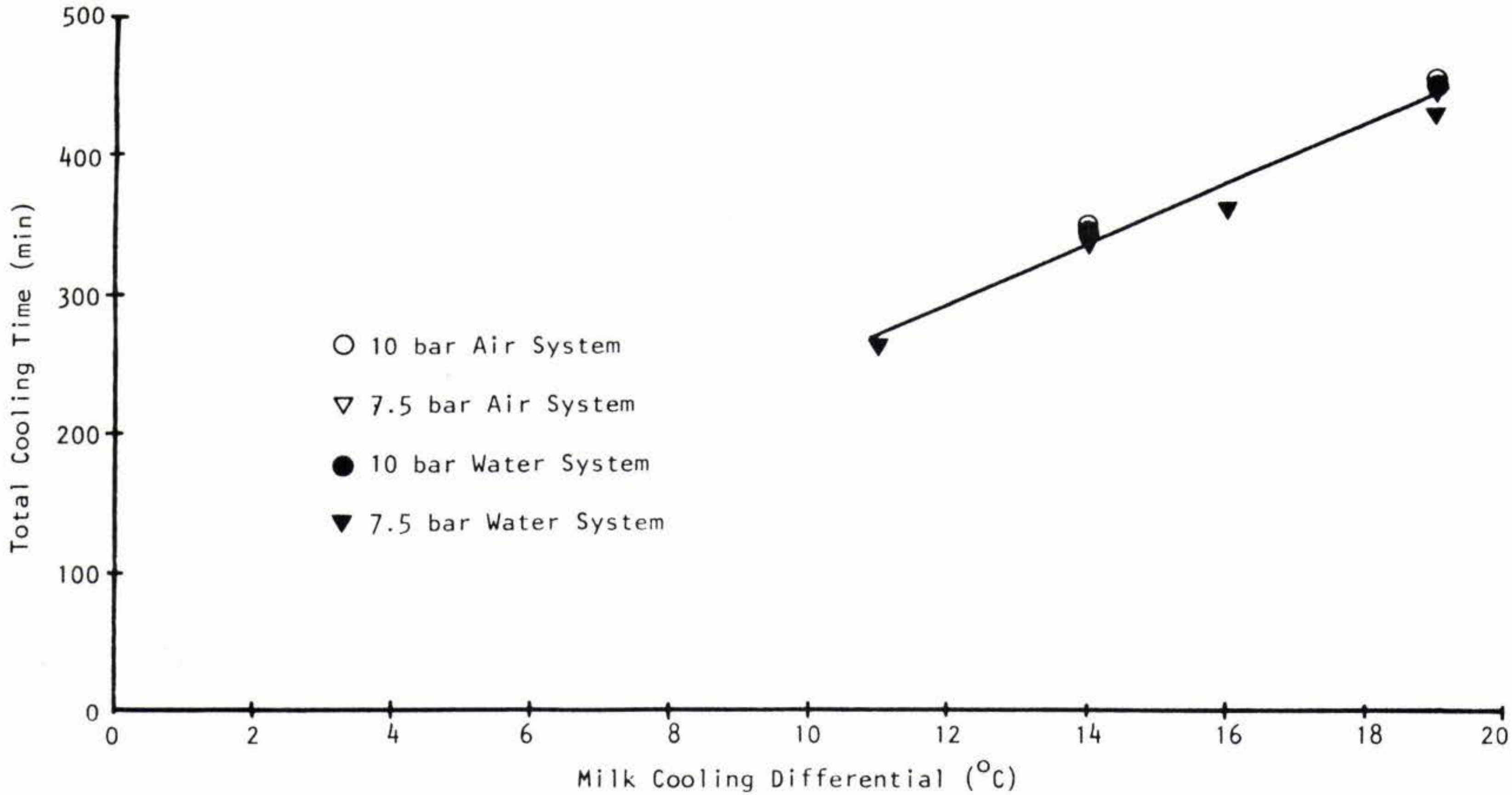


FIGURE 6:4

The effect of milk cooling differential on cooling time

6:2.4 Conclusions on Cooling Times

For constant milk volumes, cooling times increased under conditions of increasing condenser pressure, milk inlet temperatures, environmental loads and decreasing final milk temperatures. In particular, the combination of high condenser pressures (10 bar and 12 bar), milk inlet temperature (23^{o}C), environmental load (8,500 kJ) and low milk final temperature (4^{o}C) resulted in the system failing to meet the cooling regulation by up to 17 minutes when cooling 60% of the total daily production (1200 litres).

The most significant factor governing cooling times was the temperature differential through which the milk was cooled. For example, when milk inlet temperature was reduced by 5^{o}C total cooling times, on average, were reduced by 100 minutes (1.66 hours). Since the final milk temperature is set by regulation, precooling of the milk with a plate heat exchanger (or equivalent device) is required to ensure that the cooling requirements are met, given the present sizing of refrigeration plants for vat cooling.

Operating the system with either an air cooled or water cooled condenser had no effect on cooling times. Isolating the primary heat exchanger from the circuit, or increasing the receiver pressure to 7 bar, had no significant effect on cooling times.

6:3 VAT TEMPERATURE

Changes in vat temperature over time are presented in Tables 6:1 and 6:2 as a function of the average cooling rates (total load ÷ total time). Therefore, the differences in vat temperature due to changing condenser and milk temperature conditions are discussed in terms of the cooling rates.

6:3.1 The Effect of Condenser Pressure and Condenser System

Data for vat temperature under constant loading conditions is presented in Figure 6:5, for the water cooled system, and Figure 6:6, for the air cooled system.

For the 800 litre loading, there was some variation in the maximum vat temperature (13°C to 16°C) after 30 minutes due to small differences in milk inlet temperatures. This variation was reduced to insignificance after filling was completed. However, after 150 minutes the variation in vat temperature had increased again to 1.5°C, with temperatures ranging from 5.5°C for the 12 bar condenser pressure to 4.0°C for the 6.5 bar condenser pressure. This effect was due to the decrease in cooling rates with increasing condenser pressure (Table 6:1).

The effect of condenser pressure on maximum vat temperature, for the 1200 litre loading, was only slight with the temperature for both condenser systems, at the completion of filling, ranging from 10.0°C to 10.5°C. The exception to this was the A12 run which had a maximum mix temperature of 11°C. Analysis of the cooling load (Table 6:2) shows that this run had 8100 kJ more heat to extract than the A10 run.

As for the 800 litre run, the variation in vat temperature for the 1200 litre loading changed from no significant difference at the start of the run to a difference, after 240 minutes, of 1°C (4.5°C for the 6.5 bar condenser pressure to 5.5°C for the 12 bar condenser pressure). This variation was due to the different cooling rate for each condenser pressure.

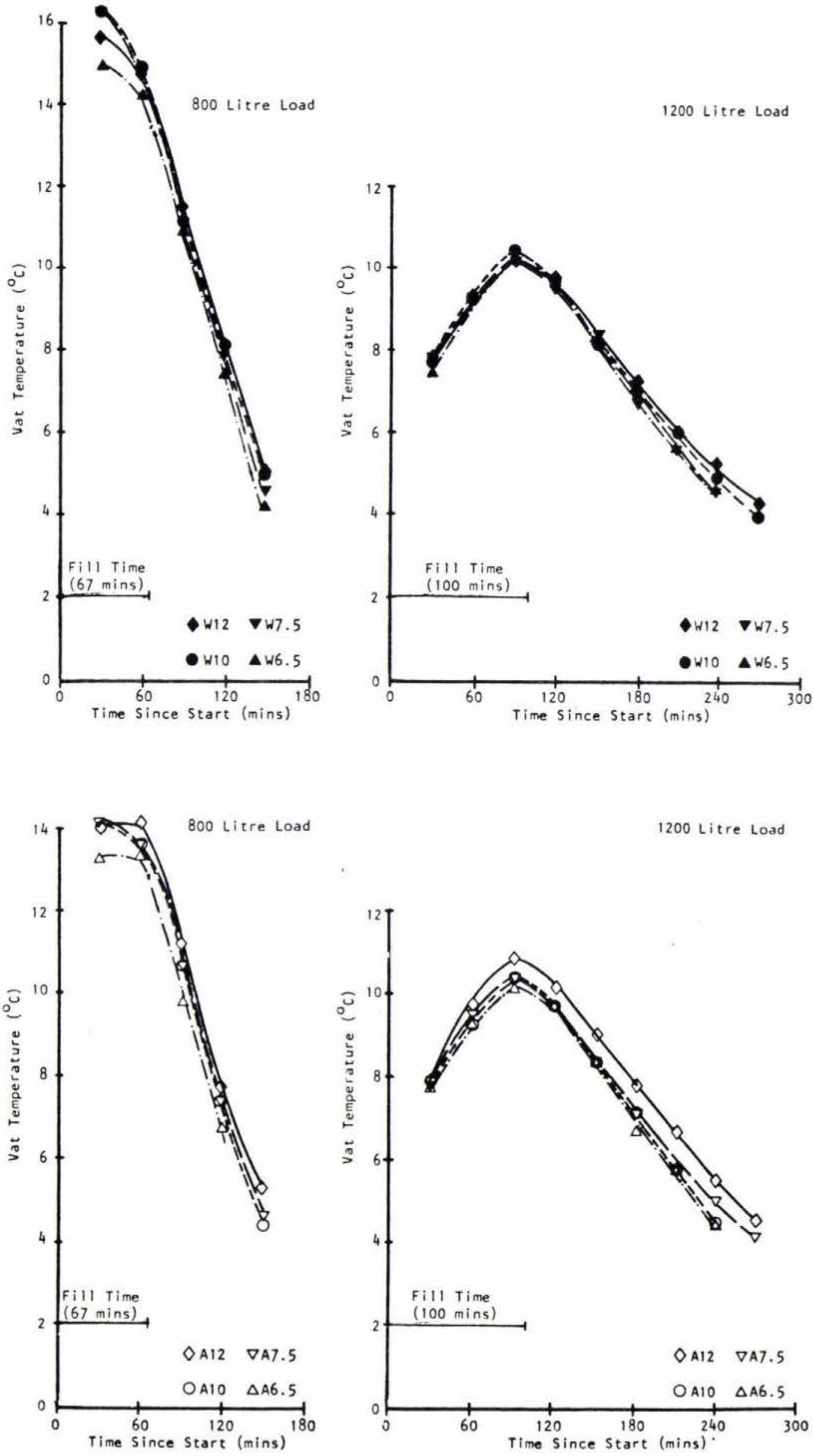
The shallower slopes in the cooling curves for the 1200 litre loading, compared to those for the 800 litre loading, were due to two factors, namely:-

FIGURE 6:5

The effect of condenser pressure on vat temperature - water system

FIGURE 6:6

The effect of condenser pressure on vat temperature - air system


800 Litre Load
1200 Litre Load
Vat Temperature (°C)
Time Since Start (mins)
Fill Time
(67 mins)
(100 mins)
W12 W7.5
W10 W6.5
800 Litre Load
1200 Litre Load
Vat Temperature (°C)
Time Since Start (mins)
Fill Time
(67 mins)
(100 mins)
A12 A7.5
A10 A6.5

1) The increased loading of 1200 litre runs, in terms of the increased volume of milk to be cooled and the increase in environmental load as a result of the longer cooling times.

2) The reduced cooling rates due to the lower overall vat temperatures for the 1200 litre loading compared with the 800 litre loading.

The comparison of Figures 6:5 and 6:6 shows that there was no significant difference in the cooling curves due to changing condenser systems once system loading had been completed.

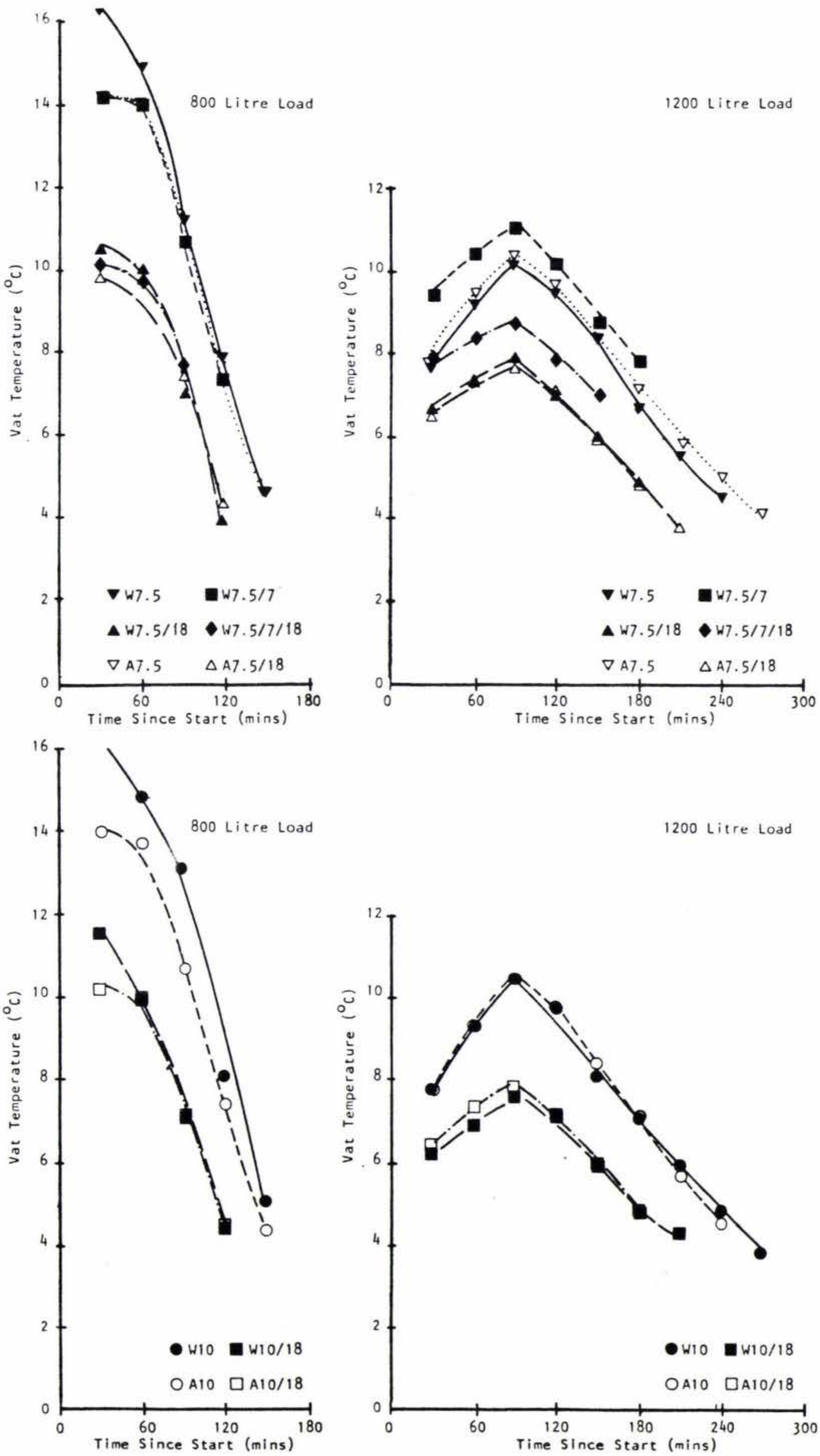
6:3.2 The Effect of Changing Milk Outlet and Final Temperature

Vat temperature data for the 7.5 bar and 10 bar condenser pressures is presented in Figure 6:7 and 6:8 respectively.

Reducing milk inlet temperature by 5°C, from 23°C to 18°C, significantly reduced the maximum vat temperature by up to 6°C at the 30 minute interval for the 800 litre loading. This difference had decreased to 3°C after 90 minutes and remained constant until the final temperature was reached. The high initial temperatures for the W7.5 and W10 runs were due to the milk inlet temperature being slightly above 23°C but below 24°C.

Increasing final temperature for the 800 litre loading had no effect on vat temperatures since the inlet temperatures were the same. The combined effect of reducing milk inlet temperature by 5°C and increasing milk final temperature by 3°C (W7.5/7/18) was to produce a curve similar to one for run W7.5/18 but was shorter due to the reduced loading.

For the 1200 litre loading, a comparison of the curves for runs with milk inlet temperatures of 23°C and 18°C shows that decreasing the milk inlet temperature by 5°C reduced the maximum mix temperature by 2.5°C. This figure is less than the expected difference of 3°C (5°C x 1200 l / 2000 l) due to


800 Litre Load
1200 Litre Load
Vat Temperature (°C)
Time Since Start (mins)
W7.5
W7.5/7
W7.5/18
W7.5/7/18
A7.5
A7.5/18
800 Litre Load
1200 Litre Load
W10
W10/18
A10
A10/18

the slightly higher cooling rates for the 23°C inlet temperature compared with the cooling rates for the 18°C milk inlet temperature (Table 6:1).

Increasing milk final temperature for the 800 litre loading increased the maximum mix temperature from 10.2°C to 11.2°C when the 1200 litre volume was added. This result compares well with the expected result of 1.2°C (3°C x 800 l / 2000 l) with the small difference being due to the slightly higher cooling rates for the W7.5/7 run compared with the W7.5 run (Figure 6:7).

Increasing final temperature by 3°C, and reducing milk inlet temperature by 5°C, produced a curve which had a maximum mix temperature of 8.8°C under 1200 litre loading conditions. This temperature was 1.5°C below the 23°C/4°C inlet/final temperature curve (run W7.5). The difference in temperature of 1.5°C was lower than the expected difference of 1.8°C ([1200 l x 5°C - 800 l x 3°C] / 2000 l) due to the slightly higher cooling rates associated with the W7.5 run compared with the W7.5/7/18 run.

A comparison of air and water condenser systems (Figures 6:5 and 6:6) again shows no significant difference nor does a comparison of Figures 6:7 and 6:8 show any significant difference due to condenser pressure.

6:3.3 Conclusions on Vat Temperature

The overall effect of condenser pressure or condenser system on vat temperature was not significant. However, small differences of 1°C were recorded after 90% of the cooling time had elapsed.

Reducing milk inlet temperatures by 5°C had the most significant effect, with initial vat temperatures being reduced by 4°C, on average, reducing to 3°C later in the run. The mix temperature for the 1200 litre loading was reduced by 2.5%.

Increasing the temperature at which refrigeration was discontinued for the 800 litre loading increased the mix temperature for the subsequent 1200 litre loading by $1^{o}C$. While this difference was experimentally significant, it was of no practical consequence.

While there is no legislation restricting mix temperatures in milk vats, it is generally recommended (Currier, 1976) that they do not rise above $10^{o}C$, for milk quality reasons (Section 2:1.2.2).

6:4 REFRIGERANT FLOW

6:4.1 The Effect of Condenser Pressure and Condenser System

Refrigerant flow rate data is presented graphically in Figures 6:9 and 6:10 for the water and air cooled system respectively.

The results show that under an 800 litre loading condition there was no significant difference in refrigerant flow due to condenser pressure. The small differences in runs W7.5, W10 and W12 compared with the corresponding runs for the air cooled system were due to the higher loading temperatures for the water cooled runs, since loading temperature has a direct effect on refrigerant flow, due to the action of the expansion valve.

There was only a small effect on refrigerant flow due to changing condenser pressures for the 1200 litre loading rate. The trend (Figure 6:9 and Figure 6:10) indicates that increasing condenser pressure reduces refrigerant flow rate, for the reasons discussed in Section 2:3.2.2.

There was no effect on refrigerant flow rate caused by changing condenser systems.

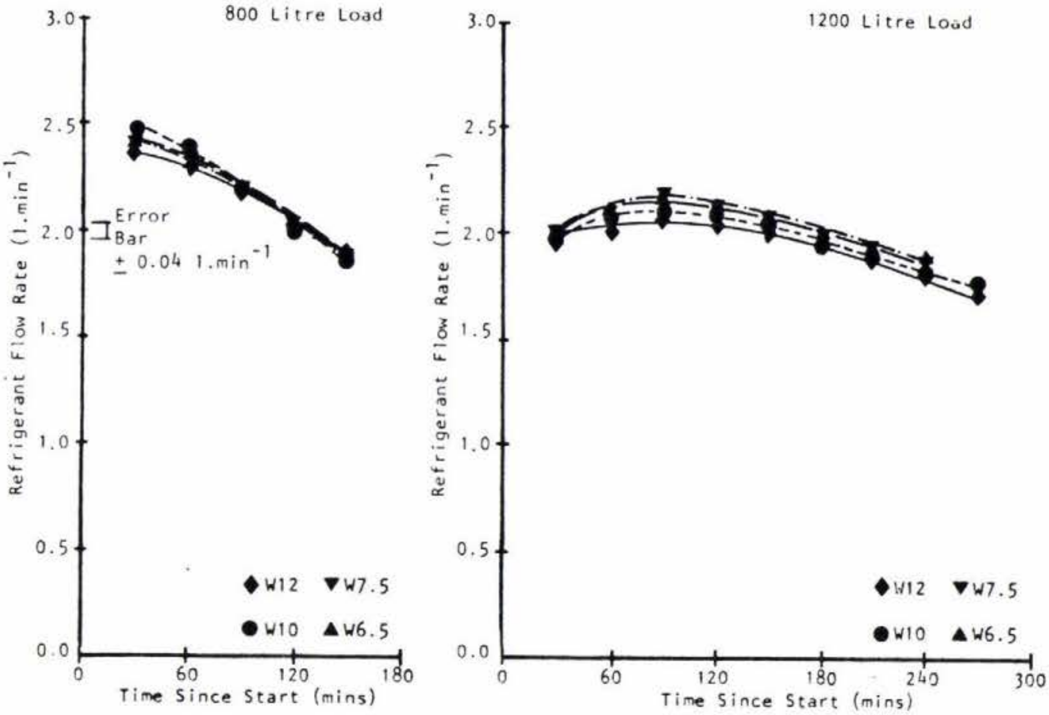


FIGURE 6:9

The effect of condenser pressure on refrigerant flow rates - water system

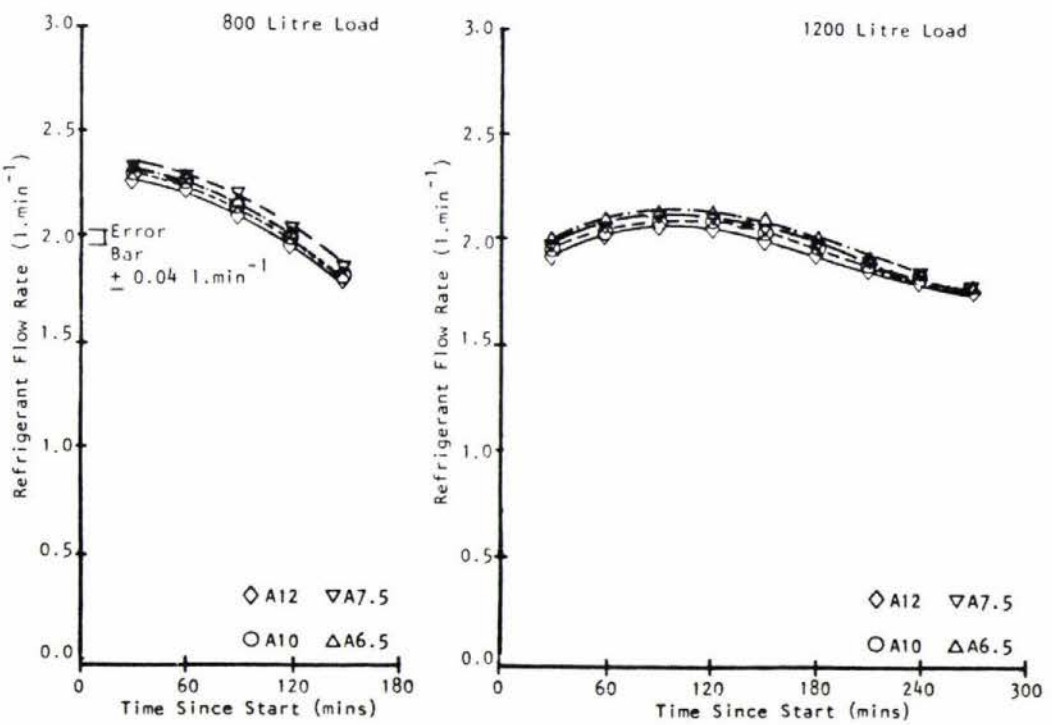


FIGURE 6:10

The effect of condenser pressure on refrigerant flow rates - air system

6:4.2 The Effect of Changing Milk Inlet and Final Temperature

Decreasing milk inlet temperature by $5^{o}C$ ($23^{o}C$ to $18^{o}C$) significantly decreased refrigerant flow rate by an average of $0.22\ l.min^{-1}$ (10%) for the 7.5 bar condenser pressure (Figure 6:11) and $0.12\ l.min^{-1}$ (6%) for the 10 bar condenser pressure (Figure 6:12), when cooling 800 litres of milk. A similar effect was recorded for the 1200 litre load. These differences in flow rate were due to the lower vat temperature, associated with the lower milk inlet temperatures, causing the expansion valve to restrict the refrigerant flow in order to maintain the preset level of suction superheat (Section 2:3.2.2).

Increasing milk final temperature from $4^{o}C$ to $7^{o}C$ had no effect on refrigerant flow for the 800 litre loading since milk inlet temperatures were constant. However, the higher mix temperatures for the $7^{o}C$ final temperature runs, when cooling the additional 1200 litres (Section 6:3.2), gave a slight increase in refrigerant flow ($0.050\ l.min^{-1}$ to $0.075\ l.min^{-1}$) during the initial stages of the cooling cycle.

6:4.3 The Effect of Increasing Receiver Pressure

Comparison of the two curves for the 800 litre loading in Figure 6:13 shows that, at the 30 minute time interval, the refrigerant flow was slightly greater at the higher receiver pressure (7 bar) compared to the flow at the lower receiver pressure (6.0 bar). This difference became insignificant as the cooling process continued.

Under situations of high vat temperature (greater than $14^{o}C$), the expansion valve was fully open and refrigerant flow was dependent upon the pressure differential (receiver pressure - evaporator pressure) across the valve, hence, the initial trend in refrigerant flow. However, as the cooling process continued, the expansion valve closed to maintain the preset level of suction superheat, thus eliminating the effect of receiver pressure.

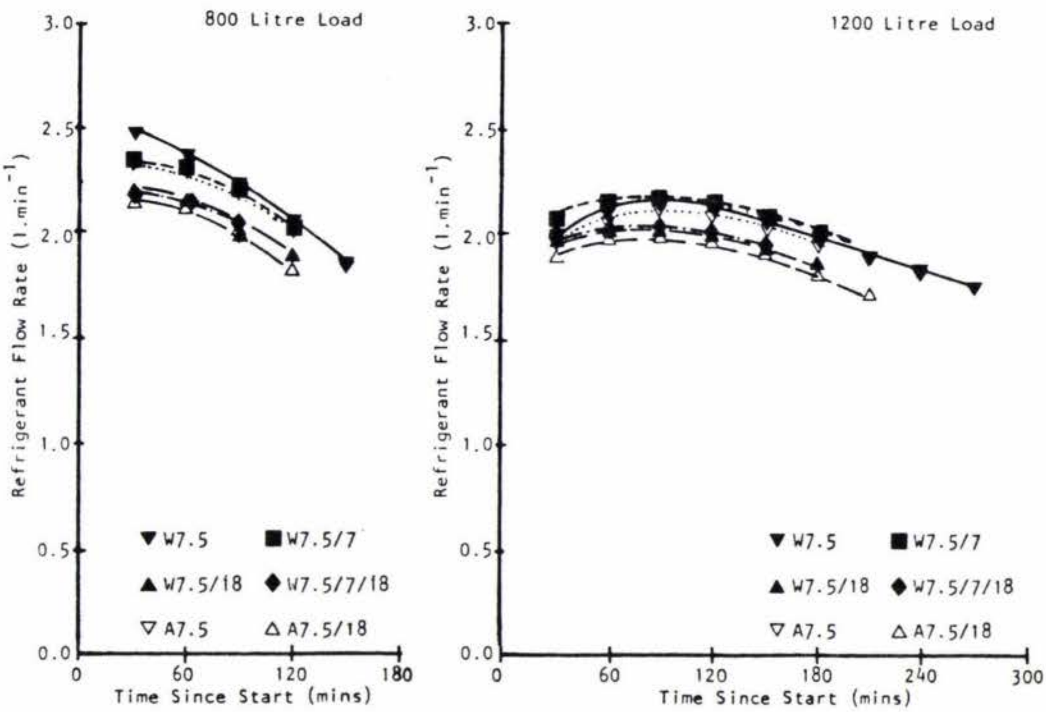


FIGURE 6:11

The effect of milk inlet and final temperatures on refrigerant flow rates - 7.5 bar condenser pressure.

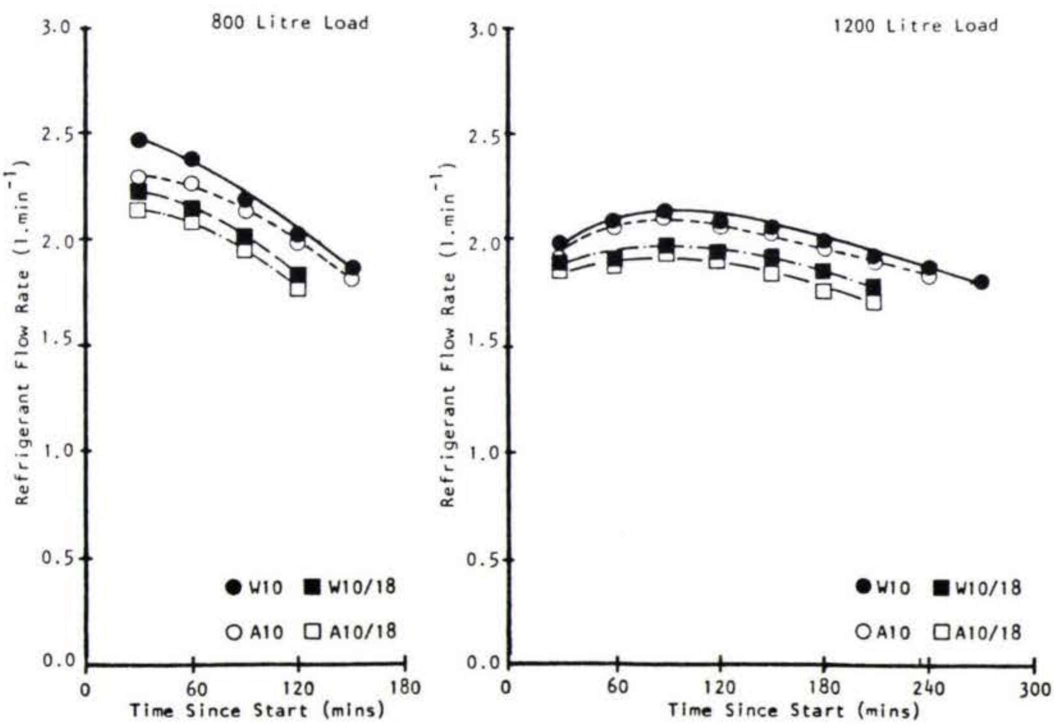


FIGURE 6:12

The effect of milk inlet temperatures on refrigerant flow rates - 10 bar condenser pressure.

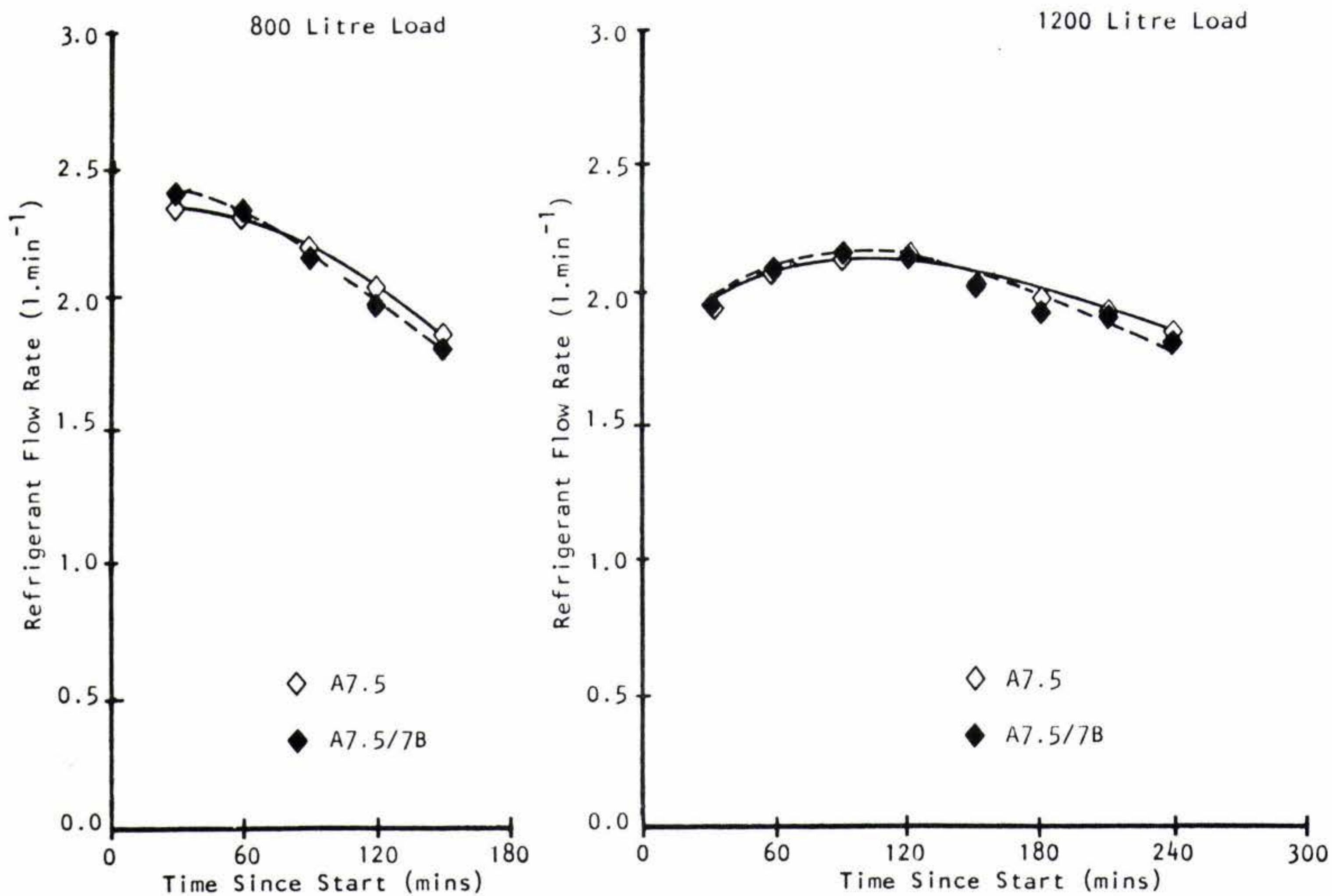


FIGURE 6:13

The effect of increasing receiver pressure from 6.0 bar to 7.0 bar on refrigerant flow

6:4.4 The Effect of the Addition of the Primary Heat Exchanger

Increasing head pressure by the inclusion of the primary heat exchanger was expected to reduce refrigerant flow rate (Section 2:3.2.1). Results (Figure 6:14) for the three runs designed to establish this effect show that the reduction in refrigerant flow due to the primary heat exchanger was insignificant, probably due to the fact that the expansion valve controlled refrigerant flow without the condenser circuit having any significant influence.

6:4.5 The Effect of Vapour By-pass

Differences in the results for the two flow meters are presented in Figure 6:15 for the air cooled condenser system. No data was recorded for the water cooled system as the by-pass was not required. Instead, runs for the water cooled condenser system were used to calibrate the evaporator flow meter against the receiver flow meter.

The results were inconclusive since virtually all the differences fell within the limits of the combined 5% error band. Most of the points lie above zero, indicating that the by-pass was operating, but the high degree of fluctuation was probably due to the intermittent action of all three refrigerant control valves, namely; the CPR, CPC and expansion valves. The exception to this was the inexplicable negative trend for the 6.5 bar condenser pressure.

The amount of potential heat available for heat recovery by-passing the primary heat exchanger was small as a result of the low heat capacity and flow rate of the refrigerant vapour.

6:4.6 Conclusion on Refrigerant Flow

Refrigerant flow rate was significantly reduced when milk inlet temperatures were reduced by $5^{o}C$. The effect of condenser pressure was minor, with only a slight trend of

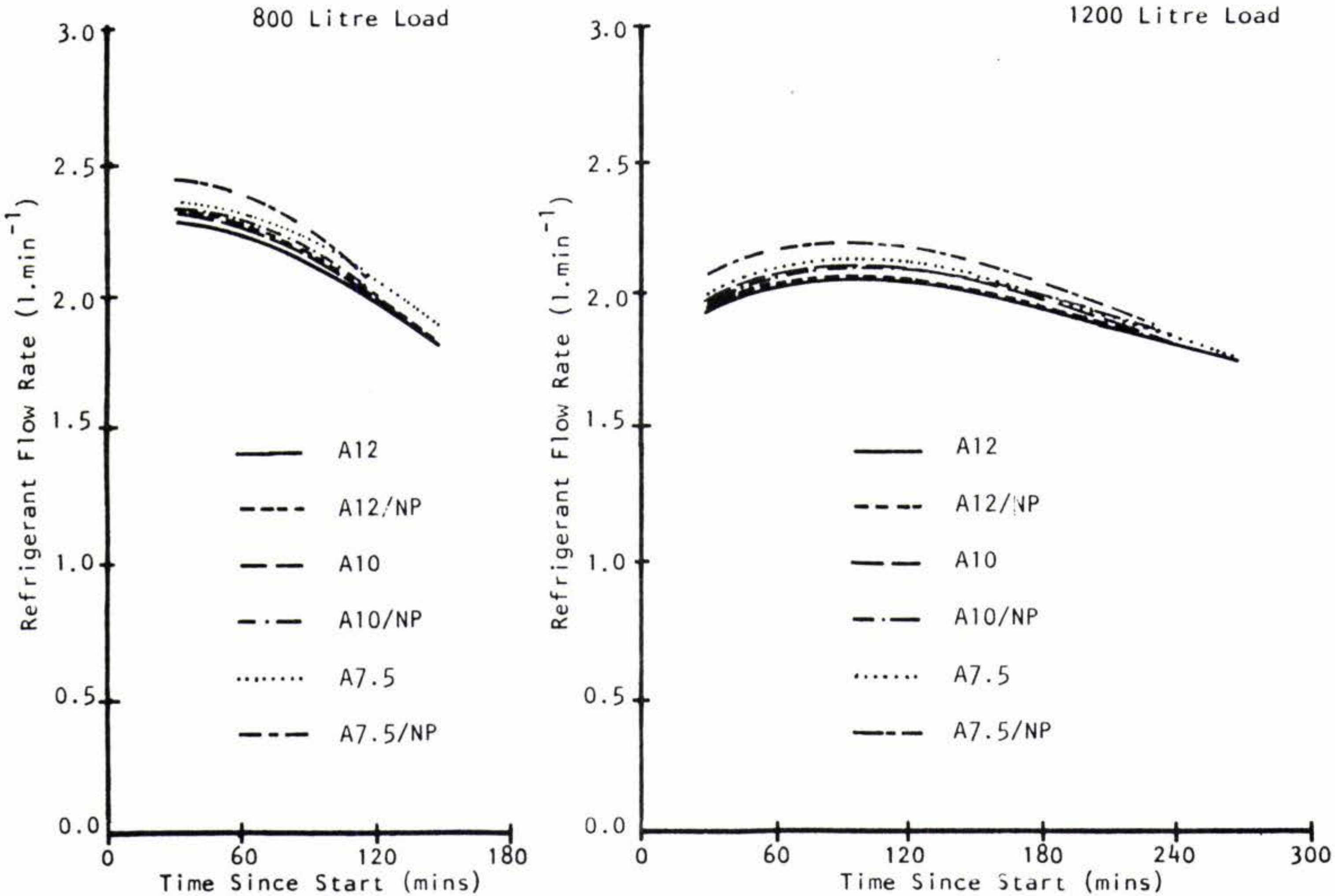


FIGURE 6:14

The effect of the inclusion of the primary heat exchanger on refrigerant flow

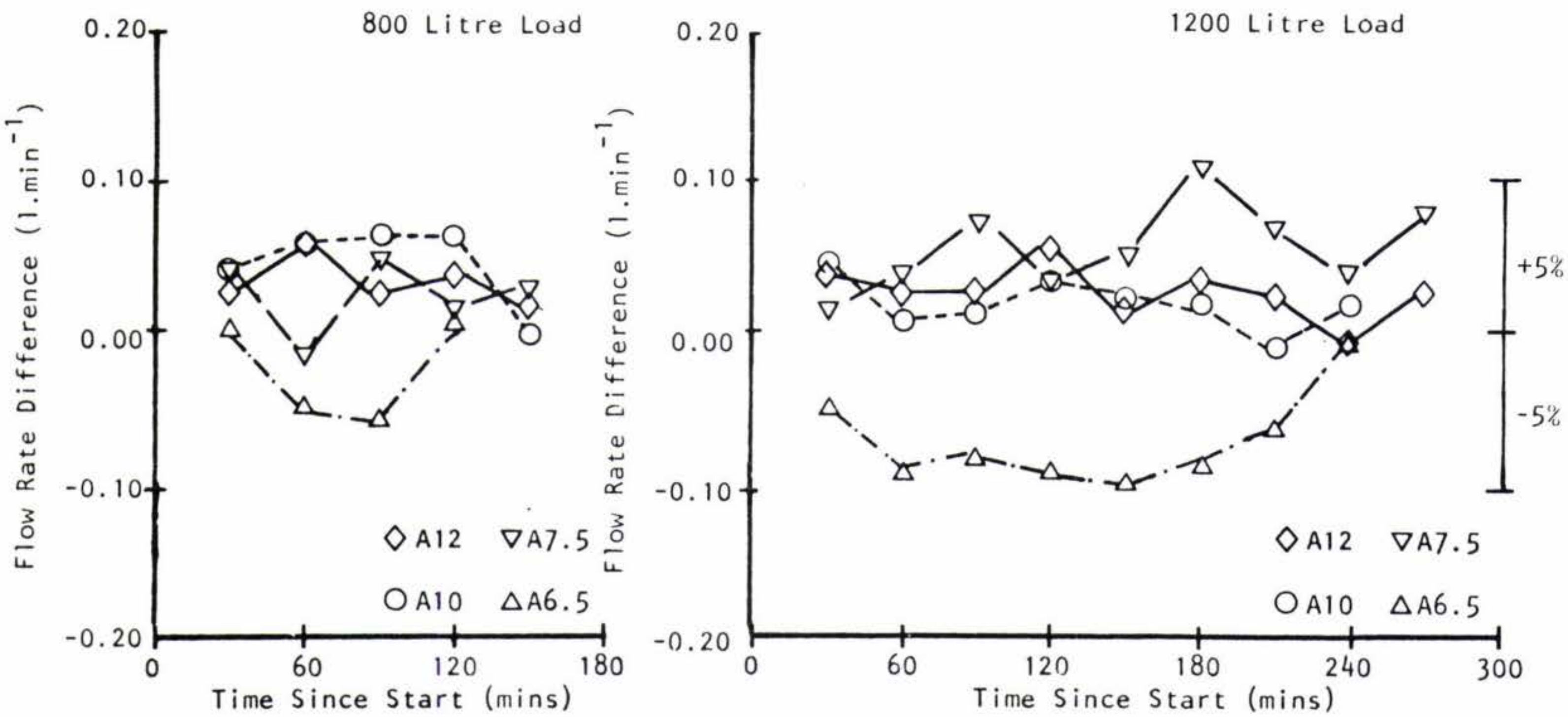


FIGURE 6:15

The effect of condenser pressure on the difference in refrigerant flow rate between receiver and evaporator flow meters - air system decreasing refrigerant flow with increasing condenser pressure being recorded for both condenser systems.

There were no measurable differences in flow rate between the two condenser systems, and no significant effects due to increasing receiver pressure or the inclusion of the primary heat exchanger.

The amount of refrigerant flow unavailable for heat recovery, due to the action of the by-pass valve, was less than 5% of the total flow rate and is of no practical consequence to the potential level of heat recovery.

6:5 REFRIGERATION EFFECT (R.E.)

6:5.1 The Effect of Condenser Pressure and Condenser System

The variation in R.E. (evaporator enthalpy change) with time, at different condenser pressures, is presented in Figures 6:16 and 6:17.

Increasing condenser pressure from 6.5 bars to 12 bars slightly decreased R.E., due to increasing expansion valve inlet temperatures with increasing condensation temperatures.

R.E. was a maximum when the vat temperature was a maximum (i.e., at the 30 minute interval of the 800 litre cooling cycle) and was a minimum at the completion of each cooling cycle. The R.E. curves have the same shape as those for refrigerant flow and vat temperature, i.e., R.E. rises to a peak at the end of loading then decreases as cooling continues. This rise and fall was due to the changing temperature differential across the expansion valve as the evaporator temperature changed to maintain a constant temperature differential between the boiling refrigerant and the milk.

A comparison of Figure 6:16 and Figure 6:17 shows that there was a small difference due to the condenser system with the

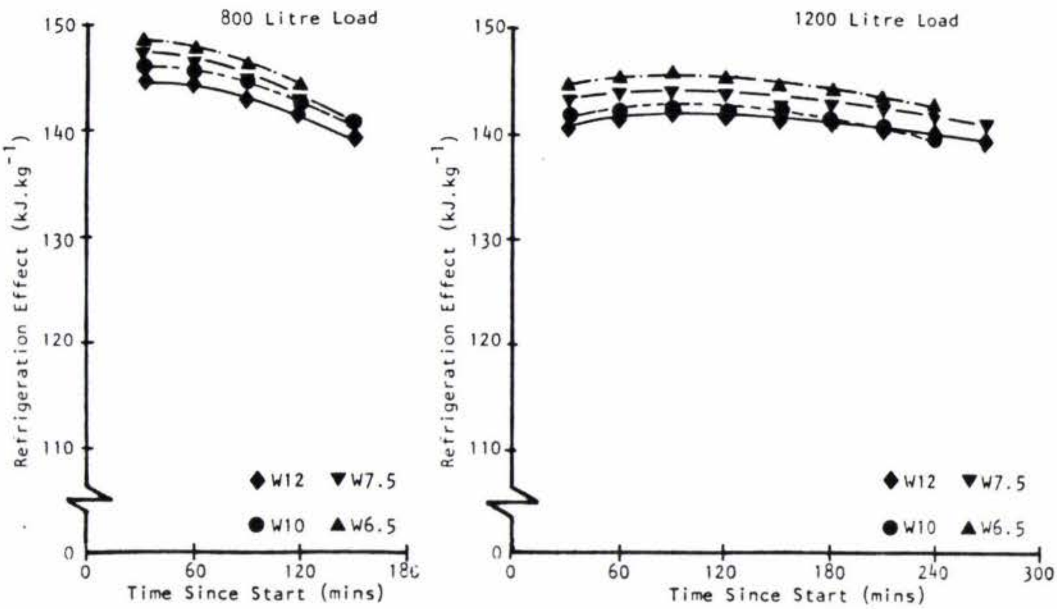


FIGURE 6:16

The effect of condenser pressure on refrigeration effect (RE) - water system

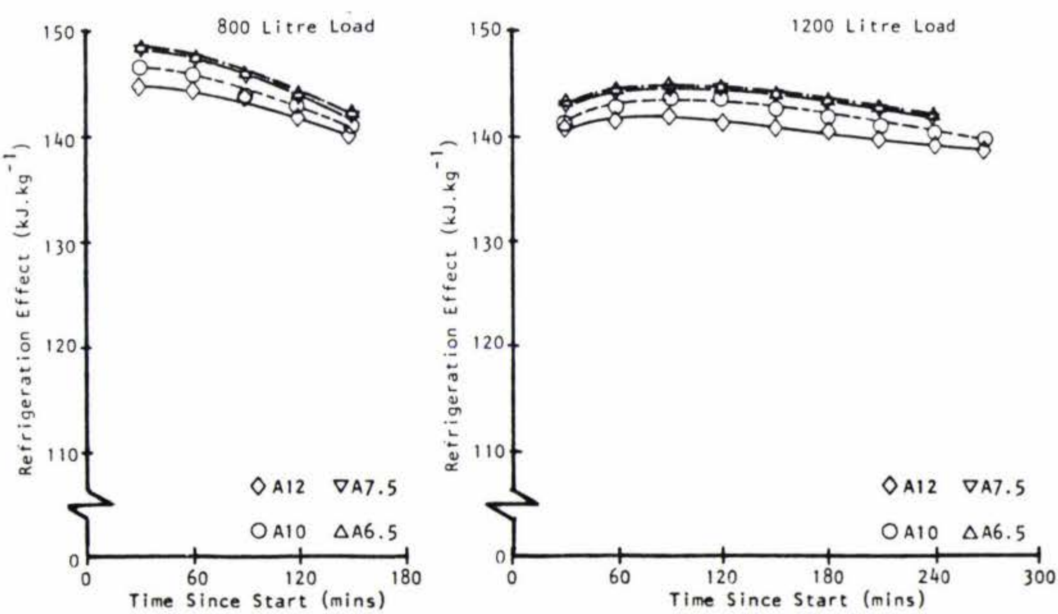


FIGURE 6:17

The effect of condenser pressure on refrigeration effect (RE) - air system air cooled system having R.E. values, on average, 1 $kJ.kg^{-1}$ higher than the water cooled system. A study of the temperatures of the liquid leaving the secondary heat exchangers showed that the air system had liquid refrigerant temperatures 2°C lower than the corresponding temperatures for the water cooled system. This effect was the result of the increased heat exchanger area available for subcooling in the air cooled system due to the action of the CPR valve (Section 4:1.1.1).

Although these differences in R.E. are small, their impact on system performance is increased when combined with refrigerant flow, to give the cooling rate, since refrigerant flow also decreases with increasing condenser pressure. The equation for cooling rate (Section 2:3.1.6) is:-

$$\text{Cooling rate } (kJ.s^{-1}) = \text{R.E. } (kJ.kg^{-1}) \times M_r \ (kg.s^{-1}) \qquad \text{.....Eqn 6:1}$$

where M_r = refrigerant flow ($kg.s^{-1}$)

6:5.2 The Effect of Changing Milk Inlet and Final Temperatures

The effect of reducing milk inlet temperature by 5°C reduced R.E. values, on average, by 3 $kJ.kg^{-1}$ and 1.5 $kJ.kg^{-1}$ for the 800 litre and 1200 litre loads respectively. Increasing milk final temperature had no measurable effect.

6:5.3 The Effect of Changing Receiver Pressure

Increasing receiver pressure, independently of condenser pressure, from 6.0 bar to 7.0 bar had no significant effect on R.E. This is understandable since it is expansion valve temperature differential that dictates the amount of "flash off" that occurs. In standard systems, condenser and receiver pressures are not controlled independently and therefore as condenser pressure increases, refrigerant liquid temperature increases (assuming a constant level of subcooling) with a resulting decrease in R.E.

6:5.4 Conclusion on R.E.

The effect of condenser pressure was small with only a a variation in R.E. of 3 $kJ.kg^{-1}$ at maximum vat temperature being recorded. The effect of changing condenser system was also small, with the air cooled system giving slightly higher results than the water cooled system. Reducing milk inlet temperature reduced R.E. but changing milk final temperature, or increasing receiver pressure, had no significant effect.

6:6 COMPRESSOR HEAD PRESSURE

Although head pressure is not a component of COP its effect on power consumption justifies some discussion.

6:6.1 The Effect of Condenser Pressure

Combined results of maximum and minimum head pressure for changes in condenser pressure are presented in Figure 6:18. The results for the air and water cooled condenser systems were combined, since the coolant used for condensation is of no consequence under constant condenser pressures.

The decreasing difference between the minimum refrigerant flow line and the zero flow line was due to the reduction in head loss as refrigerant flow decreased with increasing condenser pressure.

The difference between the maximum and minimum refrigerant flow lines reflects the small variation in refrigerant flow (Section 6:4) for a constant condenser pressure. The variation in head loss across the primary, at maximum refrigerant flow, was 1.0 bar (100 kPa) at a condenser pressure of 12 bar, and 1.5 bar (150 kPa) at a condenser pressure of 6.5 bar. These headloss figures are consistent

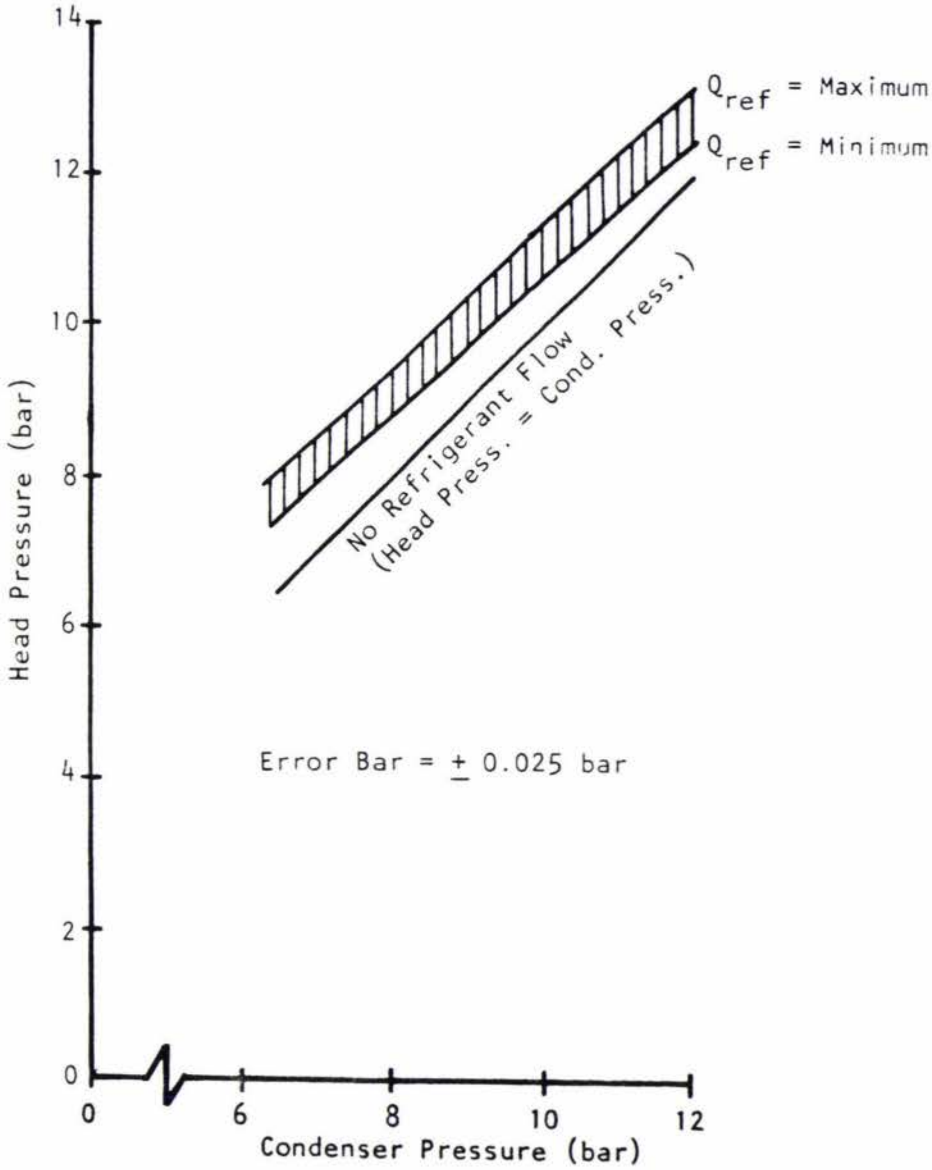


FIGURE 6:18

The effect of refrigerant flow on head pressure with the head losses expected when the heat exchanger was designed (Section 5:5).

:7 INSTANTANEOUS COMPRESSOR POWER CONSUMPTION (I.C.P.C.)

For this analysis, instantaneous power consumption (kW) was determined by calculation (Appendix A4:5.2.2).

6:7.1 The Effect of Changing Condenser Pressure

Figures 6:19 and 6:20 present the results for instantaneous compressor power consumption for the four condenser pressures and the two condenser systems tested.

The figures show that, for the 800 litre loading, instantaneous power consumption (in any one run) varied significantly by 0.4 kW during the cooling cycle in response to changes in compression ratio (Section 2:3.2.1) and refrigerant flow. The effect of increasing condenser pressure from 6.5 bar to 12 bar significantly increased instantaneous power consumption from 2.45 kW to 2.95 kW. This difference remained constant throughout the run.

For the 1200 litre loading, the effect of condenser pressure was similar to that for the 800 litre loading, with maximum instantaneous power consumption ranging from 2.35 kW (6.5 bar condenser pressure) to 2.85 kW (12 bar condenser pressure) at the 90 minute interval.

The relationship between instantaneous power consumption and condenser pressure can be seen in Figure 6:21.

6:7.2 The Effect of Changing Condenser System

The results of average instantaneous power consumption (Figure 6:21) and a comparison of Figures 6:19 and 6:20 show that there was no significant difference between the air cooled and water cooled condenser systems.

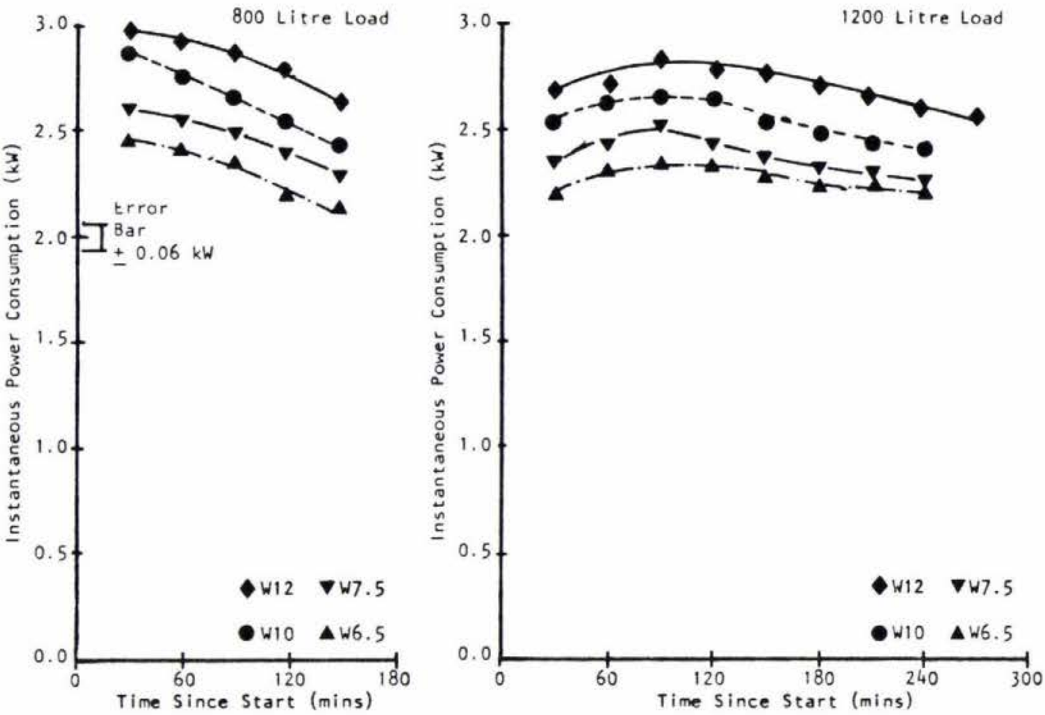


FIGURE 6:19

The effect of condenser pressure on instantaneous compressor power consumption (ICPC) - water system

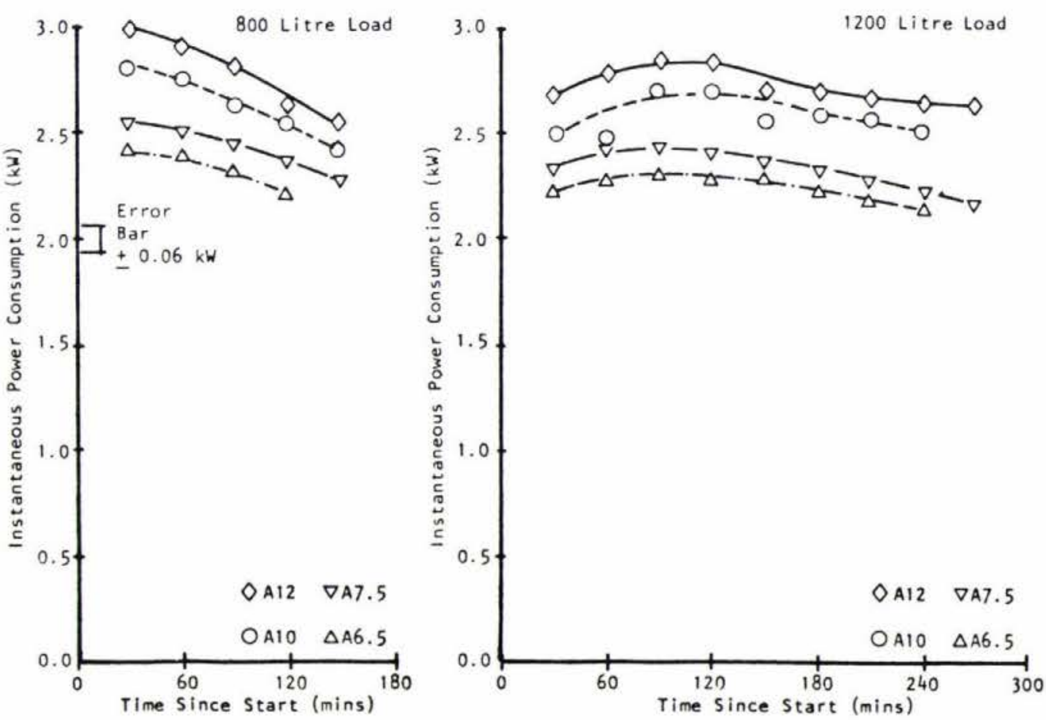


FIGURE 6:20

The effect of condenser pressure on instantaneous compressor power consumption (ICPC) - air system

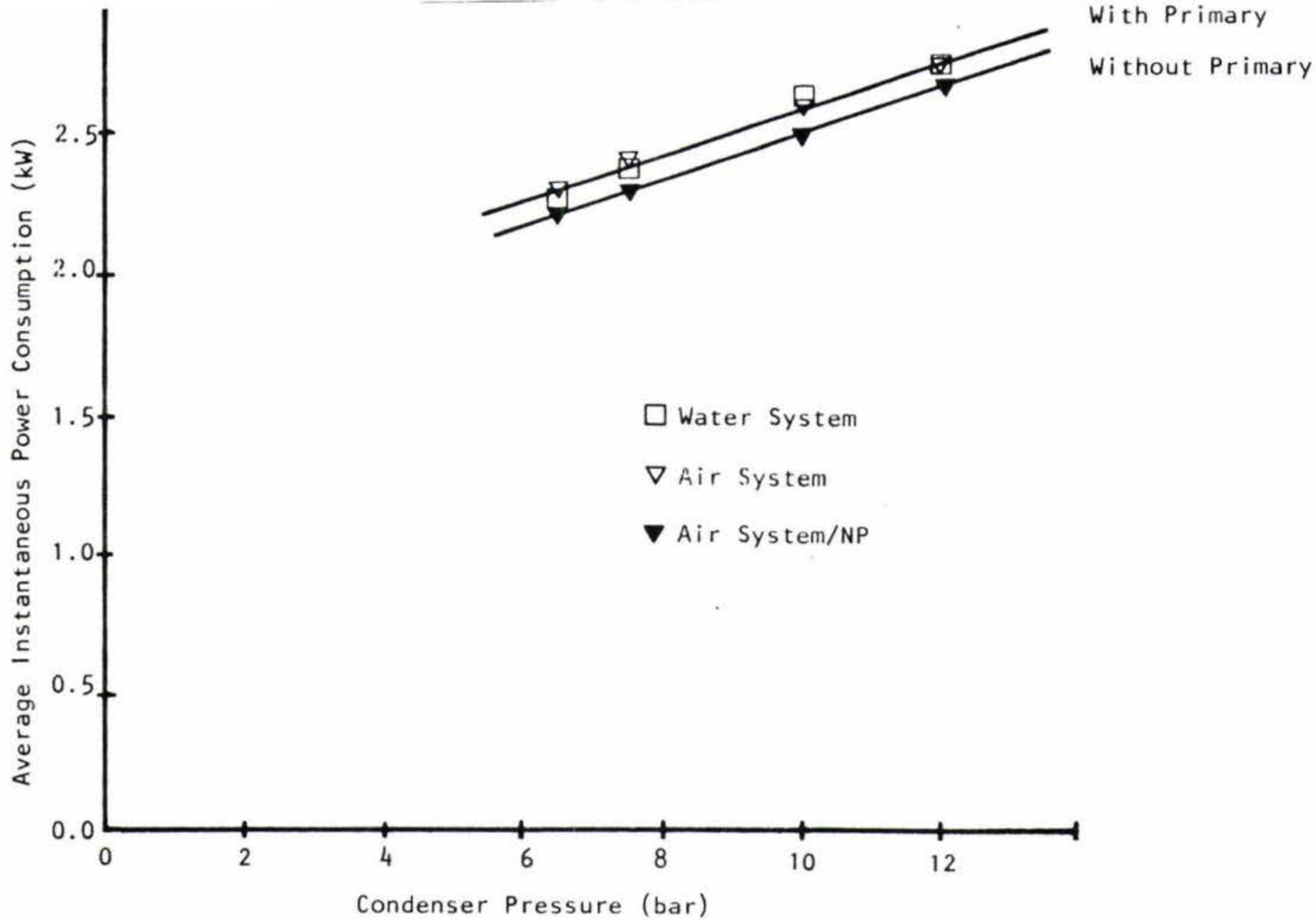


FIGURE 6:21

The effect of the primary heat exchanger on power consumption

This comparison makes no allowance for the power consumption of the fans (0.36 kW) used in the air cooled system. Fan power should be included as it constitutes an additional energy requirement of a commercial refrigeration system. In the case of the water cooled system, water is readily available on most farms and therefore no additional energy requirements have been included.

6:7.3 The Effect of the Primary Heat Exchanger

Results of the test runs with and without the primary heat exchanger isolated from the system are presented in Table 6:5. A plot of the average instantaneous power consumption for both loads is presented in Figure 6:21.

The equation for instantaneous power consumption with the primary heat exchanger, as a function of condenser pressure for the eight runs tested, is:-

$$Ip\ (kW) = b.Cp + c \qquad \text{.....Eqn 6:2}$$

where $b = 0.082 \pm 0.008$ kW/bar

$c = 1.77 \pm 0.07$ kW

Cp = condenser pressure in bars

The same analysis for the three runs without the primary heat exchanger gave the equation of:-

$$Ip'\ (kW) = b'.Cp + c'$$

where $b' = 0.0816 \pm 0.008$ kW/h

$c' = 1.677 \pm 0.08$ Kw/h

A comparison of the two y intercepts (c and c') shows that the effect of including the primary heat exchanger in the circuit was small but constant, with the difference being 0.10 ± 0.08 kW.

TABLE 6:5

Data summary for test runs with and without (NP) primary heat exchanger

800 LITRE LOAD

	A12	A12/NP	A10	A10/NP	A7.5	A7.5/NP
MAX	2.98 ±0.08	2.81 ±0.08	2.81 ±0.08	2.65 ±0.07	2.55 ±0.07	2.43 ±0.07
MIN	2.56 ±0.07	2.51 ±0.07	2.42 ±0.07	2.34 ±0.06	2.28 ±0.06	2.21 ±0.06
MEAN	2.77 ±0.07	2.69 ±0.07	2.63 ±0.07	2.52 ±0.07	2.43 ±0.07	2.35 ±0.07

1200 LITRE LOAD

	A12	A12/NP	A10	A10/NP	A7.5	A7.5/NP
MAX	2.85 ±0.08	2.79 ±0.08	2.71 ±0.07	2.58 ±0.07	2.45 ±0.07	2.35 ±0.06
MIN	2.65 ±0.07	2.47 ±0.07	2.52 ±0.07	2.29 ±0.06	2.16 ±0.06	2.18 ±0.06
MEAN	2.73 ±0.07	2.65 ±0.07	2.61 ±0.07	2.49 ±0.07	2.34 ±0.07	2.27 ±0.07

6:7.4 The Effect of Changing Milk Inlet and Final Temperature

Instantaneous power consumption was reduced slightly by 0.1 kW, on average, when the milk inlet temperature was reduced by $5^{o}C$, while increasing milk final temperature by $3^{o}C$ had no significant effect.

6:7.5 Total Refrigeration Power Consumption

Instantaneous power consumption does not indicate the total energy used by the refrigeration system in cooling a volume of milk to the required temperature. Figure 6:22 presents the total power consumption (integration of instantaneous power curves for the cooling times in Tables 6:1 and 6:2) for all the tests conducted. Included in the figure is the additional power required to operate the system above a condenser pressure of 6.5 bar with no primary heat exchanger (i.e., refrigeration system without heat recovery or condenser pressure control).

The results show that the air cooled system required more energy than the water cooled system, due to the power required to operate the fans.

The effect of increasing condenser pressure from 6.5 bar to 12 bar resulted in an increase in total power of 5.52 kWh/day.

Reducing milk inlet temperature by $5^{o}C$, or increasing milk final temperature by $3^{o}C$, significantly decreased total power consumption, due to the reduced cooling times, as discussed in Section 6:2.

The effect of environmental loading was to increase cooling times (Tables 6:1 and 6:2) and consequently total power consumption by 9% , on average. This value is consistent with the value of 10% obtained by Vickers (1980).

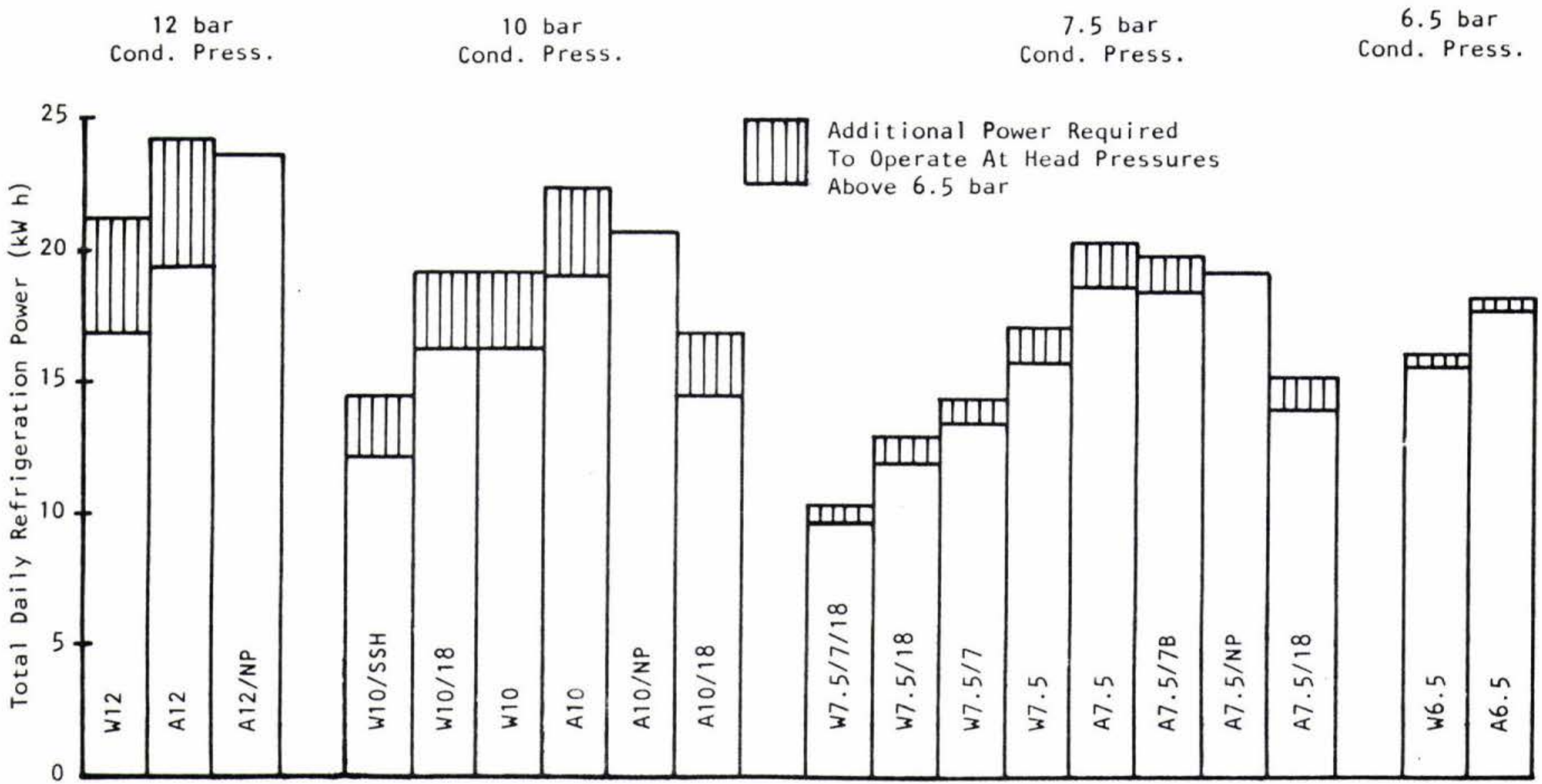


FIGURE 6:22

A comparison of total refrigeration power consumption between water and air cooled systems

6:7.6 Conclusions on Power Consumption

The effect of increasing condenser pressure from 6.5 bars to 12 bars significantly increased instantaneous power consumption by 0.5 kW for both 800 and 1200 litre loadings throughout the entire cooling period.

Changing condenser systems from water cooled to air cooled increased power consumption by 0.36 kW, as a result of the additional power required by the fans.

The inclusion of the primary heat exchanger increased instantaneous power consumption by 0.10 kW for the condenser pressures tested.

The effect of decreasing milk inlet temperature by $5^{o}C$ reduced instantaneous power consumption by 0.10 kW, on average, while increasing final temperature by $3^{o}C$ had no effect.

Total power consumption increased with increasing condenser pressure and milk temperature differential, due to the higher instantaneous power consumption and longer cooling times (Section 6:2).

6:8 COEFFICIENT OF PERFORMANCE (C.O.P.)

6:8.1 The Effect of Condenser Pressure

Results of C.O.P. for the condenser pressures and condenser systems tested are presented in Figures 6:23 and 6:24.

Analysis of the curves with respect to the error bar shows that at the 30 minute interval for the 800 litre loading, C.O.P. ranges from 3.25 for the 6.5 condenser pressures to 2.50 for the 12 bar condenser pressure. This difference remained significant throughout the run but with the range decreasing by 0.25 at the completion of cooling.

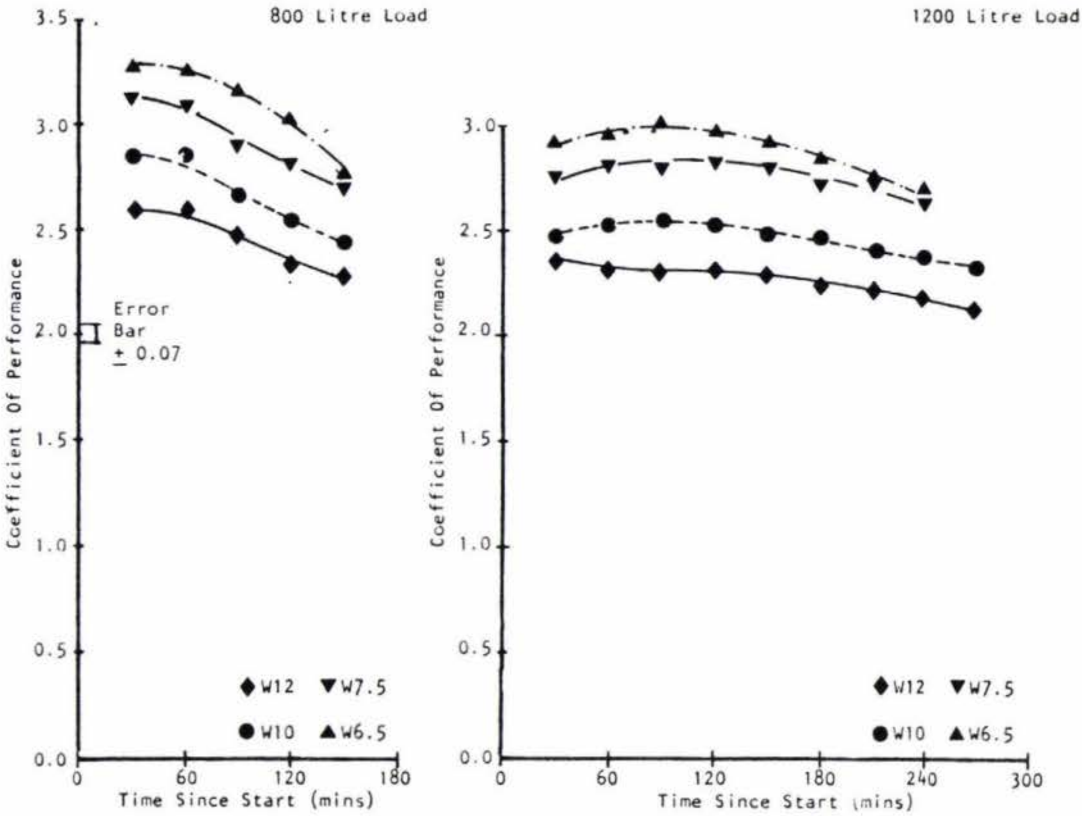


FIGURE 6:23

The effect of condenser pressure on COP - water system

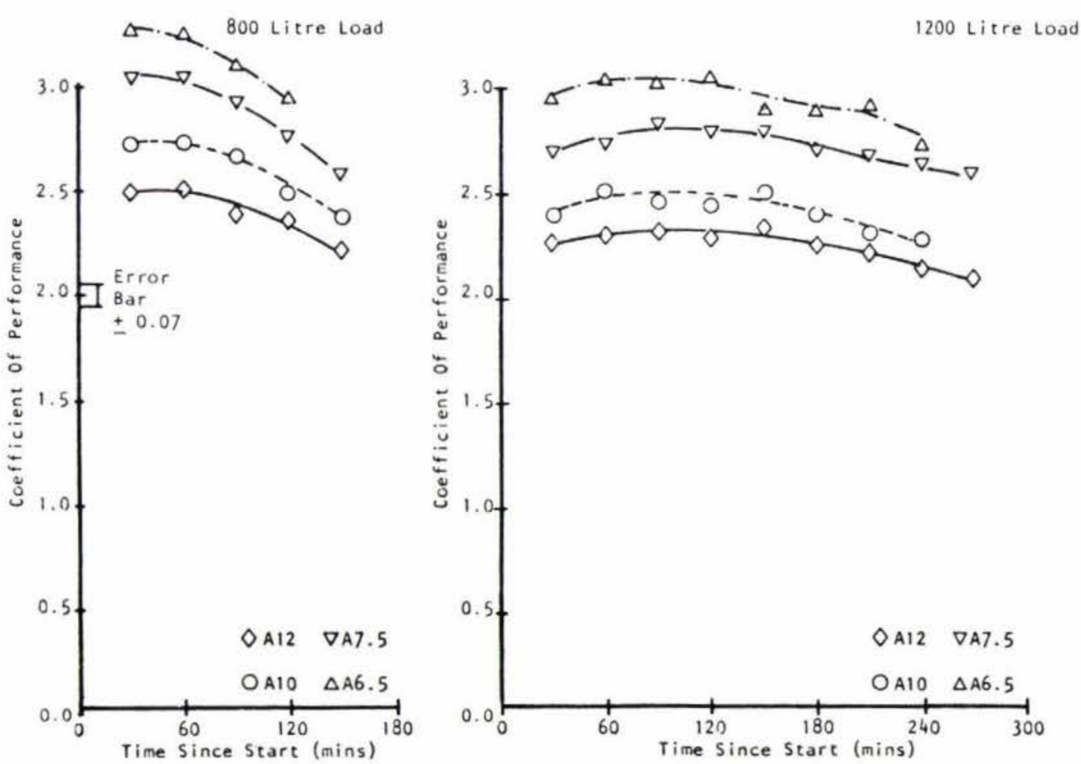


FIGURE 6:24

The effect of condenser pressure on COP - air system

For the 1200 litre loading the effect of condenser pressure was the same, but with a slightly lower range of 2.30 to 3.00 at the 90 minute time interval.

Plotting the average C.O.P. for each run against condenser pressure (Figure 6:25) shows more clearly the decrease in C.O.P. with increasing condenser pressure.

6:8.2 The Effect of Condenser System

The calculation of C.O.P. in Section 6:8.1 used the instantaneous power consumption of the compressor. A comparison of Figures 6:23 and 6:24 shows that there is no significant difference due to the condenser system used. However, when the power consumption for the fans (0.36 kW) was included in the calculations for the air cooled system, the value of C.O.P. was significantly reduced by approximately 0.4 for the 800 litre load and 0.3 for the 1200 litre load. The range of results for C.O.P. of 2.0 to 2.7 was slightly above the range of 1.8 to 2.2 obtained by Carter and Fisher (1982).

6:8.3 The Effect of the Primary Heat Exchanger

The results of C.O.P. for primary versus no primary are presented in Figure 6:26.

The results show that inclusion of the primary heat exchanger reduced the C.O.P. slightly for the 10 bar and 12 bar comparisons due to the slight increase in instantaneous power consumption (Section 6:7.3). The difference in the 7.5 bar condenser pressure readings was somewhat larger than for either the 10 bar or 12 bar condenser pressure readings. However, most of the extra difference was attributed to the slight differences in initial milk inlet temperatures for the two runs.

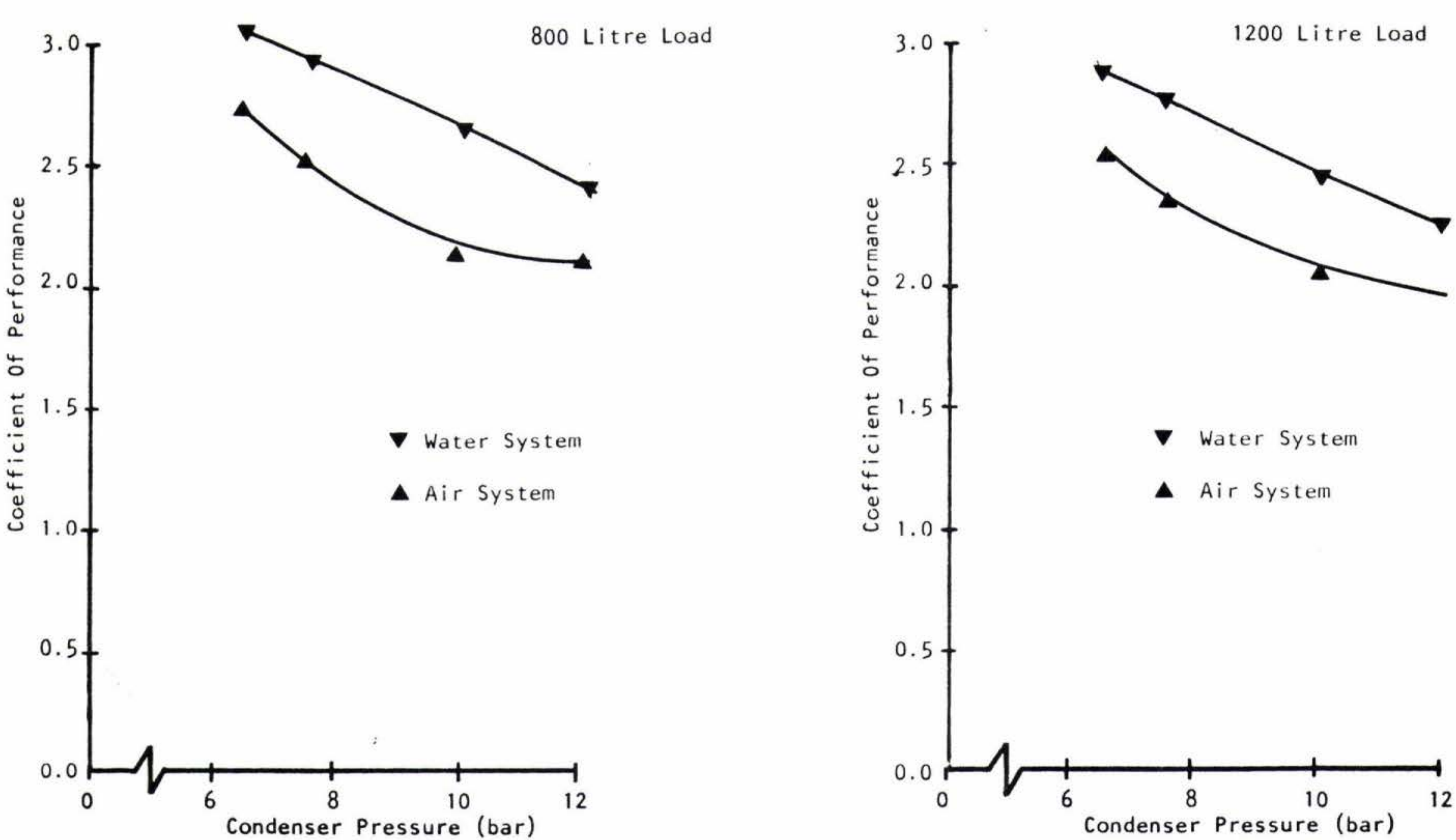


FIGURE 6:25

The Relationship between average COP and condenser pressure

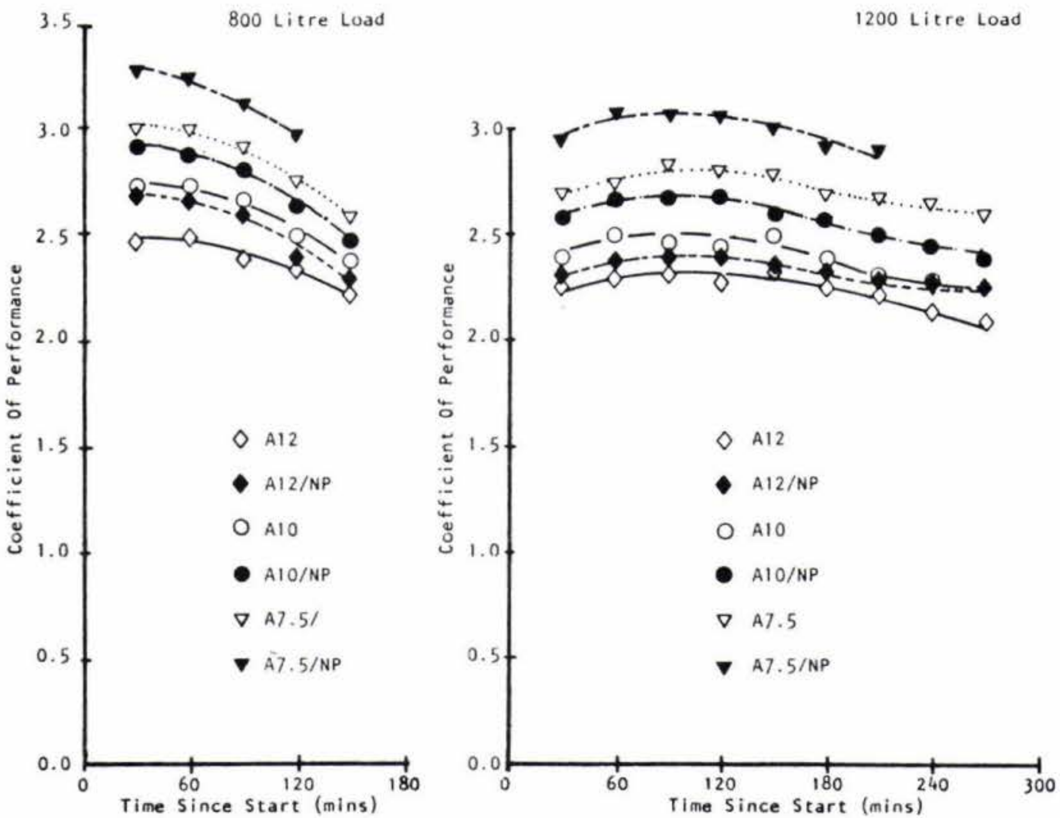


FIGURE 6:26

The effect of the inclusion of the primary heat exchanger on COP

6:8.4 The Effect of Changing Milk Inlet and Final Temperatures

Decreasing milk inlet temperature by $5^{o}C$ reduced C.O.P. slightly by 0.12, on average, for both the 800 and 1200 litre loadings. The effect of increasing milk final temperature was not significant.

6:8.5 The Effect of Suction Superheating

C.O.P. was reduced by 0.1 when the suction superheater was tested. This decrease was expected as a result of the reduction in R.E. (average of 1.34 $kJ.kg^{-1}$), refrigerant flow rate (average 0.086 $l.min^{-1}$) and the increase in compressor instantaneous power consumption (average 0.0[illegible] kW).

6:8.6 Conclusions on C.O.P.

The increasing of condenser pressure had the greatest effect on C.O.P., with a decrease of 0.25 being recorded for a condenser pressure change of 5.5 bars (6.5 bars to 12 bars). This decrease is reflected in the reduced cooling rates and increased cooling times discussed in Section 6:2.

The significant changes in C.O.P. in response to changing condenser pressures were the result of the combined changes in refrigerant flow (M_r), refrigeration effect (R.E.) and instantaneous compressor power consumption (I.C.P.C.), as summarised in Sections 6:4, 6:5 and 6:7.

Excluding the power consumption required by the fans, there were no significant differences between condenser systems. The inclusion of fan power requirements reduced the C.O.P. for the air cooled system by between 0.3 and 0.4.

There was no significant effect on C.O.P. due to the inclusion of the suction superheater or the primary heat exchanger.

A study of the C.O.P. equation where

$$COP = \frac{RE \times M_r}{ICPC} \qquad \text{.....Eqn 4:11}$$

shows that C.O.P. is proportional to Refrigeration Effect and refrigerant flow. The discussion in Section 6:5.1 showed that the product of R.E. and M_r was the cooling rate (Eqn 6:1) and the discussion in Section 6:2 found that cooling times were directly related to cooling rates. It can be concluded, therefore, that the value of C.O.P. is closely related to cooling time. As a consequence, a decrease in C.O.P., other than for an increase in instantaneous compressor power consumption, indicates an increase in cooling time. The extreme situation occurs when cooling time exceeds the time specified in the regulations. For the system tested, the regulation cooling time was exceeded by a maximum of 17 minutes (12 bar condenser pressure), for which the corresponding C.O.P. was 2.25 for the water cooled system and 1.95 for the air cooled system. The regulation cooling time (180 minutes) was met when C.O.P. was 2.45 and 2.05 for the water and air cooled systems respectively.

The value of C.O.P. also represents a combination of all the factors affecting refrigeration performance, in that it expresses the efficiency with which the refrigeration system utilizes the energy inputs to cool the milk.

6:9 PRIMARY HEAT EXCHANGER MODEL

Owing to the difficulties in controlling primary water flow rates for the water cooled condenser system, a mathematical model was developed to calculate the water outlet temperatures and heat recovery rates, under constant water flow conditions. The results obtained allowed a comparison to be made with the results from the air cooled system.

The model was developed from results of overall thermal conductance (U) values obtained from tests on the full size primary heat exchanger. These results were plotted using the Wilson plot technique from which relationships between U, water flow rates, refrigerant flow rates and heat exchanger temperatures were established.

The results for U, the relationships and the computer programme developed are presented in Appendix A6:2. Results of heat recovery rate and water outlet temperature, for the water cooled system in subsequent sections, have been corrected by the model for a water flow rate of 0.625 $l.min^{-1}$.

6:10 HEAT RECOVERY

6:10.1 The Effect of Condenser Pressure

Results of average heat recovery rates for the condenser pressures tested are presented in Figure 6:27 for the water cooled system and Figure 6:28 for the air cooled system.

The effect of increasing condenser pressure was to significantly increase the rate of heat recovery by 0.9 kW. This was expected since increasing condenser pressures resulted in increasing head pressures and associated higher vapour temperatures (Section 2:3.2.1).

6:10.2 The Effect of Condenser System

A comparison of Figures 6:27 and 6:28 shows that the water system had a significantly greater rate of heat recovery than the air system.

The differences between the condenser systems, of 0.75 kW for a 12 bar condenser pressure decreasing to approximately 0.10 kW for a 6.5 bar condenser pressure, can be attributed to the preheating of the water in the secondary water cooled heat exchanger. The degree of preheating rose with increasing condenser pressure and temperature from 0.17 kW, for a condenser pressure of 6.5 bar, to 0.87 kW for a condenser pressure of 12 bars.

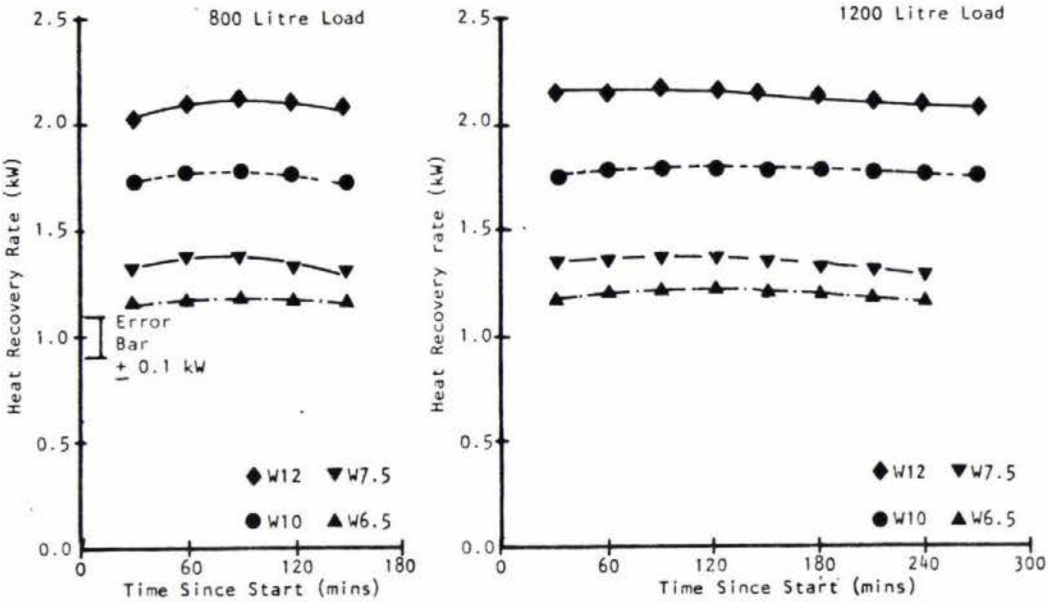


FIGURE 6:27
The effect of condenser pressure on heat recovery - water system

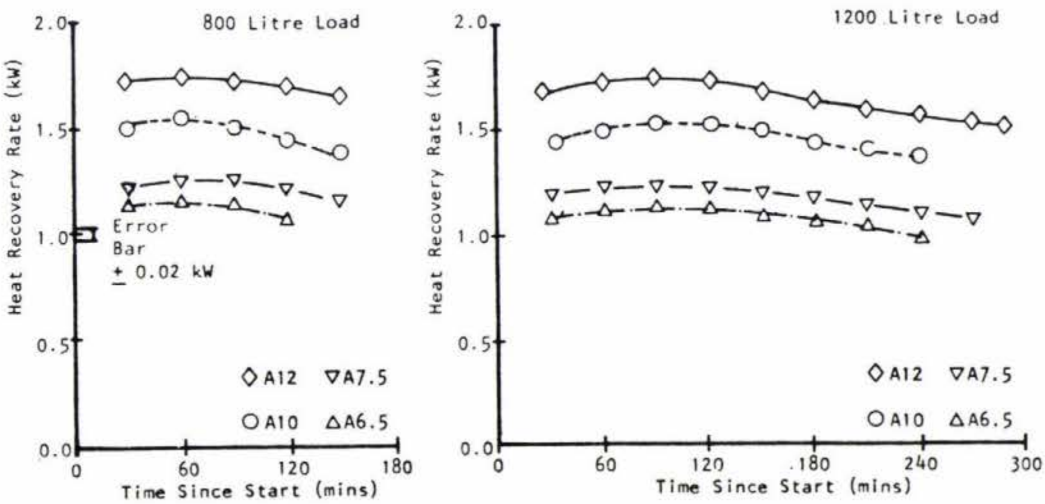


FIGURE 6:28
The effect of condenser pressure on heat recovery - air system

These effects are highlighted in Figure 6:29, in which total heat recovery in terms of $kWh.day^{-1}.m^{-3}$ of milk cooled has been plotted. Presenting the data in this form takes into account the increased cooling time of the higher condenser pressures and allows an estimation of heat recovered to be made for a selected volume of milk.

The effect of environmental loading can also be seen in Figure 6:29, with the broken lines indicating total heat recovery for a zero environmental loading based on data from Table 6:1.

6:10.3 The Effect of Changing Milk Inlet and Final Temperature

Results of total heat recovery for changing milk inlet and final temperatures are presented in Figures 6:30 and 6:31.

Reducing milk inlet temperatures by $5^{o}C$ or increasing milk final temperatures by $3^{o}C$ significantly decreased the total heat recovered by 25% and 20%, as a result of the smaller cooling differentials. Combining the two effects reduced the total heat recovered by 41%.

Heat recovery for the lower temperature differentials as a percentage of the heat recovered for the runs with a $23^{o}C$ - $4^{o}C$ differential are compared in Table 6:6 with the theoretical percentage differences (Defined in Section 6:2.2). The results show that heat recovery was dependent on milk temperature differential due, to the associated shorter cooling times.

6:10.4 The Effect of Suction Superheating

Suction superheating significantly increased the rate of heat recovery by 0.15 kW (8%), on average. This difference was due to the increase of $8^{o}C$ in the temperature of the delivery superheated vapour entering the primary heat exchanger.

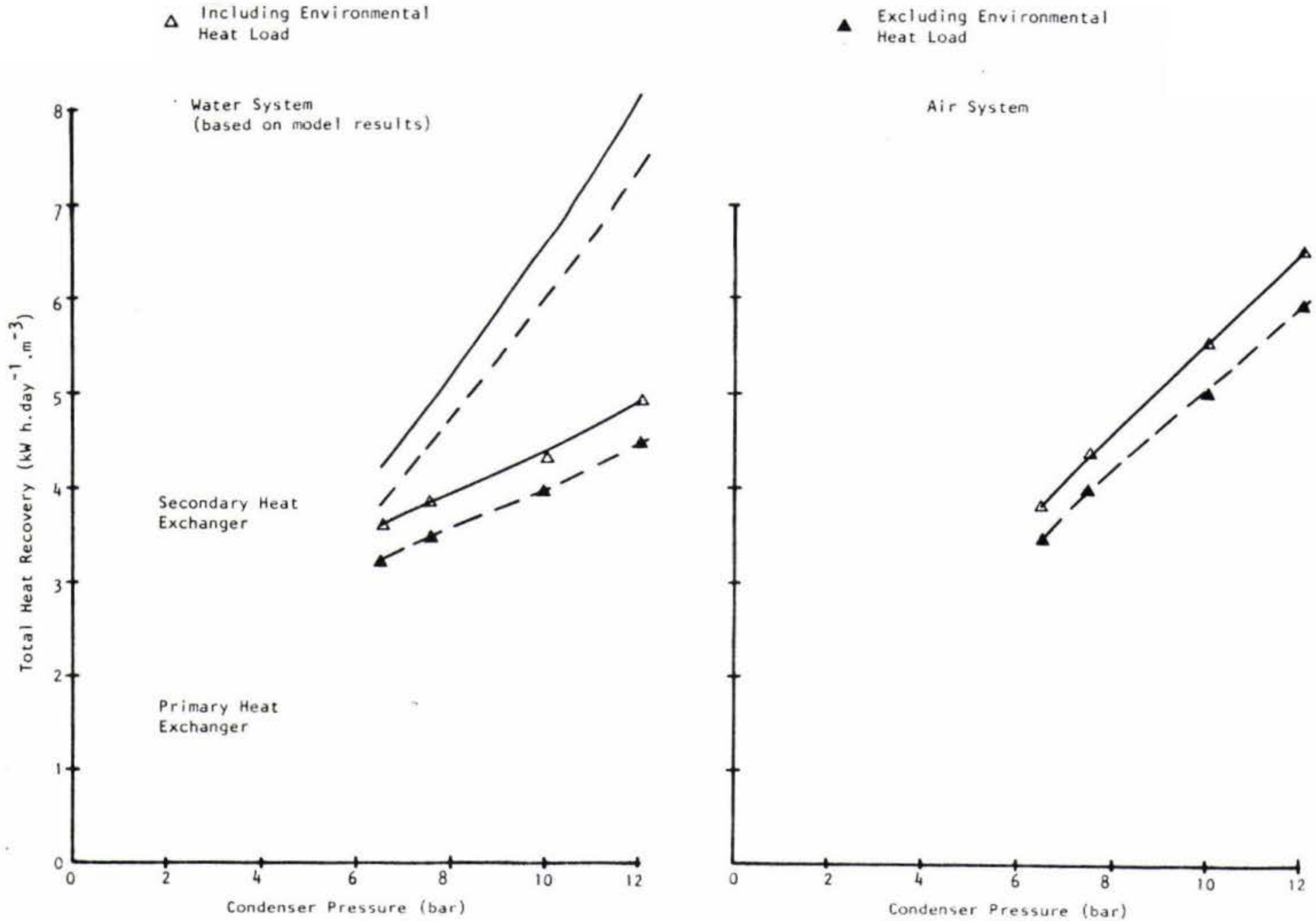


FIGURE 6:29

The effect of condenser pressure on total heat recovery

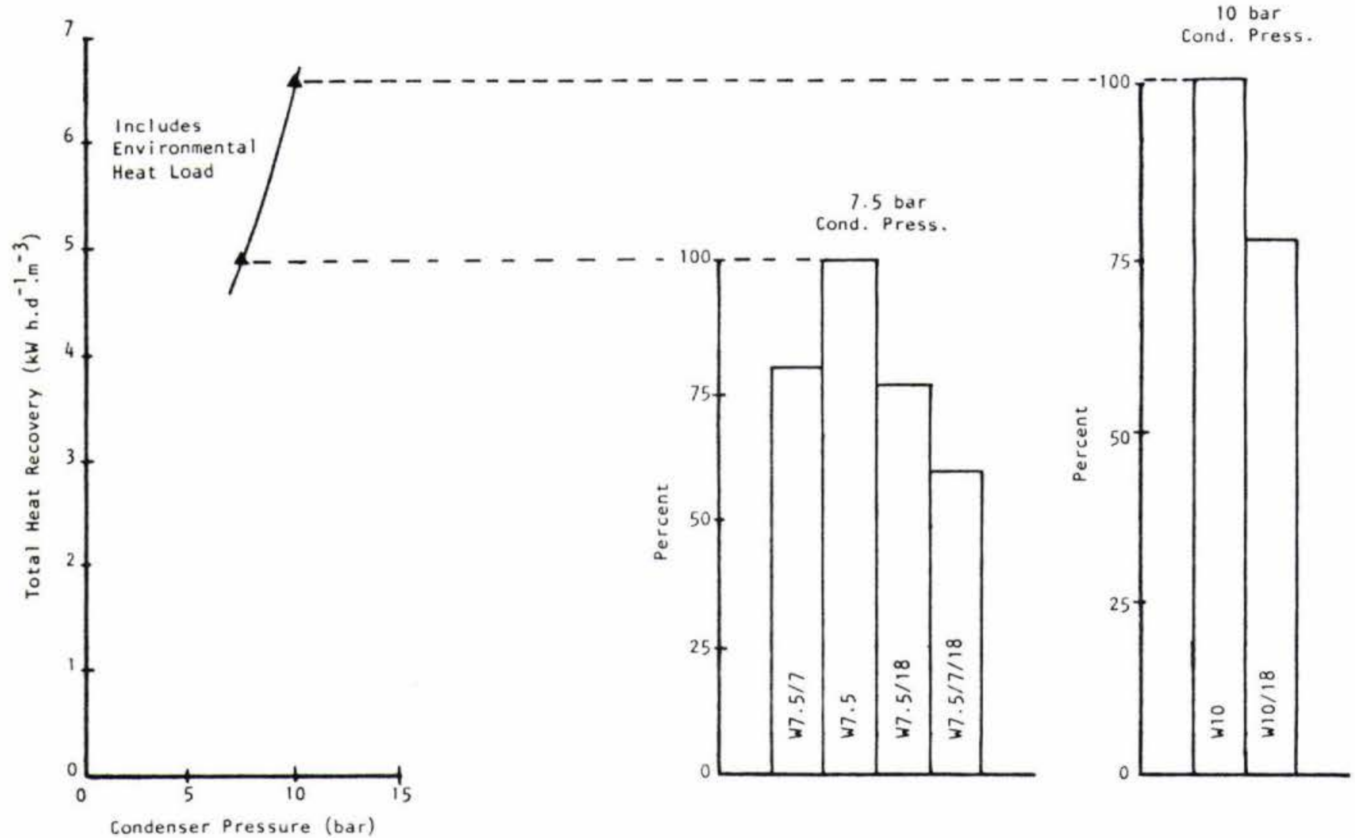


FIGURE 6:30

The effect of milk inlet and final temperatures on heat recovery - water system

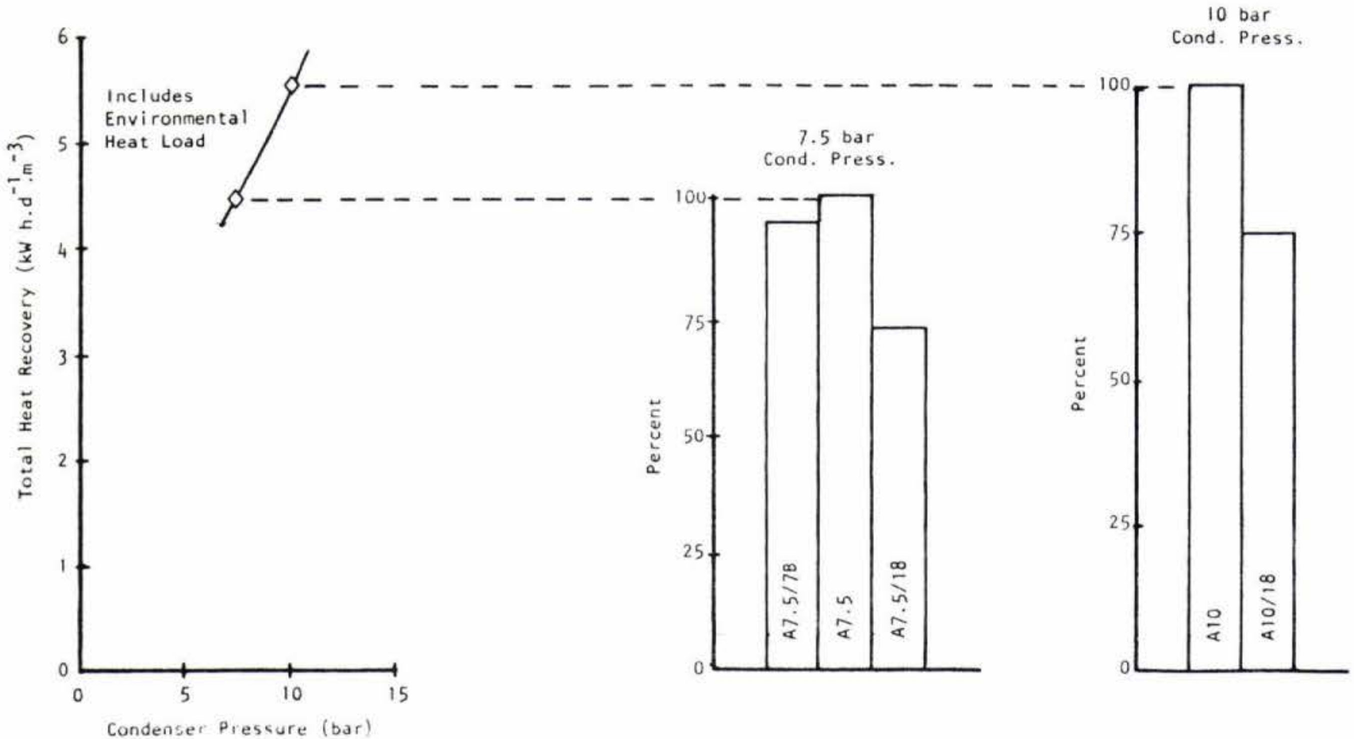


FIGURE 6:31

The effect of milk inlet temperature and receiver pressure on heat recovery - air system

TABLE 6:6

Experimental and theoretical heat recovery totals as percentages of the heat recovery for a $23^{o}C$ to $4^{o}C$ milk cooling differential (Error = $\pm$ 6%)

Run	$18^{o}C - 4^{o}C$		$23^{o}C - 7^{o}C$		$18^{o}C - 7^{o}C$	
	Exp.	Theo.	Exp.	Theo.	Exp.	Theo.
W7.5	76	74	80	84	59	58
A7.5	72	74				
W10	77	74				
A10	75	74				

Integrating the heat recovery rate over time (Tables 6:1 and 6:2) resulted in 7.42 kWh of energy per day per cubic metre of milk cooled being recovered. This value was 1.0 $kWh.day^{-1}.m^{-3}$ (15%) greater than the value for the W10 run.

6:10.5 Conclusions of Heat Recovery

Increasing condenser pressure significantly increased the quantity of heat recovered, by 3.9 $kWh.day^{-1}.m^{-3}$ of milk cooled, for the water cooled system, and 2.8 $kWh.day^{-1}.m^{-3}$ for the air cooled system.

Reducing milk inlet temperatures by 5^{o}C (26%) significantly reduced heat recovery, by an average of 25%, while increasing milk final temperature by 3^{o}C (16%) significantly reduced heat recovery, by 16%. The combined effect reduced heat recovery by 41%.

Suction superheating significantly improved heat recovery by 1.0 $kWh.day^{-1}.m^{-3}$.

6:11 PRIMARY HEAT EXCHANGER WATER OUTLET TEMPERATURE

6:11.1 The Effect of Condenser Pressure and Condenser System

Since water outlet temperature is proportional to heat flow, for a constant water flow rate, curves for water temperature versus time have the same shape as those in Figures 6:27 and 6:28. Results of average water outlet temperature for each of the condenser pressures tested are presented in Figure 6:32.

Increasing condenser pressure from 6.5 bar to 12 bar resulted in a significant increase in primary water outlet temperature, from 45^{o}C to 64^{o}C for the water cooled system, and from 38^{o}C to 55^{o}C for the air cooled system.

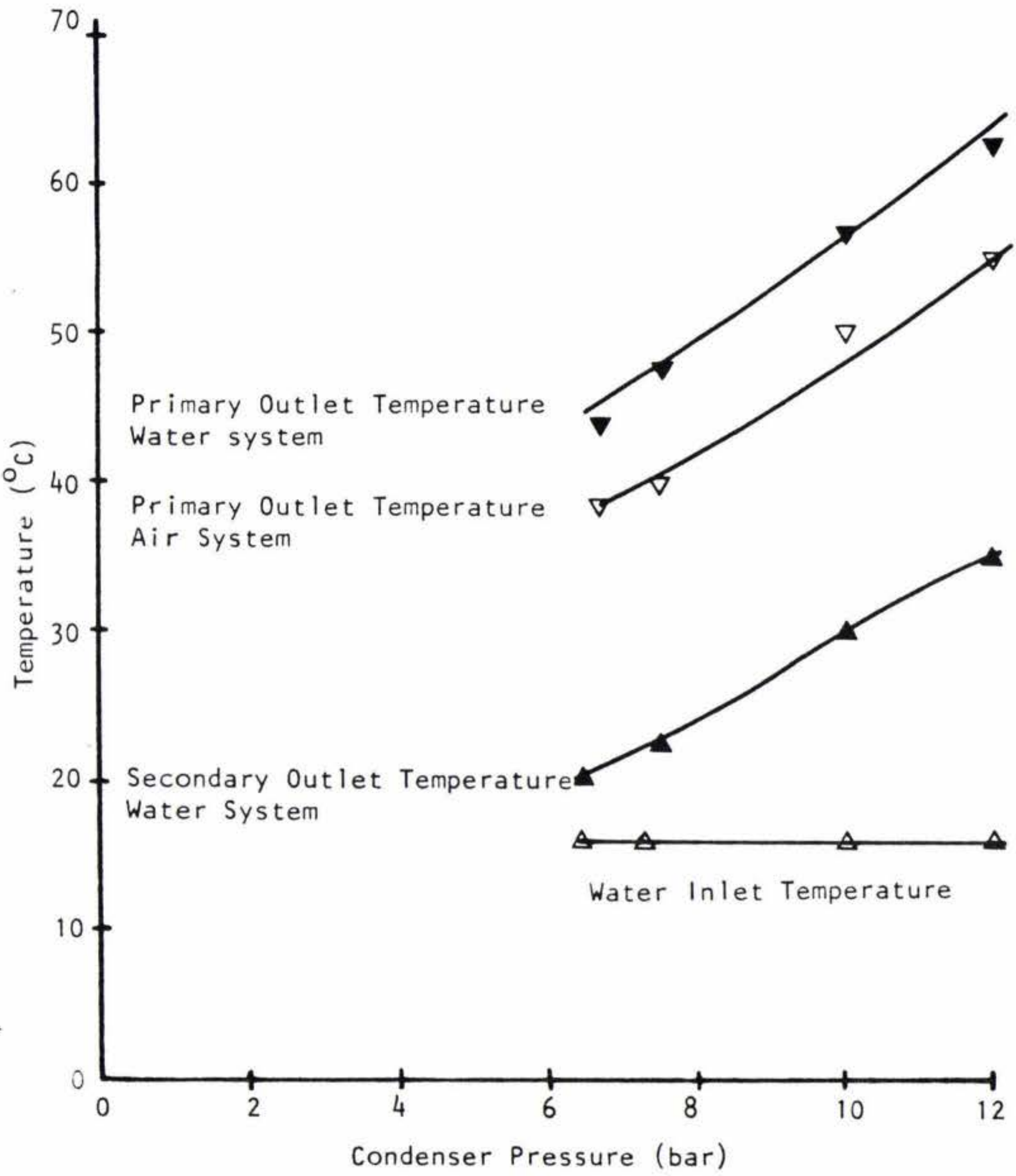


FIGURE 6:32

The effect of condenser pressure on water outlet temperatures

Figure 6:32 also shows the effect of preheating the water in the secondary heat exchanger of the water cooled system. The effect of preheating the water in the secondary water cooled heat exchanger from 20°C to 30°C was to produce water at a significantly higher temperature than the air cooled system. In particular, the 12 bar water cooled condenser system produced water with a temperature which met the requirements of a number of the cleaning systems presented in Table 2:2.

The final temperature differential of 9°C between the two condenser systems was less than the preheating temperature rise for the water cooled condenser pressures above 7.5 bar. This was due to a higher temperature gradient operating in the primary for the air cooled system compared to that of the water cooled system, as a result of the difference in inlet temperatures.

The maximum water outlet temperatures attained (55°C and 64°C) by the two systems were within the range of 33°C to 65°C discussed in the literature (Section 2:5).

6:11.2 The Effect of Suction Superheating

The effect of suction superheating was to significantly increase the primary water outlet temperature by 5°C, due to the 8°C rise in delivery superheated vapour entering the primary heat exchanger.

6:11.3 Conclusions on Primary Water Outlet Temperature

Increasing condenser pressure significantly increased water outlet temperatures, by 20°C for the water cooled system and 16°C for the air cooled system. The effect of preheating the water in the secondary water cooled heat exchanger was to produce water at a higher temperature than for the air cooled system (45°C to 64°C, c.f. 38°C to 55°C).

Inclusion of the suction superheater significantly increased water outlet temperature by $5^{o}C$.

6:12 OVERALL THERMAL CONDUCTANCE OF THE PRIMARY HEAT EXCHANGER

The overall thermal conductance values (U) for the primary heat exchanger, based on the refrigerant area, were constant at 0.056 ± 0.004 $kW.m^{-2}.^{o}C^{-1}$. This value was lower than the value of 0.109 $kW.m^{-2}.^{o}C^{-1}$ determined from the test results (Section 5:3) and can not be fully explained. However, problems with liquid polyurethane accidently being injected into one of refrigerant tubes during manufacture was a possible cause. Although this was later burnt out, polyurethane residuals and carbon probably remained on the fins reducing their heat transfer characteristics. Analysis of the headloss data (Section 6:6) does not support the hypothesis of a complete or partial blockage of the refrigerant tubes.

A comparison of these figures with the values in Section 2:5.1 shows that this design had values of U 3.32 times greater than those for a tube-in-tube heat exchanger, and 3.77 times greater than for a coil-in-tank heat exchanger. This suggests that increasing refrigerant vapour heat transfer area, through the addition of extended surfaces (fins), significantly improves the heat transfer characteristics of the primary heat exchanger.

6:13 ANALYSIS OF A COMBINED HEAT RECOVERY-REFRIGERATION SYSTEM

The analysis of the heat recovery system is in three stages. Firstly, the overall performance of the heat recovery system as it affects the refrigeration system will be discussed. Secondly, the impact of the combined heat recovery refrigeration system on the utilization of electrical energy input, for milk cooling and water heating, will be analysed. Finally, the operating conditions which most efficiently utilize the energy used for milk cooling and water heating will be selected.

6:13.1 Performance of the Heat Recovery System in Relation to Refrigeration

The evaluation of heat recovery gains, relative to the cost of heat recovery in terms of the additional power required by the compressor, can be made using Eqn A2:1, i.e.,

$$Ke = \frac{Q_{CH} - Q_C}{Q_{HR}} \qquad \text{.....Eqn 6:4}$$

where Ke = ratio of the additional compressor energy requirements to the energy recovered as hot water

Q_{CH} = compressor energy consumption with heat recovery (kWh)

Q_C = compressor energy consumption at a condenser pressure of 6.5 bar and no heat recovery (kWh)

Q_{HR} = heat recovery energy (kWh)

As only the effect of heat recovery relative to refrigeration is being analysed, then Ke is a suitable index, since it measures the additional energy required by the compressor in recovering heat energy, i.e., the value of Ke gives no indication of the total energy used by the refrigeration system. It is apparent from Eqn 6:4 that the larger the value of Ke the smaller the returns from heat recovery, i.e., as Ke approaches unity the increased compressor energy increasingly offsets any gains made from heat recovery.

If the price of energy for water heating and refrigeration are the same (i.e., a pricing ratio of 1:1) then the performance ratio is as for Eqn 6:4. However, on many farms, the cost of water heating is less (two-thirds) than the cost of energy for milk cooling (i.e., a pricing ratio of 1:3). In terms of calculating the performance ratio, the energy value of the additional compressor power consumption must be multiplied by 3 to give an assessment of the economic relationship between additional compressor energy consumption and heat recovery energy. Under these conditons the relationship is established by Ke',which is equal to 3Ke.

Plots of Ke and Ke' (calculated from data in Figures 6:22 and 6:29) are presented in Figure 6:33 for both the air and water cooled systems.

The results show that for a condenser pressure of 6.5 bar, and a pricing ratio of 1:1, only 5% of the heat recovery was required to compensate for the additional compressor power consumption (Ke = 0.05). These values increased with increasing condenser pressure such that for a 12 bar condenser pressure Ke for the air cooled system was 0.34, while for the water cooled system Ke was equal to 0.26.

This difference was due to the higher heat recovery rate for the water cooled system, as discussed in Section 6:10.1.

Increasing the pricing ratio increased the economic value of the additional compressor power consumption 3 fold,to give a Ke of 0.16 for the 6.5 bar condenser pressure. For a condenser pressure of 12 bar the air cooled system was not viable (Ke' = 1.01) since the cost of the additional energy required by the compressor was greater than the savings from heat recovery. The corresponding value for the water cooled system was 0.79.

Results for gross heat recovery, additional compressor energy and nett heat recovery (gross heat recovery minus additional compressor energy) , when cooling milk from $23^{o}C$ to $4^{o}C$ at various condenser pressures, are presented in Figure 6:34 for a pricing ratio of 1:1 and Figure 6:35 for a pricing ratio of 1:3.

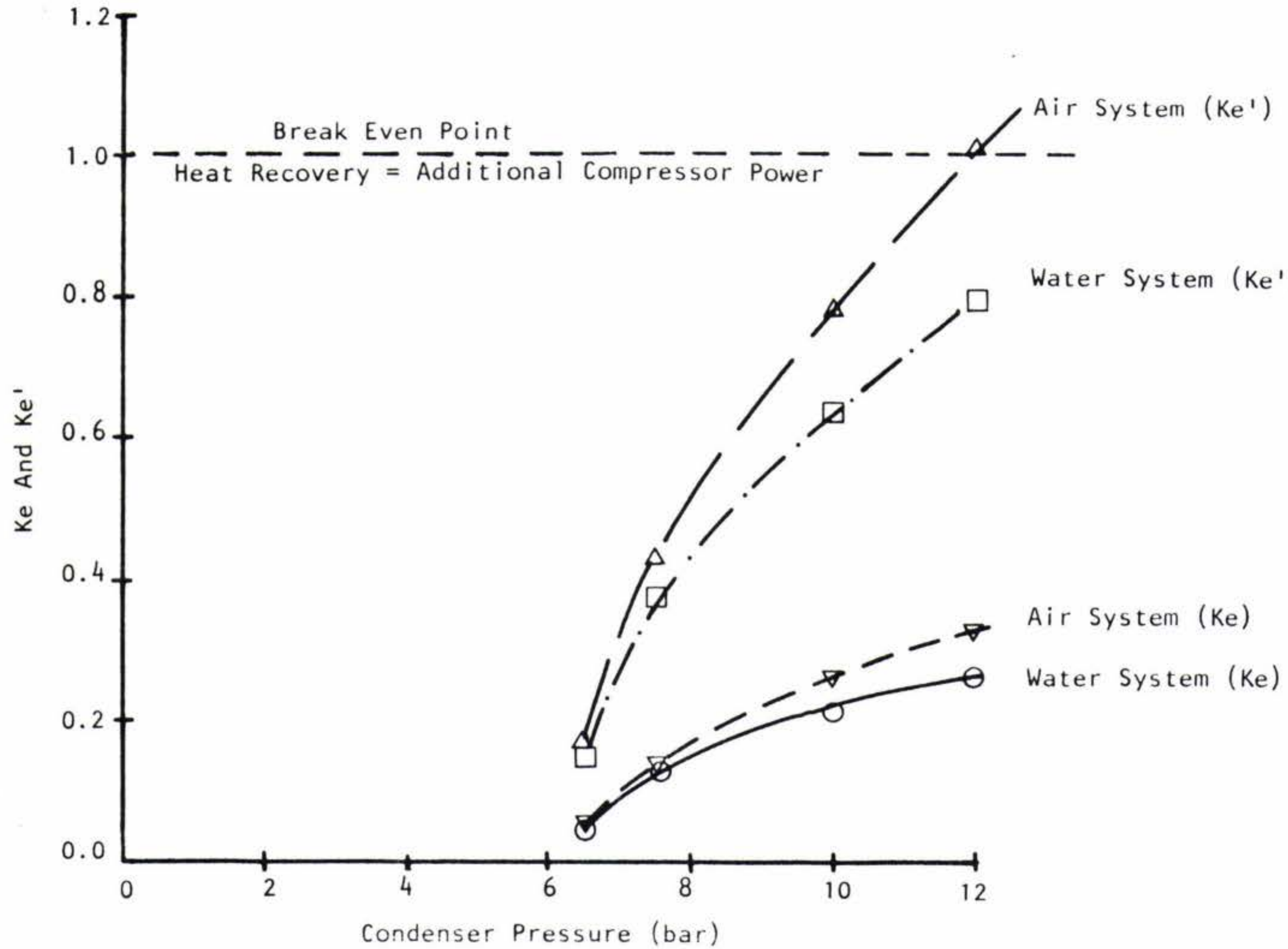


FIGURE 6:33

The effect of condenser pressure on the ratio of extra compressor power consumption and heat recovery

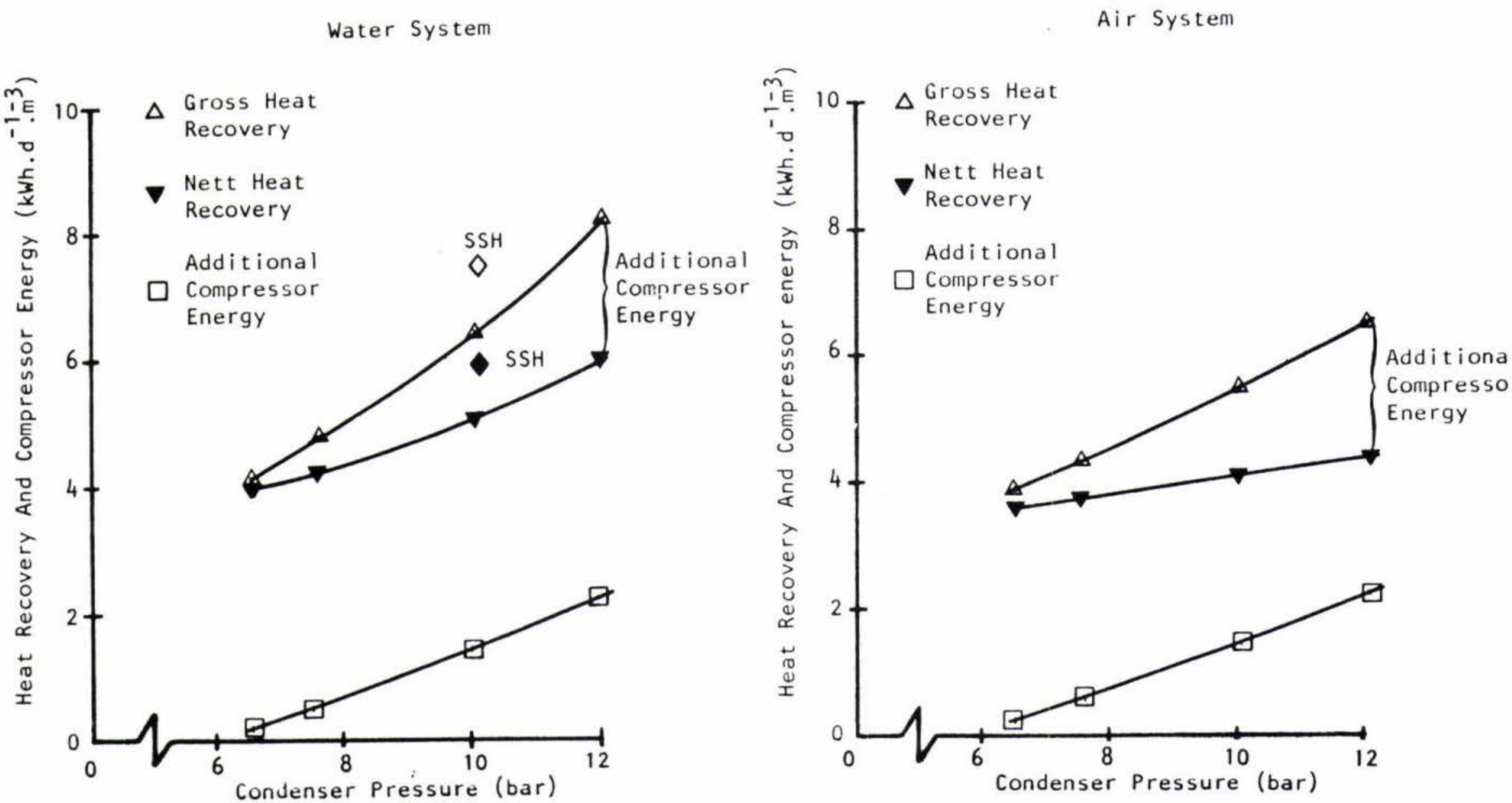


FIGURE 6:34

The relationship between heat recovery, additional compressor energy and condenser pressure - 1:1 pricing ratio

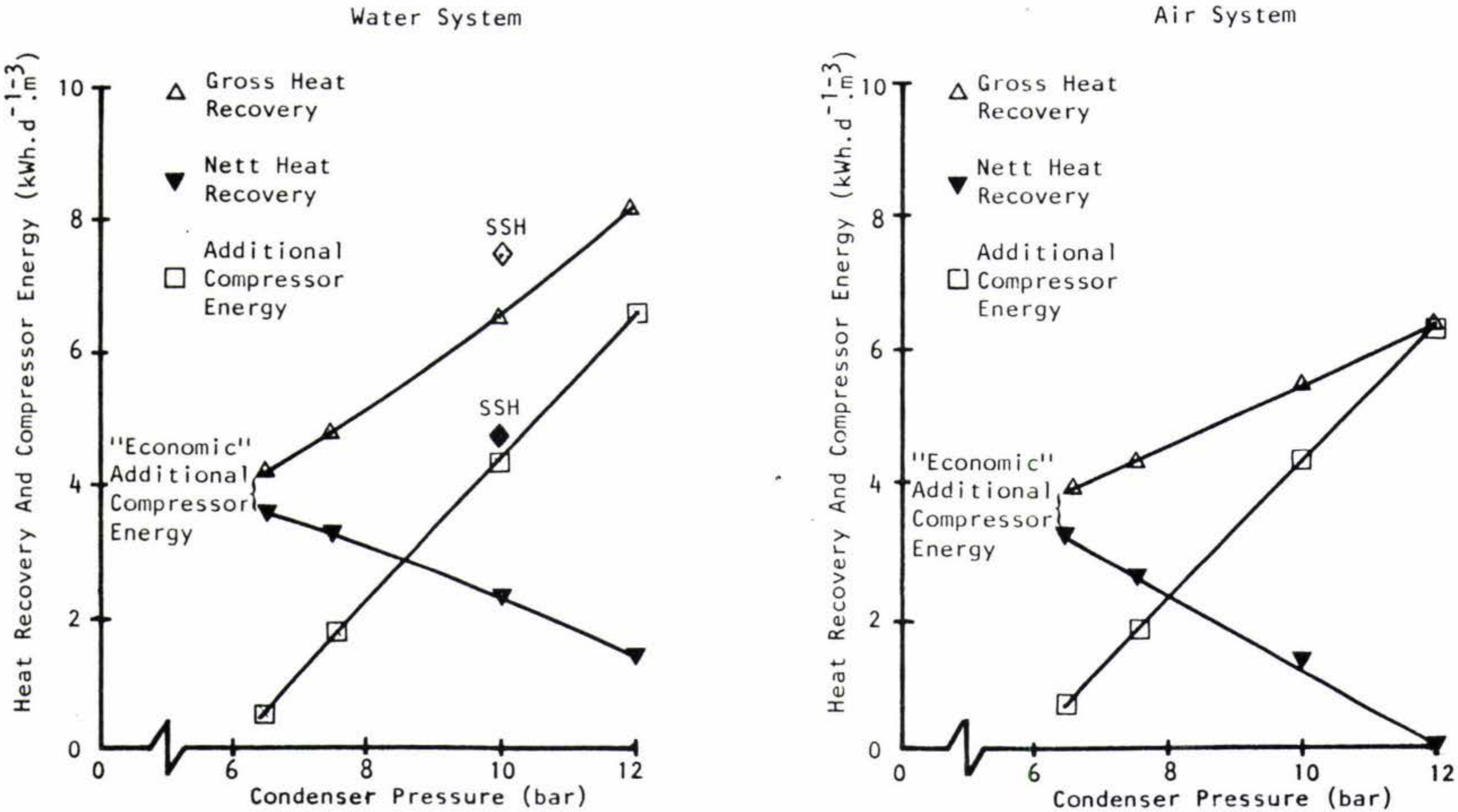


FIGURE 6:35

The relationship between heat recovery, additional compressor energy and condenser pressure - 1:3 pricing ratio

Increasing condenser pressure resulted in an increase in nett heat recovery for a pricing ratio of 1:1 (Figure 6:34) with values for the water cooled system of 3.88 $kWh.day^{-1}.m^{-3}$ at a condenser pressure of 6.5 bar and 6 $kWh.day^{-1}.m^{-3}$ for the 12 bar condenser pressure. The corresponding range of values for the air cooled system was from 3.5 $kWh.day^{-1}.m^{-3}$ to 4.3 $kWh.day^{-1}.m^{-3}$.

The effect of increasing the pricing ratio to 1:3 reversed the trend obtained for nett heat recovery using a 1:1 pricing ratio. For the water cooled system, nett heat recovery was 1.8 $kWh.day^{-1}.m^{-3}$ for the 12 bar condenser pressure, rising to 3.5 $kWh.day^{-1}.m^{-3}$ for the 6.5 bar condenser pressure. The corresponding range for the air cooled system was -0.1 $kWh.day^{-1}.m^{-3}$ to 3.3 $kWh.day^{-1}.m^{-3}$.

Inclusion of the suction superheater increased nett heat recovery significantly by 17% for the 1:1 pricing ratio and by 20% for the 1:3 pricing ratio.

The curves in Figure 6:34 show that, even with increasing Ke, nett heat recovery increases with increasing condenser pressure. The response to increasing Ke' in Figure 6:35 was a decrease in nett heat recovery with increasing condenser pressure. Analysis of the slopes of the curves shows that:-

If the slope of the curves in Figure 6:33 is such that

$$\frac{dKe}{dCp} \text{ or } \frac{dKe'}{dCp} > 0 \qquad \text{i.e., } \frac{da}{dCp} > \frac{dg}{dCp}$$

where Cp = condenser pressure

a = additional compressor energy

g = gross heat recovery energy

then for n = nett heat recovery,

1) $\frac{dn}{dCp} > 0$ when $\frac{da}{dCp} < 2 \times \frac{dg}{dCp}$

2) $\frac{dn}{dCp} = 0$ when $\frac{da}{dCp} = 2 \times \frac{dg}{dCp}$

3) $\frac{dn}{dCp} < 0$ when $\frac{da}{dCp} > 2 \times \frac{dg}{dCp}$

In Figure 6:34 $\frac{dn}{dCp} > 0$ since additional compressor energy does not increase twice as fast as gross heat recovery. The opposite is true in Figure 6:35, i.e., $\frac{dn}{dCp} < 0$, due to the effect of the increase in pricing ratio from 1:1 to 1:3.

Analysis of Ke and Ke' only evaluates the relationship between heat recovery and compressor energy requirements. However, the relationship between the quantity of heat recovered and the quantity of heat available from the milk can be estimated by determining the heat recovery efficiency (Section 2:5.2). For this system, heat recovery efficiency ranged from 17% for the air cooled system operating at a condenser pressure of 6.5 bars to 36% for the water cooled system operating at a condenser pressure of 12 bars. This range is similar to that quoted in the literature for desuperheater systems (24% to 35%).

Therefore, it can be concluded that, for a pricing ratio of 1:1, maximum efficiency of the system in terms of nett heat recovery occurs at a condenser pressure of 12 bars, but changing the pricing ratio to 1:3 results in maximum nett heat recovery being obtained at a condenser pressure of 6.5 bars. However, the overall effect on total energy requirements for water heating and milk cooling can not be determined from the analysis above.

6:13.2 Combined Water Heating and Refrigeration Heat Recovery System

The efficiency of the combined refrigeration-heat recovery and water heating system was calculated from the total energy required for the heat recovery system (refrigeration, heat recovery and additional water heating) as a percentage of the total energy requirements of the standard system (heating water to a specified temperature and cooling milk with a refrigeration system operating at a condenser pressure of 6.5 bars and without a primary heat exchanger).

Results of these calculations for cooling 2000 litres of milk from 23°C and 18°C to 4°C, and heating 300 litres of water to 65°C and 95°C from 15°C, are presented in Figure 6:36 for a pricing ratio of 1:1 and Figure 6:37 for a pricing ratio of 1:3.

For the water cooled system at a 1:1 pricing, the results (Figure 6:36) show that, for cooling milk from 23°C to 4°C and heating water to 95°C, the heat recovery system was 18% to 25% (27% for suction superheater) more efficient in energy utilization than the standard system, whereas the air cooled system, operating under the same conditions, was constant at 16%, due to the power consumed by the fans and the lower heat recovery rates compared to those for the water cooled system.

Reducing milk inlet temperature by 5°C reduced the efficiency of the heat recovery system over the 6.5 bar to 12 bar condenser pressure range by an average 4% for the water cooled system and 3% for the air cooled system. This was due to the smaller reduction in heat recovery savings and refrigeration energy costs compared to the standard system, since the reduction in heat recovery had to be compensated for by the increase in additional water heating energy. This was not the case for the standard system, since refrigeration and water heating are independent , i.e., a

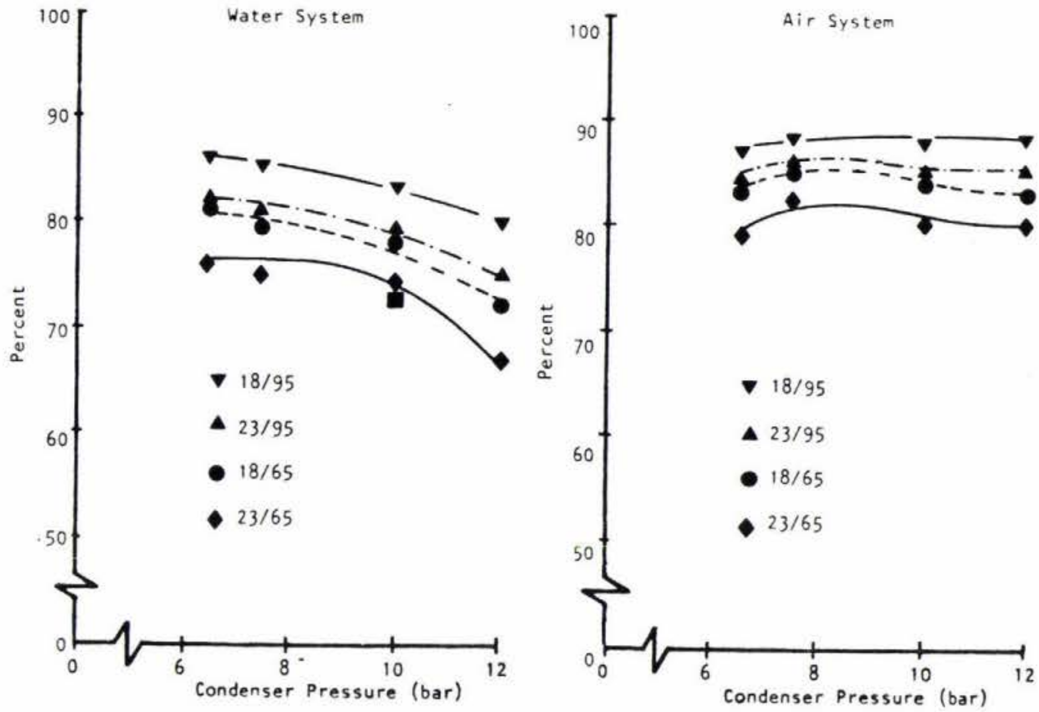


FIGURE 6:36

Total refrigeration and water heating energy as a percentage of the energy required by the standard system - 1:1 pricing ratio

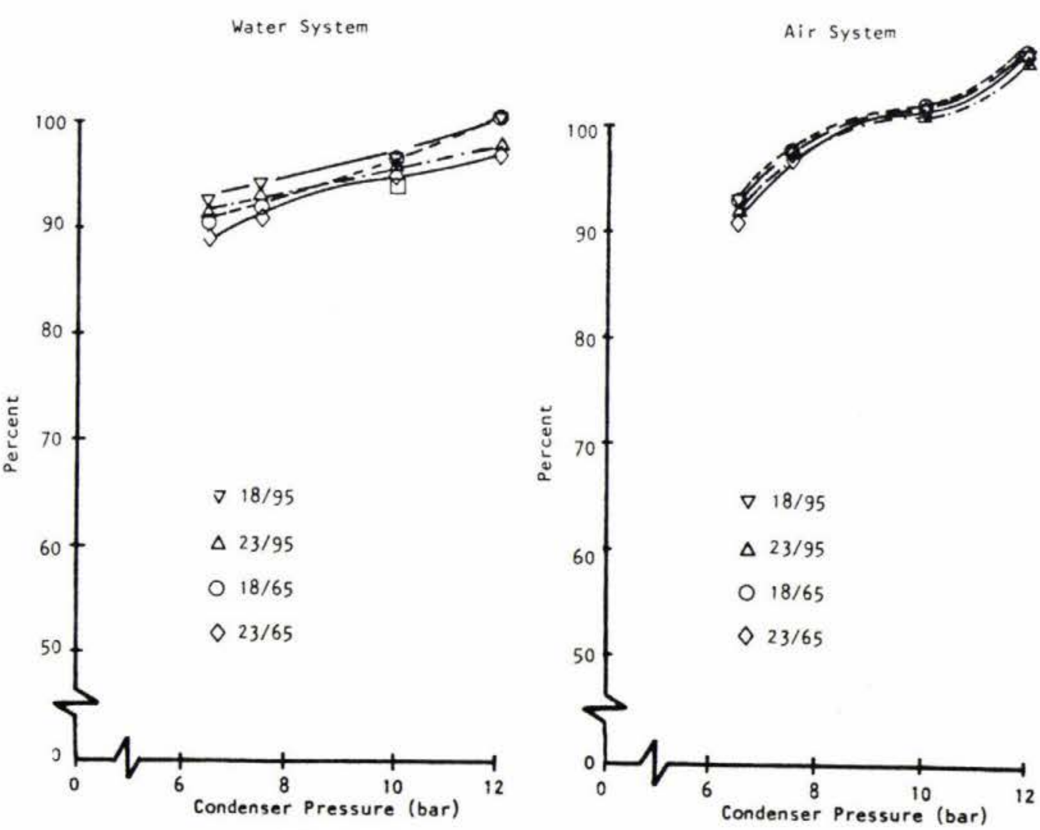


FIGURE 6:37

Total refrigeration and water heating energy as a percentage of the energy required by the standard system - 1:3 pricing ratio reduction in refrigeration energy consumption does not have to be compensated for by an increase in water heating energy.

Decreasing final water temperature from 95°C to 65°C increased the efficiency of the heat recovery system by 6%, on average, for a 23°C milk inlet temperature and only 2% for the 18°C milk inlet temperature. These increases were the same for both the air and water cooled systems.

The suction superheater, operating at a condenser pressure of 10 bars, proved to be 6% more efficient than the standard system.

For a pricing ratio of 1:3, the effect of increasing condenser pressure (Figure 6:37) from 6.5 bars to 12 bars was to decrease the efficiency of the heat recovery system from 10% to -1% for the water cooled system, and 8% to -6% for the air cooled system. This means that, for the 12 bar condenser pressure for both systems and the 10 bar condenser pressure for the air system, the cost of milk cooling and water heating with the heat recovery system was greater than for the standard system. The reversal in the efficiency trend in response to changing condenser pressure was due to the 3 fold increase in cost for refrigeration energy compared with the savings from heat recovery.

The effect of reducing milk inlet temperatures or water final temperature for a 1:3 pricing ratio was insignificant since the magnitude of the cost changes for changes in refrigeration energy consumption masked the effect of any changes in heat recovery savings.

6:13.3 Selection of Final Operating Conditions

Based on the results presented in Figures 6:36 and 6:37, it is clear that selection of the final operating conditions which would give the best overall milk cooling water heating system depends on the pricing ratio of controlled to uncontrolled electrical power.

For a 1:1 ratio the system should operate at the highest possible condenser pressures provided that the cooling regulations are met.

For a pricing ratio of 1:3 the system should operate at the minimum condenser pressure that will ensure correct operation of the expansion valve as discussed in Section 2:5.4.1.

These specifications apply to all milk inlet temperatures and water heating temperatures except that any heat recovery system should not be operated under conditions where heat recovery, in terms of the volume of water heated, is surplus to hot water requirements for plant cleaning.

6:14 CONCLUSIONS

The variation in performance of the refrigeration system, as expressed by C.O.P., varied considerably with changing operating conditions. Increasing condenser pressure from 6.5 bars to 12 bars reduced C.O.P. significantly by 0.25 due to the 0.5 kW increase in instantaneous compressor power consumption. In addition, C.O.P. for the water cooled system was significantly higher (0.35) than the air cooled system due to the power required to operate the fans (0.36 kW). Changing milk inlet or final temperature, or the inclusion of the suction superheater, had no significant effect on C.O.P.

The ability of the refrigeration system to meet the cooling regulations was independent of condenser system but was dependent upon condenser pressure and milk inlet and final temperatures. Condenser pressures above 10 bars, with a milk inlet temperature of 23^{o}C and a milk final temperature of 4^{o}C, failed to meet the cooling regulation time of 180 minutes when cooling a 1200 litre load. However, decreasing milk inlet temperature and milk final temperature significantly reduced cooling times.

The maximisation of heat recovery from the condenser circuit in the form of hot water conflicted with the maximisation of refrigeration performance, since increasing condenser pressure increased heat recovery by 3.8 $kWh.day^{-1}.m^{-3}$ for the water cooled system and 2.8 $kWh.day^{-1}.m^{-3}$ for the air cooled system. The difference between the water and air cooled system was due to the heat recovered in the secondary water cooled heat exchanger. Increasing condenser pressure also increased the temperature of the water from the primary heat exchanger by 20^oC (45^oC to 64^oC) for the water cooled system and 16^oC (38^oC to 55^oC) for the air cooled system. The inclusion of a suction superheater increased heat recovery by 1.0 $kWh.day^{-1}.m^{-3}$ and water outlet temperature by 5^oC.

The relationship between the gains from heat recovery and the cost of additional refrigeration power consumption was determined from the calculation of the performance ratio, Ke. This ratio was modified to Ke' when the influence of the energy pricing ratio of 1:3 was included. The results for Ke (i.e., a pricing ratio of 1:1) showed that operating at a condenser pressure of 6.5 bars was the most efficient but nett heat recovery increased with increasing condenser pressure. This was not the case for Ke' (1:3 pricing ratio) since nett heat recovery decreased with increasing condenser pressure.

Increasing the cost rate of refrigeration energy by three times (pricing ratio of 1:3) increased Ke' to the point where, for the air cooled system operating at a condenser pressure of 12 bars, nett heat recovery was less than zero, i.e., Ke' = 1.01.

The evaluation of Ke and Ke' provided some of the information required to establish the most efficient heat recovery water heating system by allowing the estimation of nett heat recovery available for off setting water heating costs. Nett heat recovery calculations showed that the maximum returns from the heat recovery system were to be obtained at a condenser pressure of 12 bars, for a 1:1 pricing ratio, and at a condenser pressure of 6.5 bars, for a 1:3 pricing ratio.

The results of nett heat recovery were combined with the results for refrigeration energy and additional water heating energy and compared, on a percentage basis, with the energy required for a standard system. Results of this calculation showed that at a pricing ratio of 1:1, the heat recovery system became increasingly more efficient, compared with the standard system (18% to 25%), for increasing condenser pressures in the case of the water cooled system, but remained relatively constant at 16% for the air cooled system. Changing the pricing ratio to 1:3 reversed the trend wich the heat recovery system being 10% more efficient than the standard system operating at a condenser pressure of 6.5 bars.

The effect of changing milk inlet temperatures or reducing final water temperatures only changed the result by 1% or 2%. However, the inclusion of the suction superheater increased the efficiency of the heat recovery system by 6% (i.e., up to 31%) at a pricing ratio of 1:1, but had no significant effect at a pricing ratio of 1:3.

The conclusions drawn from these results are:-

1) That overall, the water cooled heat recovery system was the most efficient system for cooling milk and heating water.

2) For both water and air cooled systems, the operating conditions for maximum utilization of energy in terms of total cost depended upon the pricing ratio of water heating power to refrigeration power. For a pricing ratio of 1:1, a 12 bar condenser pressure proved the most cost effective, whereas a 6.5 bar condenser pressure was the most efficient at a pricing ratio of 1:3.

3) Efficiencies can be improved with the installation of a suction superheater.

The impact of a heat recovery system on a refrigeration water heating system, operating under field conditions, is analysed in Chapter 8 using data from the field survey presented in Chapter 7. Using this as a model, an economic assessment of heat recovery and its impact on the New Zealand dairy industry is made.

CHAPTER 7

EXPERIMENT III RESULTS AND DISCUSSION

7:1 WATER HEATING

Of the two hot water cylinders monitored at the dairy shed, one supplied water for vat cleaning at a temperature of 78^{o}C while the other supplied hot water for plant cleaning at a temperature of 95^{o}C twice per day.

7:1.1 Vat Cylinder

Results of total power consumption, heating power (power used during the period needed to heat the water to the required temperature) and standing power (power required to maintain water temperatures against heat losses) are presented in Figure 7:1 with the data given in Table A7:1. From these results the relationships between power consumption (total, heating and theoretical heating) and the volume of water heated were established.

In some cases the volume of water entering the cylinder was either very low, indicating that no cleaning was done, or very high, indicating that water had overflowed. These cases were rejected for the regression analysis. For the remaining data, a regression analysis showed that there was a high correlation (0.98) between total power consumption and the volume of water heated. The regression equation for this relationship is given by:-

$$\text{Power (kWh)} = b \times \text{Volume (l)} + c \qquad \text{.....Eqn 7:1}$$

where $b = 0.067 \pm 0.004$ kWh.l^{-1}

$c = 5.2 \pm 0.4$ kWh

The y intercept (c) represents the significant standing heat losses (5.2 kWh) for a full cylinder over a 24 hour period. This figure is equivalent to a heat loss rate of 0.22 kW which compares well with the average figure of 0.23 kW calculated from the expression:-

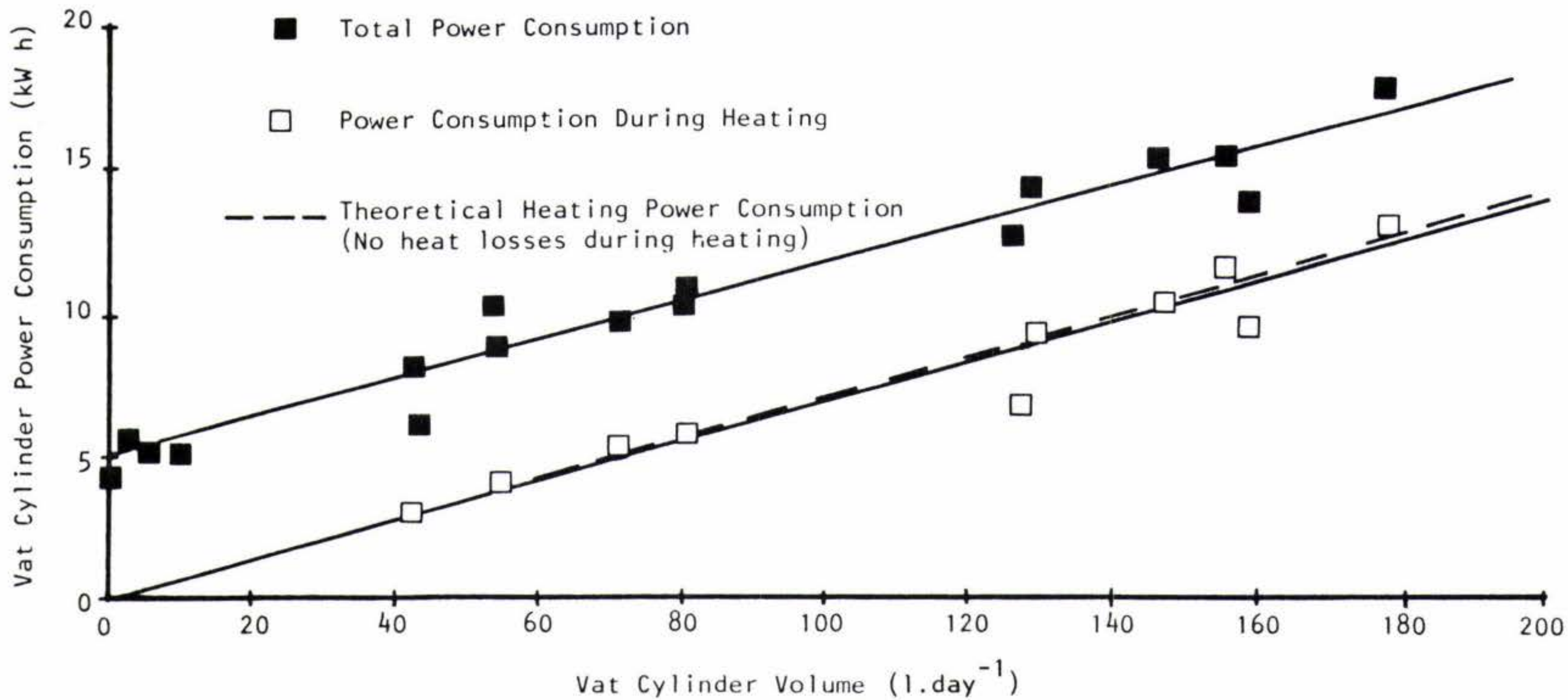


FIGURE 7:1

Relationship between vat cylinder volume and power consumption

$$\text{Heat loss rate (kW)} = \frac{\text{Standing Power (kWh)}}{(24 - \text{heating time})\ (\text{h})} \quad .....\text{Eqn 7:2}$$

The equation for heating power (Figure 7:1) was found to be

$$\text{Power (kWh)} = b' \times \text{Volume (l)} \quad .....\text{Eqn 7:3}$$

$$\text{where} \quad b' = 0.069 \pm 0.002 \ \text{kWh.l}^{-1}$$

The slopes of these two lines, and the slope of the theoretical heating curve (0.073 $\pm$ 0.002 kWh), were not significantly different. This suggests that the heat losses during heating were small.

Based on the slope of the theoretical heating curve and the standing heat losses, results of total experimental and theoretical cylinder power consumption are presented in Figure 7:2, together with experimental and theoretical heating power consumption. The figure shows that this method of predicting total power consumption gives good agreement with the experimental results.

The figure also shows that the average total standing losses (5.0 kWh per 24 hours) represent a significant proportion (47.5%) of the average total power of 10.5 kWh required to heat an average daily volume of 78 litres from 15^{o}C to 78^{o}C. This volume of water was 2 litres less than the recommended volume of 80 litres (Section 2:2.2.2). The relatively high proportion of energy wasted by heat loss was due to the long periods of time during which the water was held at the final temperature. In this instance, the use of a time clock to limit the heating process to the hours immediately prior to water drawoff could be justified (see Table A7:1). The variation in power consumption (Figure 7:2) was due to the variation in water drawoff for vat cleaning. In some instances the vats were cleaned every two days with double the required volume and, in others, once per day with the required volume. The overall effect was that the daily average was close to the required volume as previously discussed.

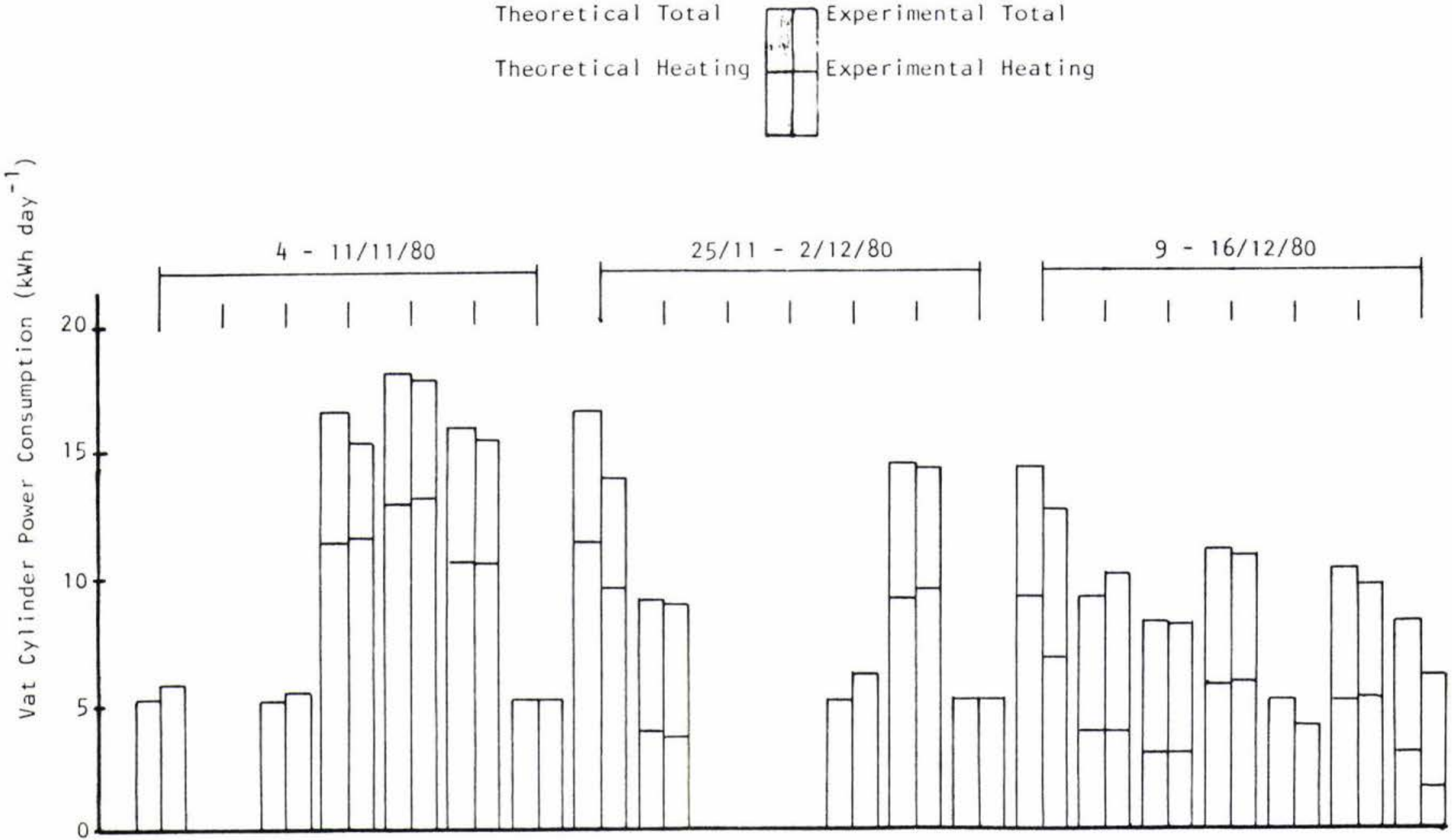


FIGURE 7:2

A comparison between theoretical and experimental total and heating power consumption - vat cylinder

7:1.2 Machine Water Cylinder

The method of analysis was the same as for Section 7:1.1 and the results are presented in Figure 7:3 (data in Table A7:2).

The curves are similar to those in Figure 7:1 with the regression equation for the total power being:-

$$\text{Power (kWh)} = b \times \text{Volume (l)} + c \qquad \text{.....Eqn 7:4}$$

where $b = 0.075 \pm 0.008 \ \text{kWh.l}^{-1}$

$c = 9.99 \pm 1.7 \ \text{kWh}$

(Correlation Coefficient = 0.92)

while the regression equation for heating power is:-

$$\text{Power (kWh)} = b' \times \text{Volume (l)} \qquad \text{.....Eqn 7:5}$$

where $b' = 0.098 \pm 0.008 \ \text{kWh.l}^{-1}$

The slopes of these two lines are significantly different. This is because there is an apparent decrease in standing heat losses for the larger water volumes used. As the volume of water to be heated per day increased, the time taken to reach the final temperature increased and the time available for standing heat losses decreased. This effect can be seen in Figure 7:4 where standing power has been plotted as a function of the volume of the water heated. The equation of this line is:-

$$\text{Standing Power (kWh)} = -0.024 \times \text{Volume (l)} + 9.9 \qquad \text{.....Eqn 7:6}$$

The total power required to compensate for heat losses (total heat loss power) was the sum of the losses during heating and after the final temperature had been reached. The value of total heat loss power can be determined from the difference

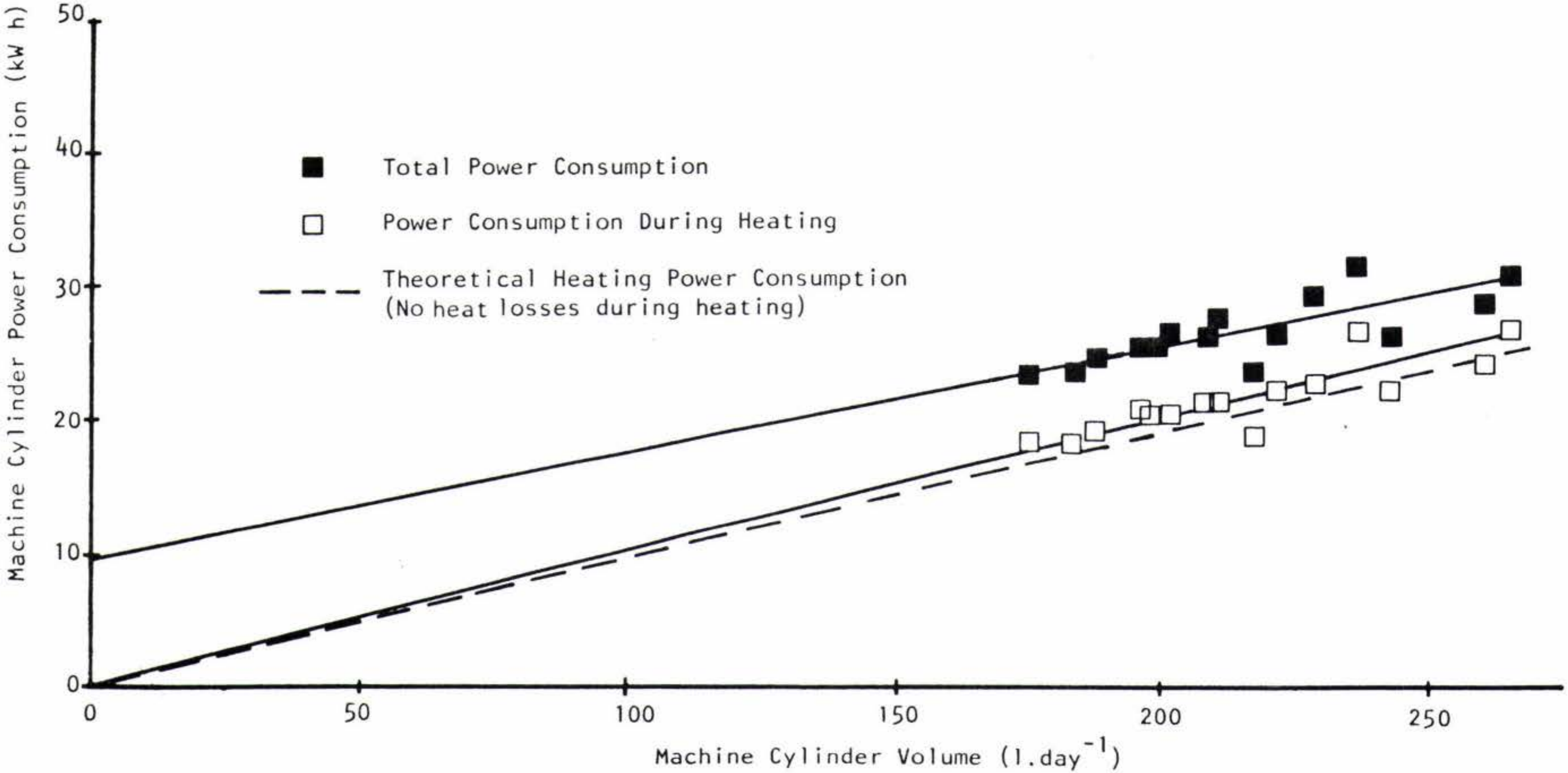


FIGURE 7:3

Relationship between machine cylinder volume and power consumption

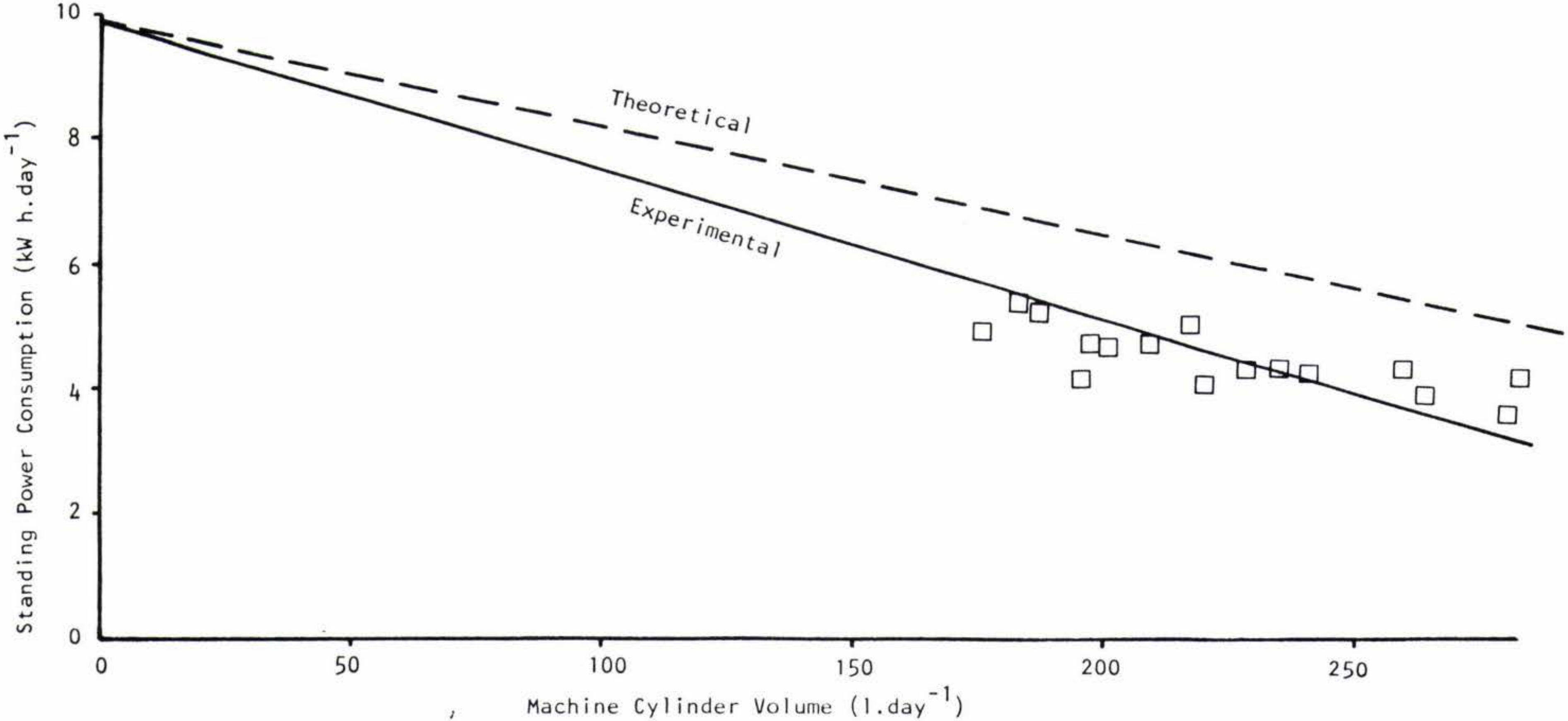


FIGURE 7:4

The relationship between standing power (after final temperature is reached) per day and the volume of water heated between Eqn 7:4 and the equation of theoretical heating power of:-

Power (kWh) = (0.093 ± 0.002) X Volume (l) .....Eqn 7:7

From these two equations the equation of the curve (Figure 7:5) for total heat loss power as a function of the volume heated is:-

Total Heat Loss Power (kWh) = -0.018 X Volume + 9.9 .....Eqn 7:8

It can be shown that for any cylinder the difference in slopes between the total heat loss power curve and the standing power curve is inversely proportional to heating element capacity, i.e., as element capacity decreases the heat losses during heating increase.

The difference in magnitude in the slopes of the power curves between the vat cylinder and the machine cylinder reflects the higher final temperature of the machine cylinder and the differences in insulation effectiveness. It was noted that the vat cylinder's insulation was in a poorer condition than the machine cylinder's. The heat loss rate for the machine cylinder was estimated to be 0.31 kW, based on Eqn 7:2, and was higher than the vat cylinder (0.22 kW) due to the higher water temperature.

Using Eqns 7:7 and 7:8, theoretical and experimental total power were plotted against time, together with theoretical and experimental heating power (Figure 7:5). The figure shows the variability in machine cylinder power consumption due to the variation in hot water use. However, the mean daily volume of 219 litres was only 31 litres less than that recommended by the regulations (Section 2:2.2.2). The high peaks reflect the irregular use of the volume used for plant cleaning (e.g., exterior cleaning of the milk cups). Average total power consumption for the machine cylinder was 25.8 kWh per day, of which 23.3% or 6.0 kWh per day was for total heat

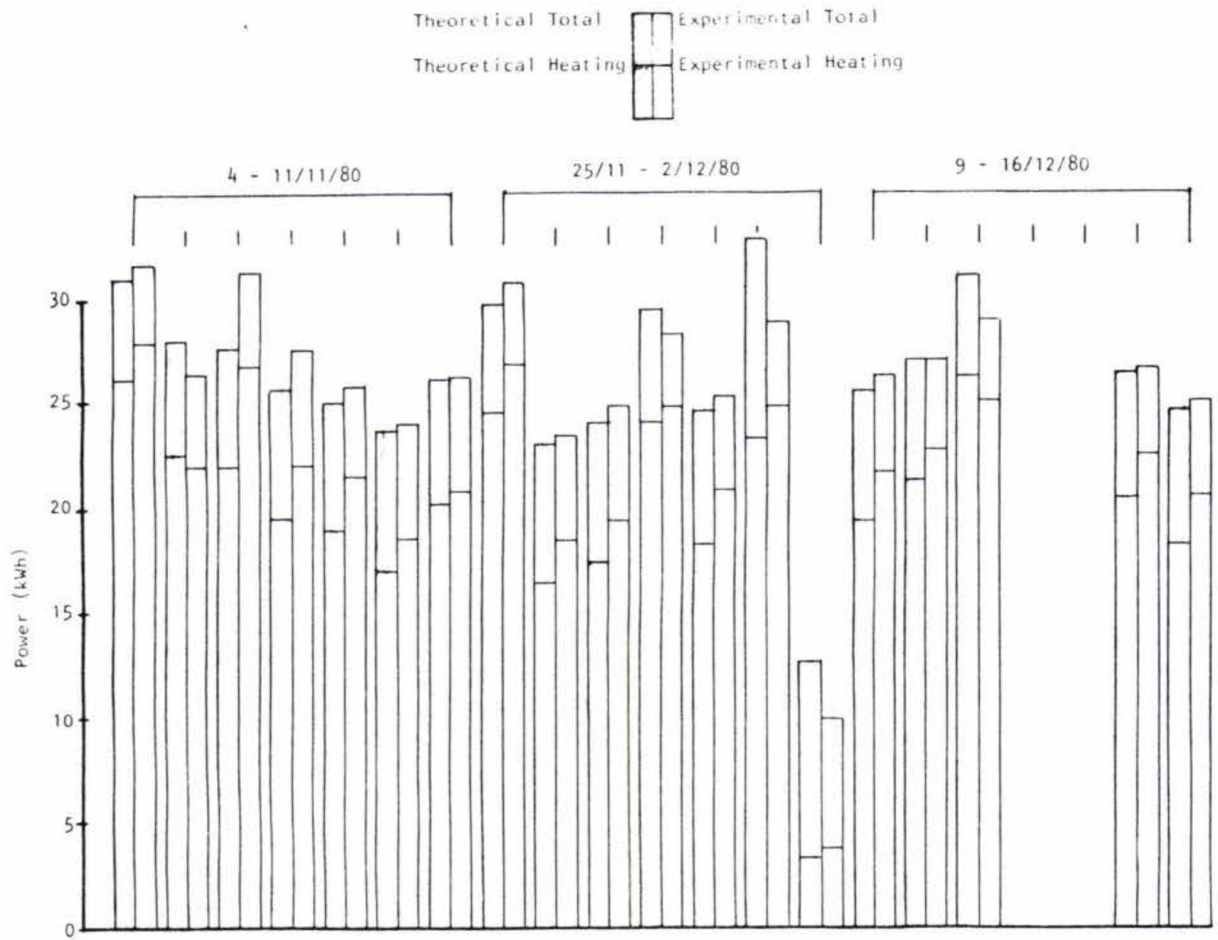


FIGURE 7:5

A comparison between theoretical and experimental total and heating power consumption - machine cylinder loss power consumption. The figure 6.0 kWh per day for the machine cylinder is 20% greater than the 5.0 kWh per day figure for the vat cylinder. Again, this reflects the higher temperature regime of the machine cylinder compared to the vat cylinder.

For the machine cylinder the use of time clocks (as for the vat cylinder) would be justifiable. In addition, the installation of the maximum capacity heating element available would assist in reducing heat losses during heating.

7:2 REFRIGERATION SYSTEMS

Results of milk volumes, milk temperatures and power consumption were determined from the data and are presented in Tables 7:1 and 7:2, for the 2.25 kW and 3.75 kW systems respectively.

Since milk pickup was within the 3 hour regulation time after the completion of the morning milking, the milk final temperature of $4^{o}C$ was not reached. To overcome the problem of comparing results for varying final temperatures, the milk volumes were corrected to give results for the equivalent volume of milk cooled to $4^{o}C$. These results are presented, with power consumption, in Figures 7:6 and 7:7.

A regression analysis of the corrected data from the 2.25 kW system produced a relationship between power consumption and milk volume of:-

Power (kWh) = b X Volume cooled to $4^{o}C$ (l) .....Eqn 7:9

where b = 0.0069 ± 0.0003 kWh.l^{-1}

(Correlation Coefficient = 0.80)

TABLE 7:1

Data summary for 2.25 kW refrigeration system

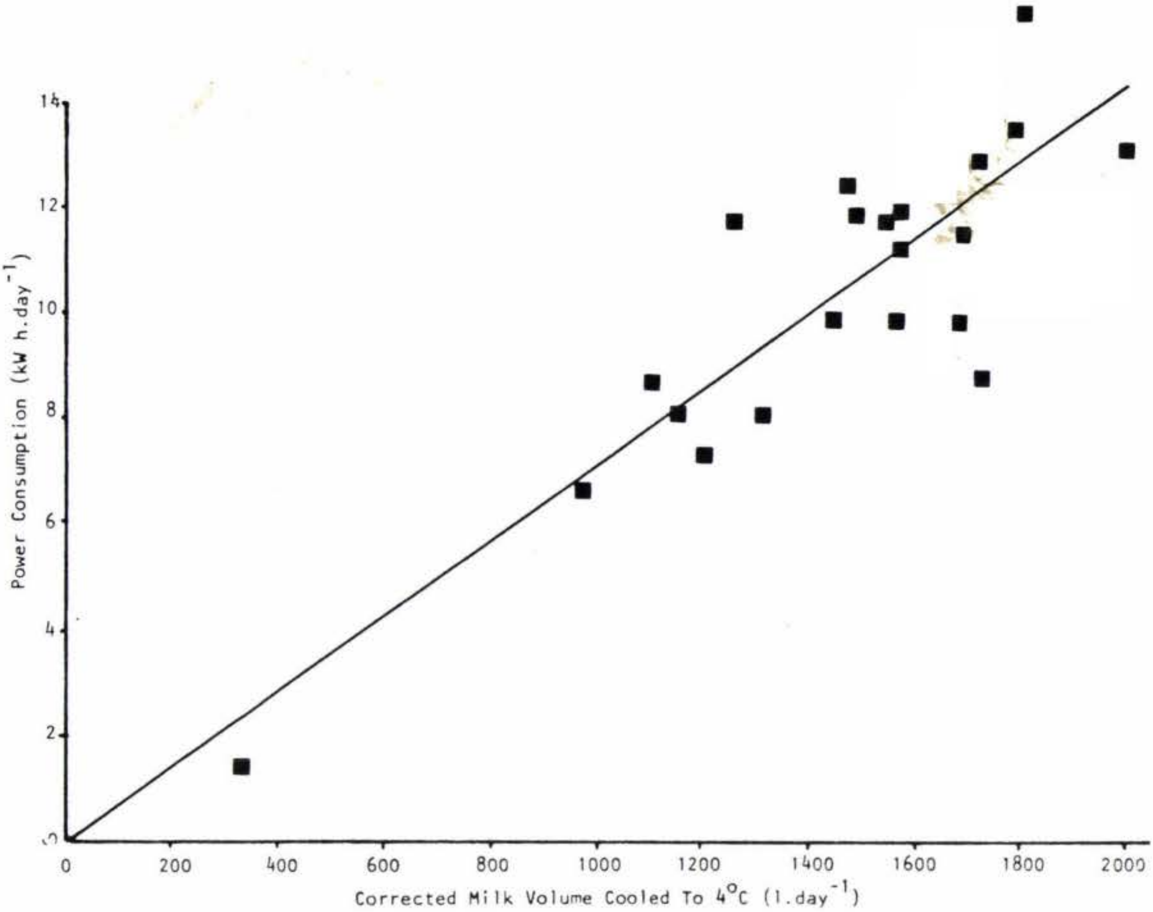
Day	Milk Volume (l)			Temperature		Total kWh	Ave. C.O.P.
	Evening	Morning	Corrected Total	Mix (°C)	Pickup		
4	1085	1210	1811	14	8	15.69	2.55
5	910	190	1100	7	4	8.67	2.80
6	1000	1160	1250	14	12	11.79	2.34
7	1000	475	1467	10	4	12.39	2.63
8	700	788	1488	14	4	11.91	2.76
9	750	740	1333	13	6	8.52	3.46
10	810	850	1310	14	8	8.07	3.59
25	930	1360	1566	15	10	11.31	3.06
26	-	-	-	-	-	-	-
27	-	-	-	-	-	-	-
28	880	1225	1440	15	10	9.96	3.19
29	-	-	-	-	-	-	-
30	960	1340	1694	15	9	9.84	3.81
31	-	-	-	-	-	-	-
39	-	-	-	-	-	-	-
40	-	-	-	-	-	-	-
41	-	-	-	-	-	-	-
42	-	-	-	-	-	-	-
43	-	-	-	-	-	-	-
44	950	1350	1573	15	10	11.97	2.90
45	-	-	-	-	-	-	-

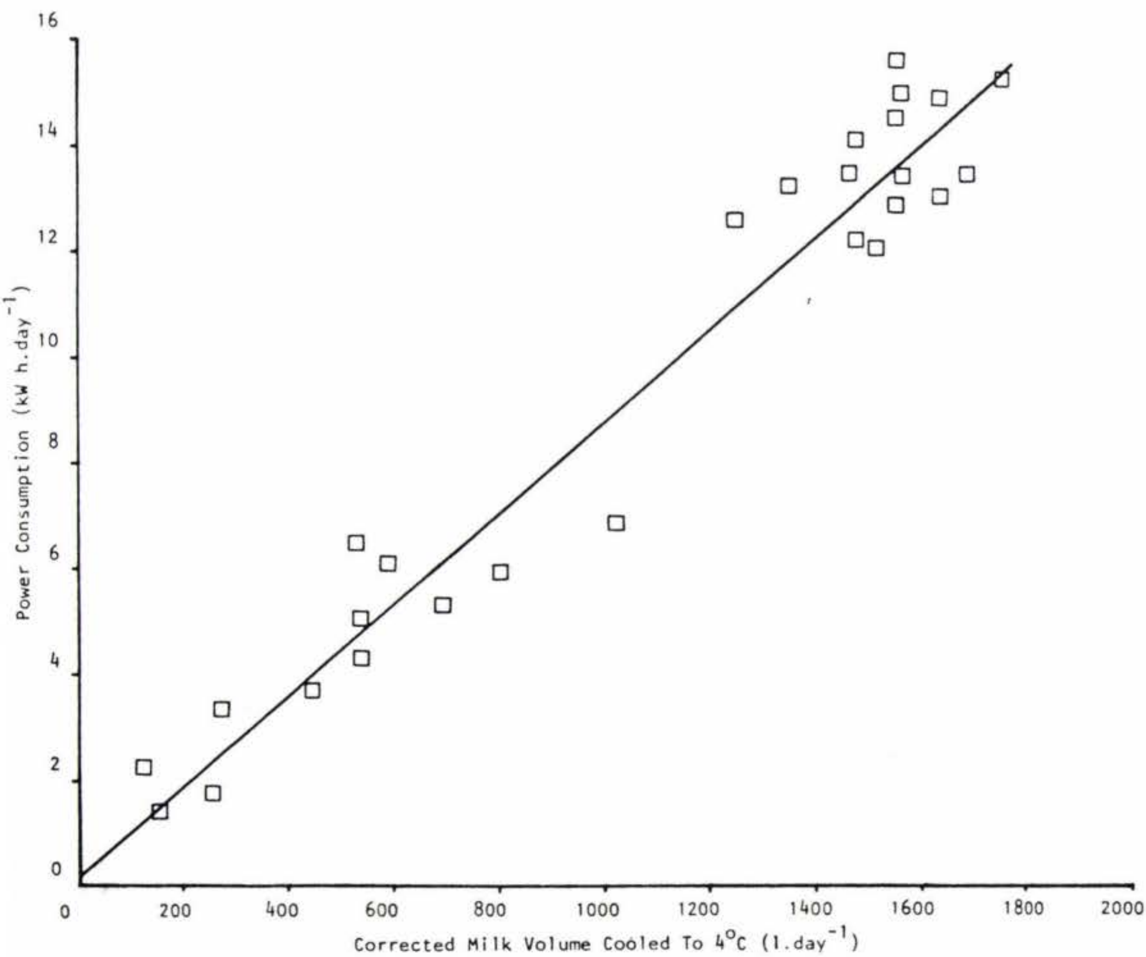
TABLE 7:2

Data summary for 3.75 kW refrigeration system

Day	Milk Volume (l)			Temperature (^oC)		Total kWh	Ave. C.O.P.
	Evening	Morning	Corrected Total	Mix	Pickup		
4	-	455	455	-	4	3.69	2.72
5	-	1485	1016	-	10	6.84	3.28
6	-	-	-	-	-	-	-
7	-	1135	537	-	14	5.07	2.34
8	300	815	645	18	12	4.56	3.13
9	200	485	685	17	4	5.34	2.83
10	-	595	595	-	4	6.12	2.15
25	-	-	-	-	-	-	-
26	1025	1185	1512	14	10	12.18	2.74
27	850	1380	1995	16	6	12.84	3.43
28	-	-	-	-	-	-	-
29	770	1280	1186	15	-	14.70	1.78
30	-	-	-	-	-	-	-
31	670	1280	1436	16	9	8.10	3.92
39	-	-	-	-	-	-	-
40	830	1240	1634	15	8	13.11	2.75
41	830	1310	1689	16	8	10.56	2.25
42	830	1555	1882	16	8	9.03	2.03
43	960	1230	-	-	20*	-	-
44	-	-	-	-	-	-	-
45	870	1410	1560	16	10	13.53	2.55

*Refrigeration power failure


Power Consumption (kW h.day^{-1})
Corrected Milk Volume Cooled To 4°C (l.day^{-1})
0
2
4
6
8
10
12
14
200
400
600
800
1000
1200
1400
1600
1800
2000


Power Consumption (kW h.day^{-1})
Corrected Milk Volume Cooled To 4°C (l.day^{-1})
0
2
4
6
8
10
12
14
16
200
400
600
800
1000
1200
1400
1600
1800
2000

and for the 3.75 kW system, of:-

$$\text{Power (kWh)} = b' \times \text{Volume cooled to } 4^{0}\text{C (l)} \qquad \text{.....Eqn 7:10}$$

where $b' = 0.0085 \pm 0.0002 \text{ kWh.l}^{-1}$

(Correlation Coefficient = 0.95)

The differences in these slopes are significant and reflect the higher power requirements of the 3.75 kW system to cool the same load. This lower performance is also expressed in the lower average C.O.P. of 2.71 for the 3.75 kW system compared to the C.O.P. of 3.1 for the 2.25 kW system.

The cooling capacities for the two systems, of 166 l per kWh and 125 l per kWh, for the 2.25 kW and 3.75 kW system respectively, were lower than the 200 l per kWh figure suggested by Currier (1976). The differences were probably due to the variation in the conditions used.

A comparison of cooling times with the regulations is not possible since the evening milk volumes were well within the system's capacity and were cooled easily within the allowed 3 hour time period, while for the morning volumes the milk was picked up at the completion of milking, i.e., within the regulation 3 hour time period.

The split between evening and morning milk production was 41% and 59% respectively. These figures are based on 50 consecutive days production and compare well with the figures of 40% and 60% discussed in Section 2:4.1.1.

Because of the short time period between the evening and morning milkings, the relatively low ambient night temperatures (12^{0}C) and vat insulation, standing heat gains by the milk were insignificant. (For example, it was noted that the refrigeration units infrequently switch on again after the milk had reached the required temperature.) These gains were therefore included in the total refrigeration power consumption and, as a consequence, may account for some of the scatter in the regression analysis.

7:3 TOTAL POWER CONSUMPTION

Total shed power consumption data is presented, along with the power consumption results for both water cylinders and refrigeration systems, in Figure 7:8 (data in Table A7:3).

The results show that variations in water heating contribute the greater proportion of the variation in total power. This was expected, since water heating represents 50% of the total power consumed, while refrigeration power consumption represents only 14.5% of the total power.

This data is very similar to the percentages for other dairy farms, as discussed in Section 2:2.2 and 2:2.3, with hot water heating being 5% above the average range of the literature values (40-45%), while refrigeration was 3.5% below the range of the literature values (18-25%) due to early milk pickup.

7:4 CONCLUSIONS

The development of relationships between the amount of power used and the volume of water heated, or the volume of milk cooled, allows the prediction of power consumption for volumes other than those recorded. In conjunction with the relationships between heating and refrigeration power consumption and total farm power consumption (50% and 14.5% respectively), the impact of either increasing hot water requirements, increasing cooling requirements, the use of time clocks on heating elements, or the installation of a heat recovery system, can be more readily assessed.

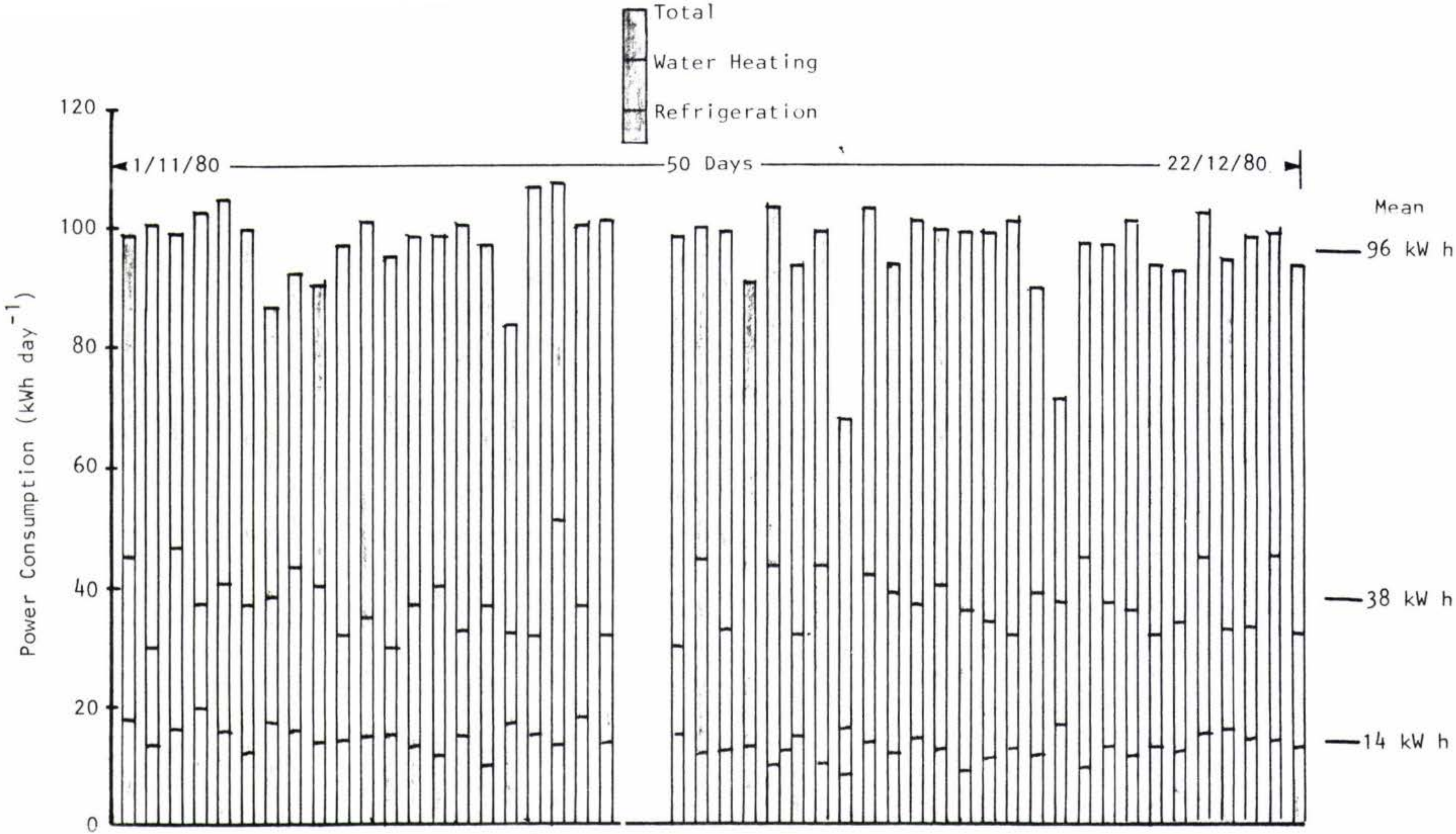


FIGURE 7:8

A comparison between total, water heating and refrigeration power consumption

CHAPTER 8

ECONOMIC ASSESSMENT OF HEAT RECOVERY AND POTENTIAL IMPACT ON THE NEW ZEALAND DAIRY INDUSTRY

8:1 ECONOMIC ASSESSMENT OF REFRIGERATION HEAT RECOVERY

8:1.1 Economic Assessment of Experimental and Field Plants

The results for the experimental heat recovery system described in Chapter 6 were applied to the results from the field survey (Chapter 7), from which an estimate of the savings potential of the heat recovery system was determined.

Firstly, the analysis required the calculation of the daily total energy used by the refrigeration and water heating system operating under field conditions without heat recovery (Standard System). In Section 7:1.1 the equation for the vat cylinder water heating energy consumption in kWh, including standing heat losses (E_v), was expressed as:-

$$E_v = 0.067\ V_{vat} + 5.2 \qquad \text{.....Eqn 7:1}$$

where V_{vat} = daily volume of eater heated in the vat cylinder (l)

The equation for the machine cylinder water heating energy in kWh (E_m) from Section 7:1.2 was expressed as:-

$$E_m = 0.075\ V_{mach} + 9.9 \qquad \text{.....Eqn 7:4}$$

where V_{mach} = daily volume of water heated in the machine cylinder (l)

Using Eqns 7:1 and 7:4 gives an equation for the total water heating energy consumption in kWh (E_{TH}) of:-

$$E_{TH} = 0.067\ (V_{vat} + 1.12\ V_{mach}) + 15.1 \qquad \text{....Eqn 8:1}$$

The equation for the daily refrigeration requirements of the 2.25 kW field refrigeration system (E_r, in kWh) from Section 7:2 was expressed as:-

$$E_r = 0.0069\ V_{milk} \qquad \text{.....Eqn 7:9}$$

where V_{milk} = milk volume cooled (l)

Adding Eqns 7:9 and 8:1 gives an expression for the total water heating and refrigeration energy requirements of the standard system (E_{TS}, in kWh):-

$$E_{TS} = 0.067\ (V_{vat} + 1.12\ V_{mach}) + 15.1 + 0.0069\ V_{milk}$$

$$\text{.....Eqn 8:2}$$

Since the daily volume of hot water required is constant for any one dairy shed, and is independent of daily milk volume, then taking the average daily water volumes of V_{vat} = 78 litres (Section 7:1.1) and V_{mach} = 219 litres (Section 7:1.2) and substituting into Eqn 8:2 gives:-

$$E_{TS} = 0.069\ V_{milk} + 36.8 \qquad \text{.....Eqn 8:3}$$

Secondly, the analysis for the heat recovery system required the calculation of the daily energy requirements of the refrigeration and water heating system and the nett heat recovery energy.

The energy requirements for water heating were the same (36.8 kWh), whether the heat recovery or the standard refrigeration system was used.

For a pricing ratio of 1:1, a 12 bar condenser pressure was recommended in Section 6:13.3. Refrigeration energy consumption for the water cooled heat recovery system (without a primary heat exchanger) operating under these conditions, and cooling 2000 litres of milk from 23^{o}C to 4^{o}C (19^{o}C differential), was 16.8 kWh (Figure 6:22). Section 7:2 indicated that refrigeration energy consumption was directly proportional to milk volume (Eqn 7:9), and if the conclusion is applied to the laboratory experiment, then the refrigeration energy consumption in kWh (E_{rH}) may be written as:-

$$E_{rH} = \frac{16.8}{2000} \cdot V_{milk}$$

$$= 0.0084\ V_{milk} \qquad .....\text{Eqn } 8:4$$

Expressing nett heat recovery (Q_n in $kWh.l^{-1}$), for the water cooled heat recovery system, as a function of gross heat recovery (Q_g in $kWh.l^{-1}$) gives:-

$$Q_n = Q_g\ (1 - Ke) \qquad .....\text{Eqn } 8:5$$

(since Q_n = Q_g - additional compressor energy ($kWh.l^{-1}$) and

Ke = additional compressor energy/Q_g)

Therefore, the total daily nett heat recovery energy in kWh can be expressed as:-

$$E_n = [Q_g\ (1 - Ke)]\ V_{milk} \qquad .....\text{Eqn } 8:6$$

(since $Q_g \ \alpha\ V_{milk}$)

For the water cooled heat recovery system, daily gross heat recovery was 0.0081 $kWh.l^{-1}$ (Figure 6:29). Substituting this value, and the value for Ke of 0.26 (Figure 6:33), into Eqn 8:6 gives:-

$$E_n = 0.0060\ V_{milk} \qquad .....\text{Eqn } 8:7$$

The total daily energy requirements of the heat recovery system (E_{TH} in kWh) can be determined from the sum of the water heating and refrigeration energy requirements less the nett heat recovery energy (Eqns 8:3, 8:4 and 8:7);

$$E_{TH} = 36.8 + 0.0084\ V_{milk} - 0.0060\ V_{milk}$$

$$= 0.0024\ V_{milk} + 36.8 \qquad .....\text{Eqn } 8:8$$

The daily energy savings in kWh (E_{sav}) from installing a water cooled heat recovery system was determined from the difference between the energy requirements of the standard system and the heat recovery system (Eqns 8:3 and 8:8);

$$E_{sav} = E_{TS} - E_{TH}$$

$$= (0.0069\ V_{milk} + 36.8) - (0.024\ V_{milk} + 36.8)$$

$$= 0.0045\ V_{milk} \qquad .....\text{Eqn } 8:9$$

Assuming that the suction superheater would increase heat recovery by 15% for the 12 bar water cooled system, then the constant in Eqn 8:9 would become 0.0057 $kWh.l^{-1}$.

Applying the same procedure to the results for the air cooled system gave a value of the constant in Eqn 8:9 of 0.0015 $kWh.l^{-1}$ at a 12 bar condenser pressure.

Figure 8:1a presents plots of Eqn 8:9 for the water cooled heat recovery system with and without the suction superheater, and also for the air cooled heat recovery system. A plot for the water cooled system at a 6.5 bar condenser pressure and a pricing ratio of 1:3 is also presented. It was evident that heat recovery for a 1:3 pricing ratio was not a viable proposition, even at this optimum condition, and therefore no further plots were done at this pricing ratio.

Calculations for other milk temperature cooling differentials showed that the percentage changes in the slope of Eqn 8:9 were the same as the percentage change in the $23^{o}C$ - $4^{o}C$ differential.

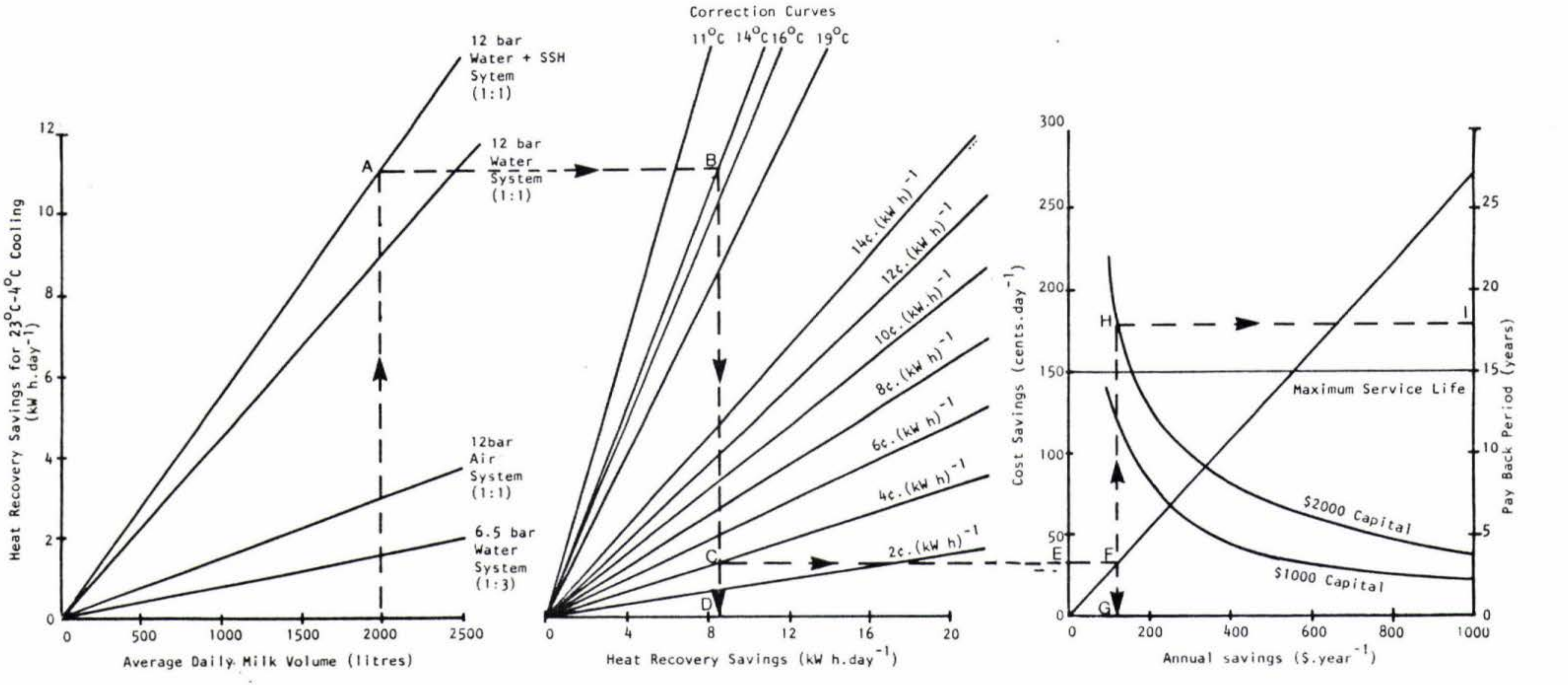


FIGURE 8:1a

The effect of milk volume on heat recovery savings

FIGURE 8:1b

Heat recovery correction curves and pricing lines

FIGURE 8:1c

Heat recovery annual savings and payback curves

The percentage variations in heat recovery savings with changing milk temperatures have been presented as correction curves in Figure 8:1b. Figure 8:1b also presents lines for selected energy unit costs from which the value of the energy recovery savings can be determined in cents.day^{-1}. The level of annual savings (\$.year^{-1}) from heat recovery is determined from Figure 8:1c.

For example, assuming that the average daily milk production over 365 days of 2000 litres is cooled by a water cooled refrigeration heat recovery system with a suction superheater from 18^{o}C to 4^{o}C, and that 78 litres of water is heated to 78^{o}C and 219 litres is heated to 95^{o}C, the energy savings gained can be determined by entering Figure 8:1a at 2000 litres, from which the energy savings for a 12 bar water cooled suction superheater system can be read from the vertical axis (Point A). Selecting the correction curve for a 14^{o}C differential (18^{o}C - 4^{o}C) in Figure 8:16 (Point B) the energy saving is determined from the horizontal axis at Point D (8.8 kWh.day^{-1}). The economic value of the saving is determined from the selection of the appropriate cost per kWh line (e.g., 4 cents per kWh at Point C) and reading across to the vertical axis at Point E to obtain the value in cents per day (35 cents .day^{-1}) and the horizontal axis of Figure 8:1c (Point G via Point F) for the value in \$.year^{-1} (\$135 year^{-1}). The 8.8 kWh.day^{-1} energy saving represents 17% of the total energy required by the standard system. Following the same procedure for an air cooled refrigeration system yields a saving of 2.3 kWh (4.5% of the total), valued at 10 cents.day^{-1}.

The capital outlay for heat recovery depends upon the system selected. For an air cooled system this involves installing a primary heat exchanger at a cost of \$1000 (1982 price). Since a large proportion of refrigeration systems are currently air cooled, converting to a water cooled refrigeration heat recovery system requires the expenditure of approximately \$2000 (\$1000 for a primary heat exchanger and \$1000 for replacing the air cooled secondary heat exchanger with a water cooled heat exchanger).

The payback period for both systems were calculated for a number of annual savings rates (a sample calculation is presented in Table 8:1), and are plotted in Figure 8:1c. The calculation of the payback period assumed a first year depreciation rate of 20%, and for subsequent years 10%, while post depreciation profit was taxed at a rate of 55 cents per dollar. Yearly post tax income was summed until the original investment was recovered. No allowance has been made for opportunity cost since this is expected to be covered by the increase in power charges.

Taking the annual savings for the water cooled system of $\$135\ .\mathrm{year}^{-1}$, and a capital investment of $2000, the payback curve in Figure 8:1c (Point H) gives a payback period of 16.75 years. For the air cooled system, the payback period exceeds 25 years. The maximum service life of the equipment was set at 15 years, which means that the water cooled system was uneconomical.

This analysis has made no allowance for heat losses from piping or the intermediate storage cylinder. The losses in pipework are dependent upon the distance between the storage cylinders and the primary heat exchanger, and the level of insulation. The heat losses from the storage and water heating cylinders are dependent upon the level of insulation and the standing time. Standing times will depend upon when hot water is used relative to the time of heat recovery, i.e., hot water management. Figure 8:2 gives a suggested sequence of events between the water heating cylinders and a buffer cylinder. The buffer cylinder stores the heat recovery water until the completion of milk cooling then dumps the water to the appropriate water heating cylinders where makeup water is added if required. Heating elements can be programmed with time clocks so that the required water temperatures are reached immediately prior to water drawoff.

No allowance has been made for the additional maintenance costs or reduced service life of the compressor due to operating at a condenser pressure of 12 bars.

TABLE 8:1

Summary of payback period calculation for a $2000 investment and annual savings of $100

(+ve tax = tax refund, -ve tax = tax paid)

Year	Capital Value ($)	Annual Savings ($)	Deprec-iation ($)	Taxable Savings ($)	Tax ($)	Post Tax Savings ($)	Accumulated Savings ($)
1	2000.0	100	400.0	-300.0	165.0	265.0	265.0
2	1600.0	100	160.0	-60.0	33.0	133.0	398.0
3	1440.0	100	144.0	-44.0	24.2	124.2	522.2
4	1296.0	100	129.6	-29.6	16.3	116.3	638.5
5	1166.4	100	116.6	-16.6	9.2	109.2	747.7
6	1049.8	100	105.0	-5.0	2.7	102.7	850.4
7	944.8	100	94.5	5.5	-3.0	97.0	947.3
8	850.3	100	85.0	15.0	-8.2	91.8	1039.1
9	765.3	100	76.5	23.5	-12.9	87.1	1126.2
10	688.7	100	68.9	31.1	-17.1	82.9	1209.1
11	619.9	100	62.0	38.0	-20.9	79.1	1288.2
12	557.9	100	55.8	44.2	-24.3	75.7	1363.8
13	502.1	100	50.2	49.8	-27.4	72.6	1436.5
14	451.9	100	45.2	54.8	-30.1	69.9	1506.3
15	406.7	100	40.7	59.3	-32.6	67.4	1573.7
16	366.0	100	36.6	63.4	-34.9	65.1	1638.8
17	329.4	100	32.9	67.1	-36.9	63.1	1701.9
18	296.5	100	29.6	70.4	-38.7	61.3	1763.2
19	266.8	100	26.7	73.3	-40.3	59.7	1822.9
20	240.2	100	24.0	76.0	-41.8	58.2	1881.1
21	216.1	100	21.6	78.4	-43.1	56.9	1938.0
22	194.5	100	19.5	80.5	-44.3	55.7	1993.7
23	175.1	100	17.5	82.5	-45.3	54.6	2048.3

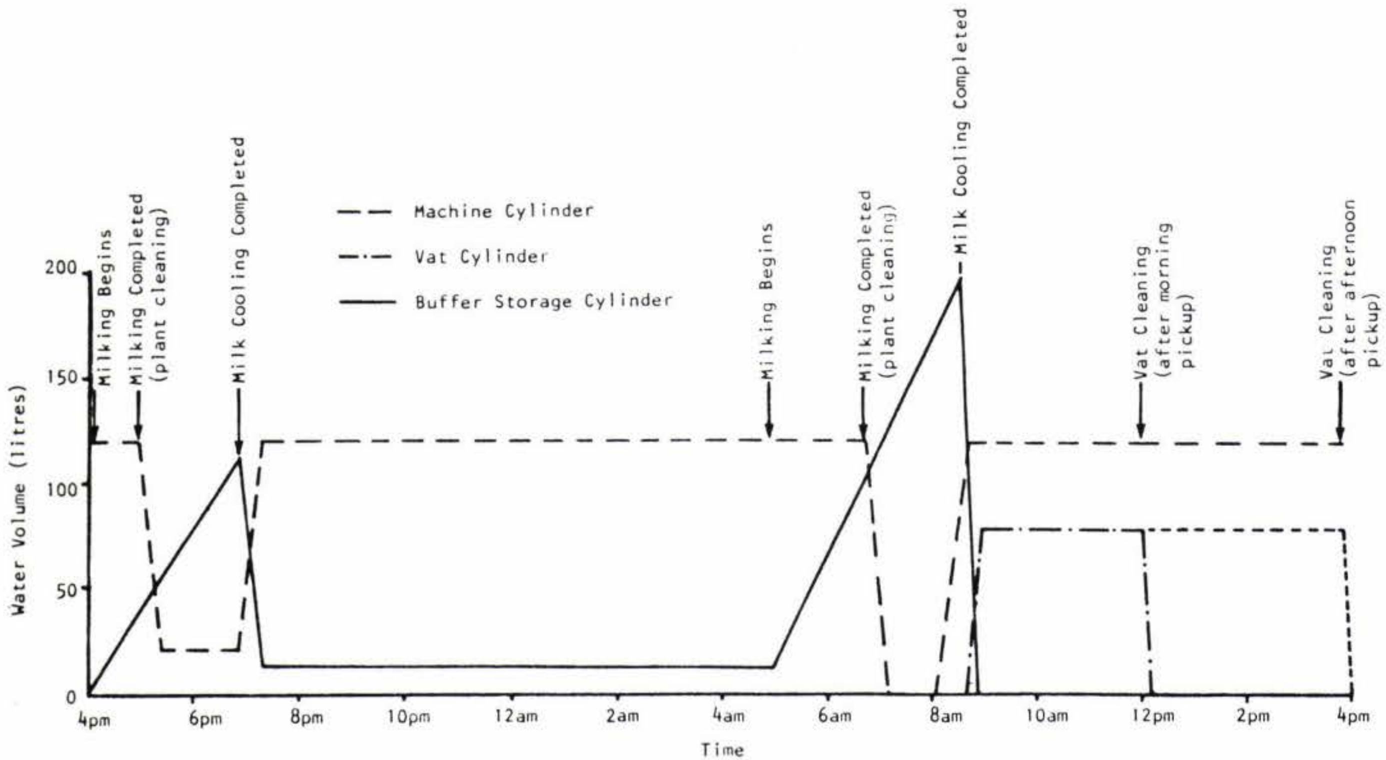


FIGURE 8:2

Filling sequence for machine and vat water heating cylinders and the heat recovery buffer storage cylinder

8:1.2 Potential for Increasing Heat Recovery Savings

Changing the energy requirements for water heating has no effect on the level of energy savings from heat recovery since the changes effect both the standard system and the heat recovery system. Reducing the energy requirements for refrigeration reduces the quantity of heat recovery and as a consequence its efficiency over the standard system (Section 6:12).

To increase the level of heat recovery for the system tested requires that the rate of nett heat recovery (kW) be increased, since the system was operating for the maximum time limit when cooling milk from 23^{o}C to 4^{o}C, i.e., no potential for increasing kWh of energy by increasing cooling time. Heat recovery rate can only be increased by increasing water flow rate through the primary heat exchanger, thereby increasing U, or by increasing the log mean temperature difference or the area of the heat exchanger.

Increasing water flow rate is not possible since this is fixed to give the required volume for plant cleaning. However, increasing the velocity of the water through the heat exchanger to improve the overall thermal conductance would require the manufacture of special cores, since none are commercially available (Appendix A4:2).

The log mean temperature difference could only be increased either by increasing the water flow rate (which would reduce the water temperature rise) or by increasing vapour inlet temperatures by increasing the compressor head pressure (which would increase power consumption) or by reducing water inlet temperatures (which would reduce water outlet temperatures).

Increasing heat exchanger area is the only option available. However, the increase in heat recovery due to an increase in area is more complex than multiplying heat recovery by an area scaling factor due to the effect of decreasing the log mean temperature difference. In addition, increasing area by increasing the heat exchanger length would increase the refrigerant head loss and, as a consequence, increase the compressor power consumption.

Heat recovery rates for varying heat exchanger areas were estimated using the model developed in Appendix A6:2. and are presented in Figure 8:3. Using the model to predict past $2m^2$ was not possible due to condensation conditions being reached. The results show that heat recovery rate for the primary heat exchanger only increased by 33% for a 238% increase in heat exchanger area, largely due to the $12.5^{o}C$ (44%) decrease in log mean temperature difference. This percentage increase was reduced to 19% when the total heat recovery rate was considered due to the addition of the 0.87 kW heat recovery rate from the secondary water cooled heat exchanger. The increase in headloss for the $2m^2$ area was estimated to 2.38 bars since the length per leg for a $2m^2$ area is 2.38 times the length for a $0.84\ m^2$ area, and headloss is directly proportional to length (1.0 bar for $0.84\ m^2$ as in Figure 6:18). The increase in instantaneous compressor power consumption was estimated to be 0.24 kW for the $2m^2$ area since the addition of the primary heat exchanger increased power consumption by 0.10 kW per bar of headloss for the $0.84\ m^2$ area (Figure 6:21).

Summing both heat recovery rate and instantaneous power consumption over the cooling time for the 12 bar water cooled condenser system resulted in 11.56 $kWh.day^{-1}.m^{-3}$ of heat recovery energy and 3.32 $kWh.day^{-1}.m^{-3}$ of additional compressor energy. Using these two values Ke was calculated to be 0.29. Working through the same procedure as for the $0.84\ m^2$ area resulted in a 0.0082 slope for the energy saving equation (Eqn 8:9).

The energy saving gained for the $2m^2$ heat exchanger area under the same conditions as for the $0.84\ m^2$ example was 16.4 $kWh.day^{-1}$ which, after correcting to an $18\text{-}4^{o}C$ cooling differential (Figure 8:1b), was reduced to 12.8 $kWh.day^{-1}$.

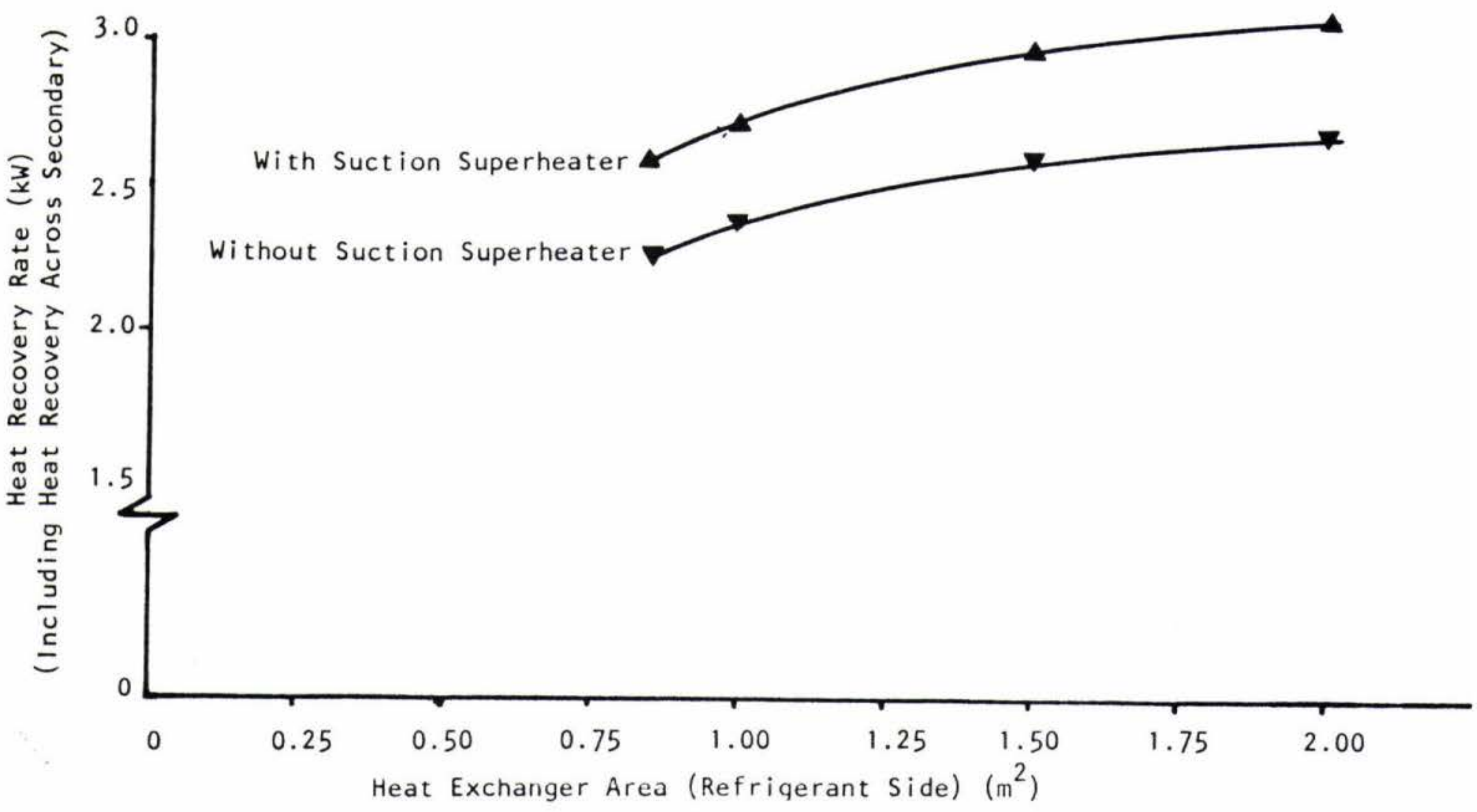


FIGURE 8:3

The effect of primary heat exchanger area on heat recovery for the water cooled system operating at a condenser pressure of 12 bar, with and without a suction superheater

This energy saving represented a 25% reduction in energy costs over the standard system and a monetary saving of 51 cents per day or \$187 per year. The payback period for a \$2000 investment was estimated to be 13.5 years (Figure 8:1). The level of investment was kept constant since the increase in materials for the increased area would be small compared to the labour costs.

It is clear that heat recovery from a 2.25 kW refrigeration system is marginal. However, installation of heat recovery heat exchangers on larger installations (3.75 kW and 5.63 kW systems) would give better returns since higher energy savings could be obtained for a similar capital outlay.

8:2 THE IMPACT OF HEAT RECOVERY ON WATER HEATING ENERGY REQUIREMENTS

Analysis of the impact of gross and nett heat recovery on water heating energy requirements (Table 8:2) shows that gross heat recovery for the water cooled suction superheater system accounted for 51% of the energy required for water heating. This percentage was within the range predicted from the literature (Section 2:6).

The range in the literature did not take into consideration the additional energy requirements of the refrigeration system, which, for the system tested, reduced the contribution made to water heating (12%) for thewater cooled suction superheater system. A similar trend was also recorded for the other systems considered (Table 8:2), but their percentage contributions were lower due to the lower heat recovery rates.

8:3 THE IMPACT OF HEAT RECOVERY ON THE NEW ZEALAND DAIRY INDUSTRY

Although the heat recovery system tested only reduced energy requirements by 17%, an increase in heat exchanger area is anticipated to increase these savings to 25%. Applying this figure to the estimated refrigeration and water heating requirements for the whole industry of 2.08×10^8 kWh.year^{-1} results in an energy saving of 0.52×10^8 kWh.year^{-1} or \$2.08 million at 4 cents per kWh. To achieve this level of

TABLE 8:2

The contribution of heat recovery to water heating energy requirements from heat recovery systems cooling 2000 litre of milk from 23°C to 4°C

Heat Recovery Systems	Water Heating Energy (297 litres total volume) ($kWh.day^{-1}$)	Refrigeration Energy (No heat recovery) ($kWh.day^{-1}$)	Gross Heat Recovery ($kWh.day^{-1}$)	Gross Heat Recovered as a Percentage of Water Heating Energy (%)	Nett Heat Recovery ($kWh.day^{-1}$)	Nett Heat Recovered as a Percentage of Water Heating Energy (%)
Water Cooled With Suction Superheater	36.8	16.8	18.63	51	14.4	39
Water Cooled Without Suction Superheater	36.8	16.8	16.20	44	12.0	33
Air Cooled	36.8	19.4	12.80	35	8.6	23

saving, the industry would have to invest $33 million to convert the 16,506 (number of dairy farms from Section 1:1) air cooled refrigeration systems to water cooled heat recovery systems with suction superheating capabilities. The annual return on this investment is calculated to be 6.3%, which is not considered by the dairy industry to be economical for equipment alterations.

While it is not proposed that water cooled heat recovery systems should be installed en masse, it is suggested that farms, with daily milk production in excess of 2000 litres and with water requirements above 300 litres per day, requiring a new refrigeration system (greater then 2.25 kW) would benefit from the installation of a water cooled refrigeration heat recovery system.

CHAPTER 9

CONCLUSIONS

9:1 THE REFRIGERATION SYSTEM

The performance of the refrigeration system varied considerably in response to changing operating conditions, as follows:-

1) Increasing condenser pressure from 6.5 bars to 12 bars reduced C.O.P. from 3.05 to 2.35 for the water cooled system, and increased cooling times to the point where the system failed to meet the cooling regulations.

2) Reducing milk cooling differentials from 19°C to 11°C reduced cooling times by the same proportion but had no significant effect on C.O.P.

3) The additional 0.36 kW in power consumption rate (fan power) for the air cooled system resulted in a C.O.P. range of 2.70 to 2.00, which was 0.35 lower than the water system. This had no effect on cooling times.

4) Receiver pressure below 6.0 bars resulted in incorrect operation of the expansion valve.

In conclusion, to maximise refrigeration energy utilisation, a water cooled system operating at the lowest possible condenser pressure, consistent with the correct operation of the expansion valve, should be selected.

9:2 HEAT RECOVERY

The performance of a heat recovery heat exchanger, installed in the refrigeration system to reduce the energy requirements for water heating, also varied with changing operating conditions, as follows:-

1) Increasing condenser pressure from 6.5 bars to 12 bars increased heat recovery from 4.2 $kWh.day^{-1}.m^{-3}$ to 8.1 $kWh.day^{-1}.m^{-3}$ (a 93% increase) for the water cooled system (compared with a change of 3.8 $kWh.day^{-1}.m^{-3}$ to 6.8 $kWh.day^{-1}.m^{-3}$ (71%) for the air cooled system) and

increased water outlet temperatures from 45°C to 64°C (44%) (compared with 38°C to 55°C (44%) for the air cooled system). The cost of these increases was increased power consumption (up to an extra 2.2 $kWh.day^{-1}.m^{-3}$ (26%) for both systems).

2) Decreasing milk cooling differentials proportionally reduced the total heat recovered, due to shorter cooling times, but had no significant effect on water outlet temperature, because condenser pressures remained constant.

3) The water cooled system recovered up to 1.5 $kWh.day^{-1}.m^{-3}$ (23%) more than the air cooled system, as a result of the extra heat recovered in the secondary heat exchanger.

4) The inclusion of a suction superheater increased heat recovery by 15%, with no significant effect on C.O.P. or cooling times.

9:3 ECONOMIC ASSESSMENT

The rationalisation of the conflict between heat recovery and refrigeration performance was made on the basis of the ratios (Ke and Ke') representing the additional cost of operating the system above a 6.5 bar head pressure, to the savings gained from heat recovery. From this analysis, the conclusions were:-

1) The water cooled system was more efficient (Ke = 0.26) than the air cooled system (Ke = 0.34) at a 12 bar condenser pressure.

2) For a pricing ratio of 1:1, increasing condenser pressure was justified since nett heat recovery increased, whereas, for a 1:3 pricing ratio, nett heat recovery decreased with increasing condenser pressure due to the greater cost of compressor power.

9:4 APPLICATION TO A TOWN SUPPLY FARM

The application of heat recovery systems to a typical town milk supply farm, cooling 2000 litres of milk from 18°C to 4°C and heating 297 litres of water per day, showed that:-

1) The maximum reduction in total energy was achieved from a 12 bar water cooled suction superheated system, with 8.8 $kWh.day^{-1}$ (17%) being saved over the standard system. This represented a saving of $\$135.year^{-1}$ for an energy cost of 4 $cents.(kWh)^{-1}$.

2) For this system, a capital cost of $2000 resulted in a 16.75 year payback period.

3) Increasing heat recovery by increasing cooling times was not possible, since the system would fail to meet the legal maximum cooling time, Increasing heat recovery, by increasing heat exchanger area 2.38 times, produced only a small increase (8%) in monetary savings. This reduced the payback period, for the water cooled suction superheater system, by 3.25 years.

9:5 GENERAL CONCLUSIONS

In general, heat recovery is uneconomical for 2.25 kW refrigeration systems as a result of the relationship between capital costs and heat recovery savings. However, it is expected that larger systems could be economical due to the increase in heat recovery rates for a negligible increase in capital costs.

In all cases, it is recommended that other methods of reducing energy consumption be investigated. These include:-

1) reducing hot water consumption,

2) reducing water temperatures in conjunction with medium or low temperature detergents, and

3) reducing water heating cylinder heat losses by improved insulation, or by fitting time clocks.

In many farm situations these methods are the most cost effective. Therefore, heat recovery systems should only be considered in conjunction with other alternatives available.

APPENDIX A2

DERIVATION OF UBBELS HEAT RECOVERY LIMITING EQUATION FOR FULL CONDENSING, COIL-IN-TANK, HEAT RECOVERY SYSTEM

ased on a constant milk load (M) the power required for a heat ecovery refrigeration system is Q_c. The power required for the same ystem with an air cooled condenser without heat recovery is Q_{ca}. From which:

$$\text{Amount of heat recovery} = Q_{HR} \quad (kJ)$$

$$\text{Amount of power increase due to heat recovery} = Q_c - Q_{ca} \quad (kJ)$$

$$\text{Heat load of milk} = M \quad (kJ)$$

$$\therefore \ COPR_c = \frac{M}{Qc}$$

$$COPR_{ca} = \frac{M}{Q_{ca}}$$

$$COPH = \frac{HR}{Q_c}$$

$$\text{Ratio of additional power to heat recovery} = Ke$$

$$\text{i.e.,} \quad Ke = \frac{Q_c - Q_{ca}}{Q_{HR}} \qquad \text{.....Eqn A2:1}$$

$$\text{but} \quad Q_c = \frac{M}{COPR_c}$$

$$Q_{ca} = \frac{M}{COPR_{ca}}$$

$$Q_{HR} = Q_c.COPH$$

$$\text{Now} \quad Ke = \frac{\dfrac{M}{COPR_c} - \dfrac{M}{COPR_{ca}}}{Q_c . COPH}$$

$$= \frac{M \left(\dfrac{1}{COPR_c} - \dfrac{1}{COPR_{ca}} \right)}{Q_c \qquad COPH}$$

$$\text{but} \quad \frac{M}{Q_c} = COPR_c$$

$$\therefore \quad Ke = \frac{COPR_c \left(\dfrac{1}{COPR_c} - \dfrac{1}{COPR_{ca}} \right)}{COPH}$$

$$\therefore \quad Ke = \frac{\left(1 - \dfrac{COPR_c}{COPR_{ca}} \right)}{COPH}$$

APPENDIX A3

THEORETICAL ANALYSIS OF A COMPLETE CONDENSING HEAT RECOVERY SYSTEM

A3:1 ASSUMPTIONS

A3:2 CALCULATION OF TEMPERATURES

A3:3 CALCULATION OF WATER FLOW RATES

A3:4 FINAL ARRANGEMENT

A complete condensing heat recovery system was required to condense the refrigerant vapour and heat the specified quantity of water to a reasonable temperature (approximately $60^{o}C$).

A3:1 ASSUMPTIONS

Assuming that requirements of the system are:-

1) A water outlet temperature = $60^{o}C$
2) An effectiveness ratio = 0.8
3) A water inlet temperature = $16^{o}C$
4) A refrigerant (Freon 12) flow rate = $0.082\ kg.s^{-1}$
5) A condensing temperature = $34^{o}C$ @ 8.5 (bar abs.)

then the refrigerant superheated vapour temperature and water flow rates could be determined.

A3:2 CALCULATION OF TEMPERATURES

Calculation of the temperature leaving the condensing region (Figure A3:1) is by:-

$$E_c = 0.8 = \frac{t_{woc} - t_{wi}}{t_{ric} - t_{wi}} \qquad .....\text{Eqn A3:1}$$

where E_c = Effectiveness ratio

t_{woc} = temperature of the water leaving the condensing region (^{o}C)

t_{wi} = water inlet temperature (^{o}C)

t_{ric} = temperature of the refrigerant entering the condensing region (^{o}C)

Rearranging Eqn A3:1 gives:-

$$t_{woc} = 0.8\ (34 - 16)^{o}C + 16^{o}C$$

$$= 30.4^{o}C$$

Using this result for t_{woc} the superheated vapour temperature required to heat the water to $60^{o}C$ can be calculated from:-

$$E_c = 0.8 = \frac{t_{wosh} - t_{woc}}{t_{rish} - t_{woc}} \qquad \text{.....Eqn A3:2}$$

where t_{wosh} = water outlet temperature from superheat region (^{o}C)

= $60^{o}C$

t_{rish} = refrigerant vapour inlet temperature to the superheat region (^{o}C)

Rearranging Eqn A3:2 gives:-

$$t_{rish} = \frac{(60 - 30.4)^{o}C}{0.8} + 30.4^{o}C$$

$$= 67.5^{o}C$$

A3:3 CALCULATION OF WATER FLOW RATES

The water flow rate required to remove the heat in the condensing region can be calculated from the equation:-

$$E_r = M_w \times Sp\ Ht \times \Delta t \qquad \text{.....Eqn A3:3}$$

where E_r = Heat in refrigerant to be removed ($kJ.s^{-1}$)

M_w = Water mass flow rate ($kg.s^{-1}$))

$Sp\ Ht$ = Specific heat of water ($kJ.kg^{-1}.^{o}C^{-1}$)

Δt = Water temperature difference (^{o}C)

The value of E_r in Eqn A3:3 can be calculated from the refrigerant mass flow rate (M_r) and the enthalpy change determined from the Freon 12 chart. Substituting the appropriate values into Eqn A3:3 gives:-

$$E_r = 130\ (kJ.kg^{-1}) \times 0.082\ (kg.s^{-1})$$

$$= M_w\ (kg.s^{-1}) \times 4.186\ (kJ.kg^{-1}.^oC^{-1})$$

$$\times\ (30.4 - 16)^oC$$

from which:-

$$M_w = 0.177\ kg.s^{-1}$$

Using Eqn A3:3 for the superheated vapour region and substituting values for temperatures, refrigerant flow rate and enthalpy changes gives:-

$$E_r = 0.082\ kg.s^{-1} \times 27\ kJ.kg^{-1}$$

$$= M_w\ (kg.s^{-1}) \times 4.186\ (kJ.kg^{-1}.^oC^{-1})$$

$$\times\ (60 - 30.4)^oC$$

From which the water flow rate required to cool the superheated vapour is:-

$$M_w = 0.018\ kg.s^{-1}$$

A3:4 FINAL ARRANGEMENT

On the basis of these flow rates the minimum flow rate required to condense the refrigerant is 0.177 $kg.s^{-1}$. For a single heat exchanger the outlet temperature can be determined from:-

$$E_r = (130 + 27)\ (\mathrm{kJ.kg^{-1}}) \times 0.082\ (\mathrm{kg.s^{-1}})$$

$$= 0.177\ (\mathrm{kg.s^{-1}}) \times 4.186\ (\mathrm{kJ.kg^{-1}.^{o}C^{-1}})$$

$$\times\ (t_{wosh} - 16)\ ^{o}\mathrm{C}$$

from which:-

$$t_{wosh} = 33.4^{o}\mathrm{C}$$

Since large quantities of low temperature water were not required then a single heat exchanger operating as a complete condensing heat recovery system was not a viable alternative.

APPENDIX A4

EXPERIMENTAL EQUIPMENT, METHODS AND DATA ANALYSIS

:1 ARRANGEMENT AND WIRING DIAGRAMS

(See following Figures A4:1, A4:2 and A4:3.)

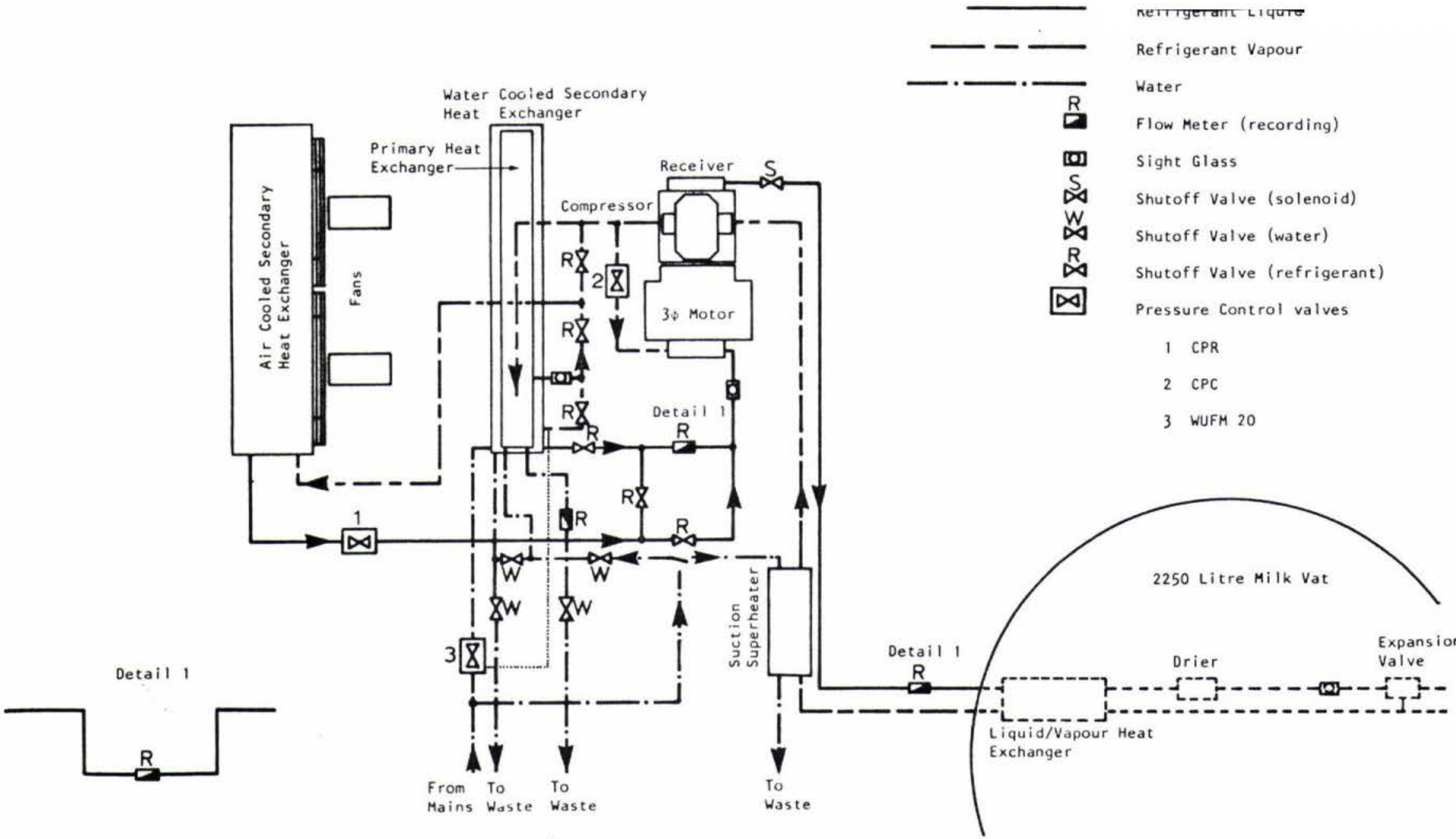


FIGURE A4:1

Arrangement diagram of experimental plant

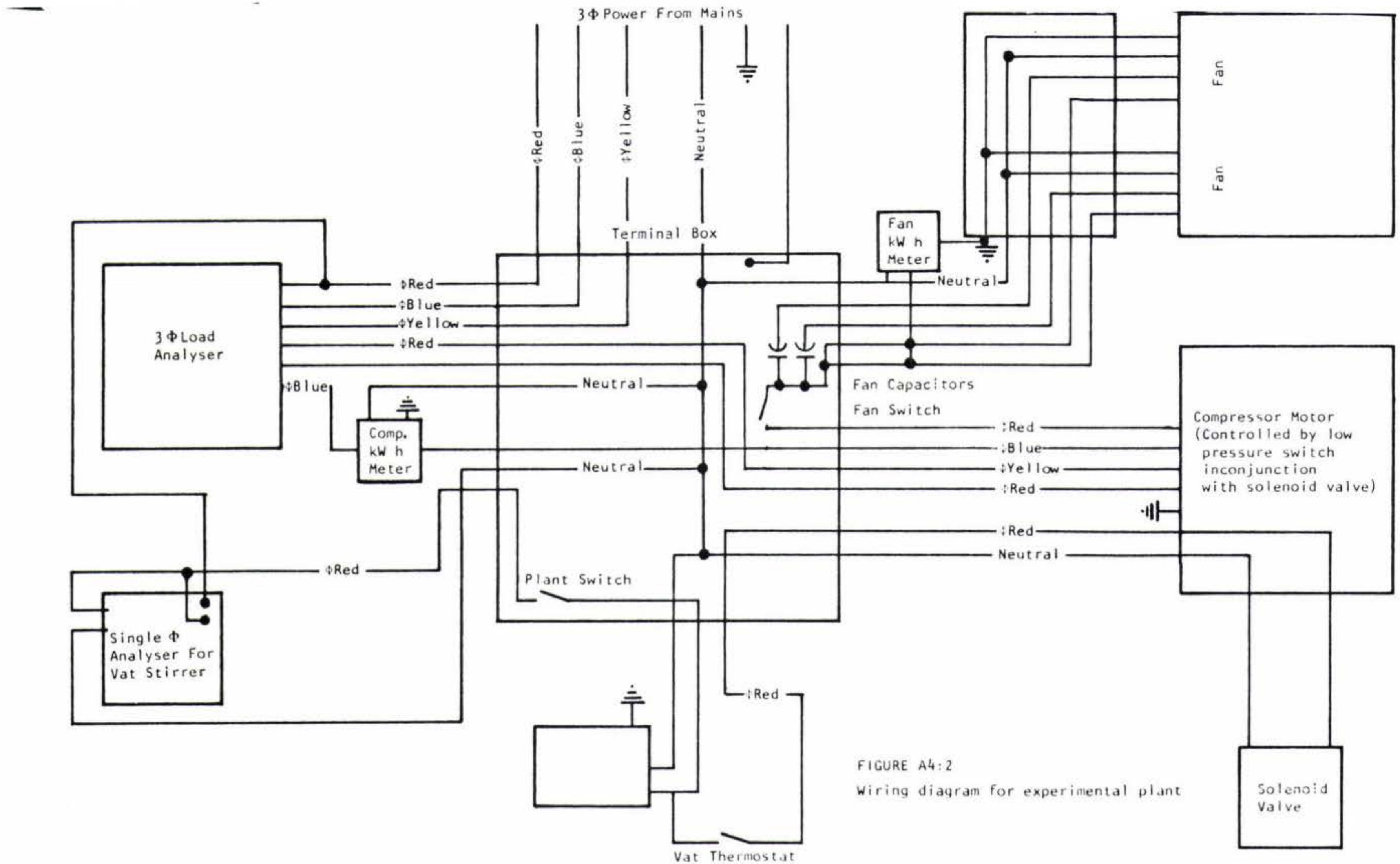


FIGURE A4:2
Wiring diagram for experimental plant

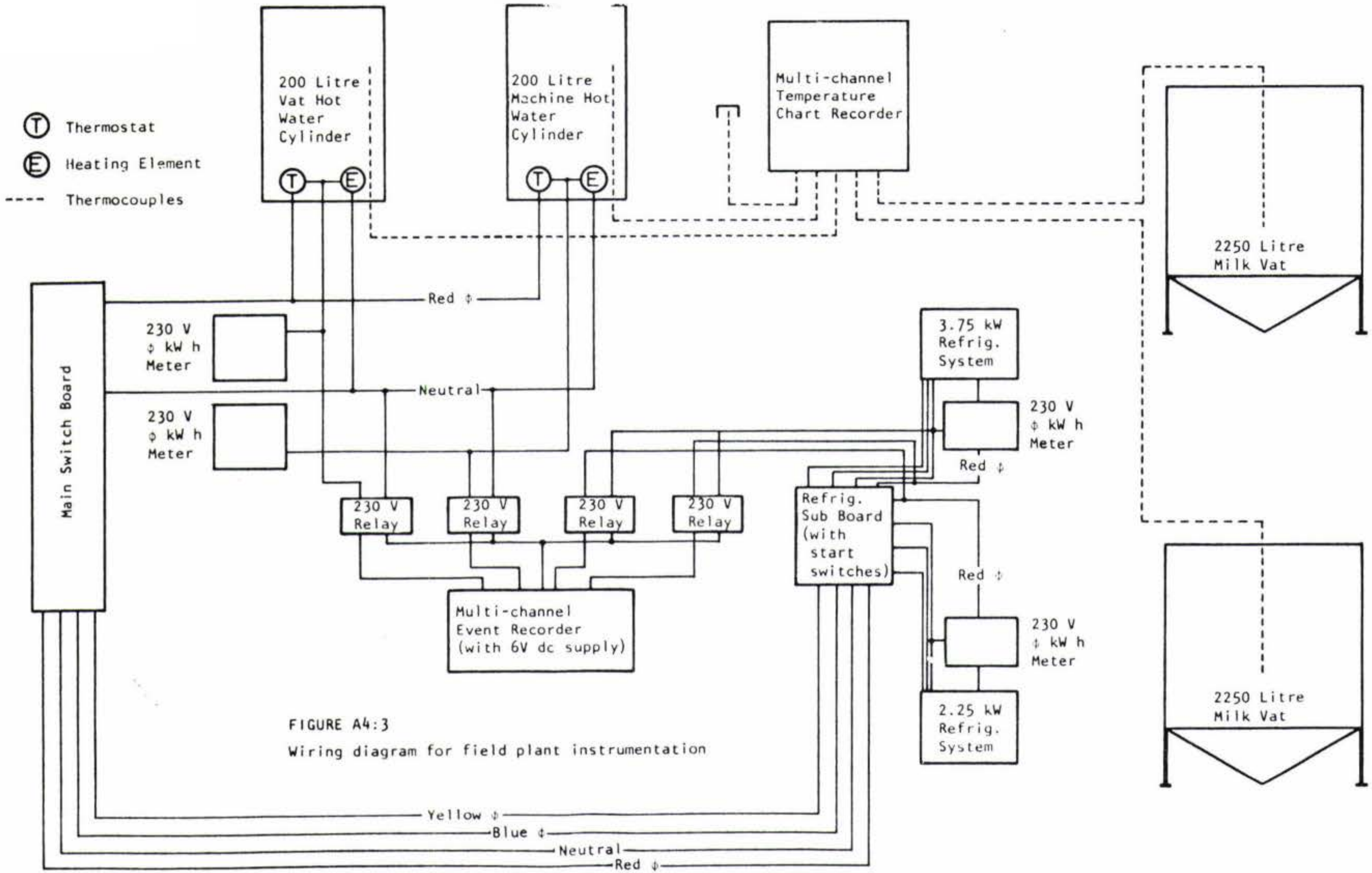


FIGURE A4:3

Wiring diagram for field plant instrumentation

A4:2 TEST PRIMARY HEAT EXCHANGER DESIGN

The design parameters for the primary heat exchanger were

1) turbulent flow conditions for a water flow of 0.625 litres/min

2) turbulent flow conditions for the refrigerant

3) construction from currently existing technology.

A4:2.1 Water Tube

Table A4:1 gives the Reynolds No. for three sizes of tube which are commercially available. Since the 4 mm I.D. tube was not available with spiral finning and the 10 mm tube gave Reynolds No.s in the laminar region, then the only remaining alternative was the 7 mm I.D. stainless tube.

A4:2.2 Cores

To improve the heat transfer characteristics (by increasing water velocity) the use of cores was investigated. The only available sizes of stainless core had diameters of 6.4 mm, 4.7 mm and 3.2 mm. The 6.4 mm diameter core would have resulted in a flow path which had potential blockage characteristics and the 3.2 mm diameter core would not achieve the desired flow pattern. Therefore, the 4.7 mm core was selected.

A4:2.3 Spiral Fins

Spiral finning 9 mm high was available in three pitches; 158 fins/m (4 F/inch), 236 fins/m (6 F/inch) and 314.88 fins/m (8 F/inch). To ensure that a high degree of turbulence was generated, while paying due consideration to headloss, the 158 fins/m pitch was selected.

TABLE A4:1

Reynolds Number for three commercially available tubes (without cores) and a water flow rate of 0.625 $l.min^{-1}$

Tube Diameter (I.D.) (m)	Cross Sectional Area (m^2)	Water Flow Rate ($l.min^{-1}$)	Velocity ($m.s^{-1}$)	Mean Water Temperature Rise (^oC)	Viscosity ($N.S.m^{-2}$)	R.E.
0.010	0.079×10^{-3}	0.625	0.132	40	0.654×10^{-3}	2016
0.007	0.038×10^{-3}	0.625	0.274	40	0.654×10^{-3}	2934
0.004	0.013×10^{-3}	0.625	0.801	40	0.654×10^{-3}	4900

A4:2.4 Outer Tube

To force the refrigerant vapour to principally flow between the fins, so that a high film coefficient was developed, an outer tube giving minimum fin tip clearance was selected. The only commercially available tube meeting this requirement had an O.D. of 31.8 mm and an I.D. of 30.5 mm.

A4:2.5 Connections

Refrigerant connections were 15.875 mm O.D. flared refrigeration connectors cut in half and welded to the outside of the outer tube. Water connections were standard 9.25 mm flared nuts.

A4:2.6 Insulation

Standard polyurethane tube insulation ('Armaflex') was used. This insulation had a thermal conductance of 1.59×10^{-3} $kW.m^{-2}.^{o}C^{-1}$.

A4:2.7 Length

The length of the test heat exchanger was set at 1 m. This length resulted in a water surface area of 0.027 m^2 and a refrigerant surface area of 0.28 m^2 (fins plus pipe). The final arrangement is presented in Figure A4:4.

A4:3 INSTRUMENTATION CALIBRATION CURVES AND ERRORS

A4:3.1 Temperature

Copper Constantan thermocouples, digital volt meters and the thermocouple rotary switch were calibrated against an ice bank and boiling water. Since the ice bank, by definition, is at $0^{o}C$ and an ice reference was used then there is no error at this point. The error at the boiling end of the range was limited by the reading accuracy of the mercury in glass thermometer. This was $\pm$ $0.5^{o}C$.

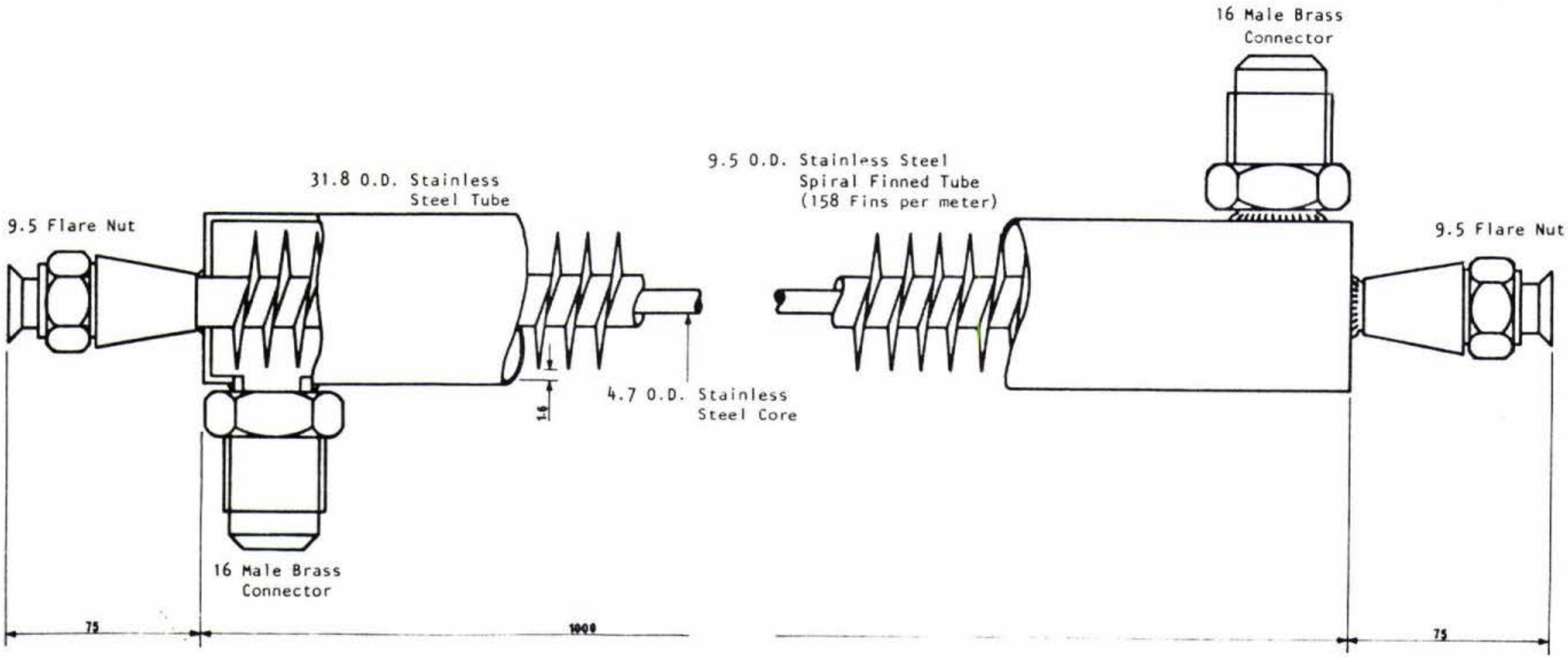


FIGURE A4:4

Construction drawing of test primary heat exchanger

The digital volt meter had a $4^1/2$ figure read out, resulting in a millivolt accuracy of $\pm$ 0.005 mV.

Using these two points the mean slope of the calibration curve was $23.36 \pm 0.12\ ^{o}C.mV^{-1}$. At $100^{o}C$ this slope resulted in a percentage error of

$$\text{Temperature } (^{o}C) = (23.36 \pm 0.12).mV$$

$$= (23.36 \pm 0.5\%)\ mV$$

A4:3.2 Refrigerant Flow

The relationship between the number of pulses and volume for the receiver flow meter from the manufacturer's calibration data was 1979 pulses.$l^{-1} \pm 6\%$.

Three checks on the variation in the "Racal Counter" were made by counting pulses for 1 second and 10 second intervals.

1) 20 consecutive 10 sec counts $S = 0.7406$ Hz

2) 10 consecutive 10 sec counts $S = 0.826$ Hz

3) 19 consecutive 1 sec counts $S = 0.496$ Hz

$$\therefore \left.\begin{array}{ll} 1) & = 67.8 \pm 0.7 \text{ Hz} \\ 2) & = 67.2 \pm 0.8 \text{ Hz} \\ 3) & = 66.6 \pm 0.5 \text{ Hz} \end{array}\right\} 4030 \pm 40 \text{ pulses.min}^{-1}$$

$$\Rightarrow \text{pulses.min}^{-1} \pm 1\%$$

The conversion equation from pulse frequency to flow rate is given by:-

$$\text{Flow rate } (\text{l.min}^{-1}) = \frac{\text{pulses.min}^{-1}}{\text{pulses.l}^{-1}}$$

$$= \text{Hz} \pm 1\% \times \frac{60}{1979 \pm 0.6\%}$$

$$= \text{Hz} \pm 1\% \times 0.030321 \pm 0.6\%$$

$$= 0.030321 \times \text{Hz} \pm (0.01^2 + 0.006^2)^{1/2}$$

$$= (0.0303 \times \text{Hz}) \pm 1.2\%$$

The evaporator flow meter was calibrated against the receiver flow meter, from which the calibration equation was:-

$$\text{Flow rate } (\text{l.min}^{-1}) = (0.0289 \times \text{Hz}) \pm 1.2\%$$

A4:3.3 Primary Water Flow

Frequency for a primary water flow rate of 0.625 l.min^{-1}

$$= 9.4 \text{ Hz}$$

$$= 0.0665 \text{ l.min}^{-1}/\text{Hz}$$

$$\text{Error in counter} = \pm 0.05 \text{ Hz}$$

$$= \pm 0.2\% \text{ at } 25 \text{ Hz } (1.6 \text{ l.min}^{-1})$$

$$\text{Error in stopwatch} = \pm 0.5 \text{ secs}$$

$$= \pm 0.7\% \text{ at } 74 \text{ seconds } (1.6 \text{ l.min}^{-1})$$

$$\text{Error in measuring cylinder} = \pm 0.005 \text{ litres}$$

$$= \pm 0.25\% \text{ for } 2.00 \text{ litres}$$

Therefore, error in flow rate calculation

$$\text{Flow rate } (\text{l.min}^{-1}) = (0.0665 \times \text{Hz}) \pm (0.7^2 + 0.25^2)^{1/2}$$

$$= (0.0665 \times \text{Hz}) \pm 0.74\% \text{ at } 1.6 \text{ l.min}^{-1}$$

$$= (0.0665 \times \text{Hz}) \pm 0.7\%$$

A4:3.4 Direct Reading Instrumentation

The instruments which did not require the development of conversion equations are summarised in Tables A4:2 and A4:3. In all cases the accuracy to which the instrument was read was taken as the error for the value of the variable. Instrument checks were carried out on the low pressure Bourdon gauges and the Honeywell chart recorder, and in all cases calibration errors were considered insignificant compared to the reading error.

A4:4 EXPERIMENTAL VARIABLES, DATA SHEETS AND COMPUTER PROGRAMME (EXPERIMENT II)

A4:4.1 List of Measurements for the Experiment Plant

A4:4.1.1 List of Independent and Dependent Variables

1) Independent Variables

   (i) Condenser pressure
   (ii) Primary heat exchanger water flow rate
   (iii) Primary heat exchanger refrigerant flow (Experiment I only)
   (iv) Refrigeration load (total)
   (v) Milk inlet temperature
   (vi) Milk final temperature

2) Dependent Variables

   (i) Cooling rates
   (ii) Power consumption
   (iii) C.O.P. refrigeration
   (iv) Condenser water temperature
   (v) Refrigerant flow rate
   (vi) Temperatures
   (vii) Secondary water cooled heat exchanger water flow rate
   (viii) Overall thermal conductance (U)

TABLE A4:2

Summary of instrumentation used on experimental plant

Variable	Instruments	Reading Accuracy
Pressures		
a) Condenser	Bourdon gauges (0-1400 kPa)	± 2.5 kPa
b) Evaporator	Bourdon gauges (2 x 0-1000 kPa)	± 5 kPa
	(1 x -100 - 1000 kPa)	± 5 kPa
Compressor Power Consumption		
a) Direct Reading	Cambridge 3 phase load analyser	± 2.5%
b) Calculation	Volts, Current/phase and Power Factor	± 2.7%
Total Compressor and Fan Power Consumption	kWh meters	± 0.5%
Vat Stirrer Power Consumption	Cambridge Single Phase load analyser	± 0.5%
Secondary Heat Exchanger Flow Rate	Weighed volume and stopwatch	± 1.6%

TABLE A4:3

Summary of instrumentation used on field plant

Variable	Instruments	Reading Accuracy
Temperature	Honeywell multichannel chart recorder, copper Constantan thermocouples	$\pm$ 1^{o}C
Water Volumes	'Kent' water meters	$\pm$ 0.5 litres
Power Consumption		
a) Total	3 phase kWh meter	$\pm$ 0.5%
b) Water Heating and Refrigeration	Single phase kWh meters	$\pm$ 1.0%
Cooling and Heating Times	Multichannel event recorder	$\pm$ 1 minute

A4:4.1.2 List of Measurements

1) Temperatures

Refrigerant

T_1 Compressor inlet

T_2 Compressor outlet

T_3 Primary heat exchanger inlet

T_4 Primary heat exchanger outlet

T_5 Water cooled secondary heat exchanger inlet

T_6 Air cooled secondary heat exchanger inlet

T_7 Water cooled secondary heat exchanger outlet

T_8 Air cooled secondary heat exchanger outlet

T_9 Receiver inlet

T_{10} Receiver outlet

T_{11} Expansion valve inlet

T_{12} Evaporator inlet

T_{13} Evaporator outlet

Water

T_{14} Water cooled secondary heat exchanger inlet

T_{15} Water cooled secondary heat exchanger outlet

T_{16} Primary heat exchanger inlet

T_{17} Primary heat exchanger outlet

Miscellaneous

T_{18} Ambient

T_{19} Compressor head

T_{20} Milk inlet

T_{21} Milk-vat

2) Refrigerant Pressures

P_1 Compressor inlet

P_2 Compressor outlet

P_3 Secondary inlet

P_4 Receiver

P_5 Evaporator inlet

P_6 Evaporator outlet

3) Flow Rates

R_1 Primary water

R_2 Secondary water

R_3 Milk flow

R_4 Refrigerant (receiver)

R_5 Refrigerant (evaporator)

4) Power

K_1 Compressor kWh

K_2 Compressor Watts

K_3 Compressor Amps

K_4 Compressor P.F.

K_5 Compressor Volts

K_6 Fan kWh

K_7 Stirrer Watts

A4:4.2 Sample Calculations and Data Sheets

The calculated variables presented in Chapter 6 are calculated below for one set of measurements taken during the W12 run. These measurements are presented in Table A4:4, as an example of the computer data collection sheets. For Freon properties, table values are quoted in brackets for comparison.

A4:4.2.1 Temperatures

Using the temperature conversion equation in Appendix A4:3.1, and T_1 in mV (Table A4:4), as an example:-

$$T_1\ (^{o}C) = 23.36 \times 0.57\ (mV)$$

$$= 13.3^{o}C$$

from which $T_1 = 286.3^{o}K$

A4:4.2.2 Pressures

Pressures P_1 to P_6 were converted from kPa to bar abs by:-

$$P\ (\text{bar abs}) = \left(\frac{P\ (kPa)}{100}\right) + 1$$

e.g. $P_1\ (\text{bar abs}) = \left(\frac{150}{100}\right) + 1 = 2.50$

A4L4.2.3 Refrigeration Effect (R.E.)

From Eqn 2:5,

$$R.E. = (h_1 - h_{3b})\ \ kJ.kg^{-1}$$

where h_1 = specific enthalpy at the evaporator outlet

TABLE A4:4

Sample computer data sheets

Power Consumption Results Sheet

Comp. kW h K_1	Comp. Watts K_2	Comp. Amps K_3	Comp. Volts K_5	Stirrer Watts K_7	P.F. K_4	Fan kW h K_6	Time Since Start
244.60	2.80	6.25	410	97.0	0.63	44.78	295

Pressure And Flow Rate Results Sheet

Pressures (kPa)						Flow Rates ($l.min^{-1}$ and Hz)						
P_1	P_2	P_3	P_4	P_5	P_6	Refrig. (rec.) R_4	Second. Water R_2	Primary Water R_1		Refrig. (evap.) R_5	Freon Total1 (rec.)	Freon Total2 (evap)
160	1300	1210	615	220	180	67.6	4.56	9.2	0.0	71.6		

Thermocouple Results Sheet 1

T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}
0.57	4.19	4.12	2.45	2.41	0.97	0.86	0.78	0.86	0.88	0.17	0.06	0.36	0.71	1.51

Thermocouple Results Sheet 2

T_{16}	T_{17}	T_{18}	T_{19}	T_{20}	T_{21}
1.50	2.84	3.87	0.66	0.42	0.72

4 4

h_{3b} = specific enthalpy at the expansion valve inlet.

The calculation of h_1 requires that the specific enthalpy of the superheated vapour (h_{SH}) be calculated for T_{13} and P_6 using Eqn A4:11. This equation requires that values of h_f (Eqn A4:9), V_f (Eqn A4:7), h_{fg} (Eqn A4:10) and C_{VAP} (Eqn A4:5) be determined using T_{13} and P_6 and the saturation temperature, calculated from Eqn A4:3. The results of these calculations were:-

T_{13} = $281.4^{o}K$

P_6 = 2.80 bar abs

T_{SAT} = $270.1^{o}K$
for P_{SAT} = 2.80 bar abs (270)

V_f = $7.059 \times 10^{-4}\ m^3.kg^{-1}$
for T = $270.1^{o}K$ (0.71099×10^{-4})

h_f = $33.39\ kJ.kg^{-1}$
for T = $270.1^{o}K$ (33.29)

h_{fg} = $153.74\ kJ.kg^{-1}$
for P_{SAT} = 2.80 bar (152.955)

C_{VAP} = $0.648\ kJ.kg^{-1}.^{o}K^{-1}$
for P_{SH} = 2.80 bar

The final value of h_{SH} is given by:-

$$h_{SH} = 33.39 + 153.74 + 0.648\ (T_{13} - T_{SAT})$$

$$= 194.45\ kJ.kg^{-1}$$

The determination of h_{3a} was from Eqn A4:9 and Eqn A4:7 for the expansion valve inlet temperature (T_{11}).

From Table A4:4, T_{11} = 291.0°K and substitution into Eqn A4:7 yields a value for V_f of:-

$$V_f = 7.421 \times 10^{-4}\ m^3.kg^{-1}$$
$$\text{for } T_{SAT} = 291.0^{o}K\ (7.4846\times10^{-4})$$

and substitution into Eqn A4:9 gives a value for h_f of:_

$$h_f = 52.86\ kJ.kg^{-1}$$
$$\text{for } T_{SAT} = 291.0^{o}K\ (52.958)$$

The difference between h_{SH} and h_f is the refrigeration effect, i.e.,

$$R.E. = (194.45 - 52.86)$$

$$= 141.59\ kJ.kg^{-1}$$

A4:4.2.4 Refrigerant Flow

The flow rate in $l.min^{-1}$ was calculated using the conversion equations developed in Appendix A4:3.2. Converting frequency values for R_4 and R_5 from Table A4:4 gives:-

$$\text{Receiver flow } (R_4) = 67.6\ H_2 \times 0.0303$$
$$= 2.04\ l.min^{-1}$$

$$\text{Evaporator flow } (R_5) = 71.6\ H_2 \times 0.0289$$
$$= 2.07\ l.min^{-1}$$

Refrigerant mass flows were calculated from Eqn 4:10, in which V_f (Eqn A4:7) is calculated using T_9 (293.1°K) for the receiver flow meter and T_{10} (293.6°K) for the evaporator flow meter. Results of these calculations were:-

Receiver mass flow (M_{r1}) = 0.0456 $kg.s^{-1}$

Evaporator mass flow (M_{r2}) = 0.0462 $kg.s^{-1}$

A4:4.2.5 Instantaneous Compressor Power Consumption (I.C.P.C.)

Using Eqn A4:12 and the values of:-

K_3 = 6.25 $Amps.phase^{-1}$

K_4 = 0.63 (P.F.)

and K_5 = 410 Volts

results in I.C.P.C. being equal to 2.80 kW.

A4:4.2.6 Coefficient of Performance

Substituting the values of:-

M_{r2} = 0.0462 $kg.s^{-1}$

R.E. = 141.59 $kJ.kg^{-1}$

and I.C.P.C. = 2.80 kW

into Eqn 4:11 results in a C.O.P. value of 2.34.

A4:4.2.7 Primary Heat Exchanger Water Flow Rates

Using the conversion equation in Appendix A4:3.3, and the value for R_1 of 9.2 Hz from Table A4:4, results in a value for primary water flow rate of 0.612 $l.min^{-1}$. The mass flow rate (M_w) was calculated to be 0.0102 $kg.s^{-1}$.

A4:4.2.8 Primary Heat Exchanger Heat Flow Rate

Primary heat exchanger heat flows were calculated by two independent methods. Firstly, the rate of heat gain was calculated using Eqn 4:2. Taking values of:-

$$T_{16} = 308.0^{o}K$$

$$T_{17} = 339.3^{o}K$$

$$M_w = 0.0102 \text{ kg.s}^{-1}$$

$$\text{Sp Ht} = 4.186 \text{ kJ.kg}^{-1}.^{o}C^{-1}$$

gives a rate of heat flow into the water, from Eqn 4:2), of 1.34 kW.

A check calculation for the heat lost by the refrigerant vapour was made, using the following expression:-

$$Q_r = M_r \times (h_{SH1} - h_{SH2})$$

Taking values from Table A4:4 of:-

$$\left.\begin{aligned} T_3 &= 369.24^{o}K \\ P_2 &= 14.0 \text{ bar abs} \end{aligned}\right\} h_{SH1} \text{ calculations}$$

$$\left.\begin{aligned} T_4 &= 330.23^{o}K \\ P_3 &= 13.1 \text{ bars abs} \end{aligned}\right\} h_{SH2} \text{ calculations}$$

$$M_{r1} = 0.0462 \text{ kg.s}^{-1}$$

Q_r was calculated to be 1.38 kW.

A4:4.2.9 Overall Thermal Conductance (U)

Overall thermal conductance was calculated using Eqn 4:1. Taking values from Table A4:4 of:-

$$T_3 = 369.2^{o}K$$

$$T_4 = 330.2^{o}K$$

$$T_{16} = 308.0^{o}K$$

$$T_{17} = 339.3^{o}K$$

the log mean temperature difference was calculated to be 25.8°C. The area of the heat exchanger was 0.84 m^2. Substituting these values, and the value of Q_W of 1.34 kW, into Eqn 4:1 yields a value for U of 0.062 $kW.m^{-2}.^{o}C^{-1}$.

```
ANALYSIS:PROCEDURE;
       /**********************************************************************
        *                                                                    *
        *                                                                    *
        *                REFRIGERATION DATA ANALYSIS PROGRAM                 *
        *                                                                    *
        *  Aim:-     The aim of this program is to process the raw data  for *
        *            the  runs  conducted  on the  experimental  plant  from *
        *            which results of power, flow, enthalpies, COPs and heat *
        *            recovered are written to output files for checking  and *
        *            analysis by MINITAB.                                    *
        *                                                                    *
        *  Input:-   Data from the four data sheets(Appendix A4:4.4)are read *
        *            into the DATA structure through the file DATAIN.        *
        *                                                                    *
        *  Method:-  Processing the data by calculation to give:             *
        *                1) temperatures and flows from conversion equations *
        *                2) refrigerant enthalpies                           *
        *                3) compressor and fan power results                 *
        *                4) primary heat exchanger heat flows U and Ec values *
        *                5) COPs                                             *
        *                                                                    *
        *  Output:-  Results  are  printed  to  5  files (#1-#5) for MINITAB *
        *            analysis . A complete  set  of results is printed in a  *
        *            table format for printing (file TABLE).                 *
        *                                                                    *
        *                                                                    *
        **********************************************************************/


                         /*****************************
                          *                           *
```

```
                    *  DECLARATION OF VARIABLES   *
                    *                             *
                    *******************************/

DCL 1 DATA(20),/* Data structure -holds all test data*/
      2 POWER,
        3 COMPKWH FIXED DECIMAL(5,2),  /* compressor kW h meter readings & running total*/
        3 WATT FIXED DECIMAL(3,2),     /* compressor Watt meter reading*/
        3 WATT_CAL FIXED DECIMAL(3,2), /* compressor calculated Watts*/
        3 AMP FIXED DECIMAL(3,2),      /* compressor amps/phase*/
        3 VOLT FIXED DECIMAL(3),       /* compressor volts*/
        3 STWATT FIXED DECIMAL(4,1),   /* stirrer Watts*/
        3 PF FIXED DECIMAL(2,2),       /* compressor power factor*/
        3 FANKWH FIXED DECIMAL (5,2),  /* fan kW h readings & running total*/
      2 PRESSURE(6) FIXED DECIMAL(8,4),/* compressor pressures
                                          PRESSURE(1) = comp inlet
                                          PRESSURE(2) = comp outlet
                                          PRESSURE(3) = condenser
                                          PRESSURE(4) = receiver
                                          PRESSURE(5) = evaporator inlet
                                          PRESSURE(6) = evaporator outlet*/
      2 FLOWS(6) FLOAT DECIMAL,        /* flow rates
                                          FLOWS(1) = refrig receiver(l/min)
                                          FLOWS(2) = water condenser(l/min)
                                          FLOWS(3) = water primary(l/min)
                                          FLOWS(4) = refrig receiver(kg/s)
                                          FLOWS(5) =        evaporator(l/min)
                                          FLOWS(6) =        evaporator(kg/s)*/
      2 TEMPS(21) FIXED DECIMAL(5,2),  /* temperatures
                                          TEMPS(1) = Refrig. comp inlet
                                          TEMPS(2) =         comp outlet
                                          TEMPS(3) =         primary inlet
                                          TEMPS(4) =         primary outlet
                                          TEMPS(5) =         water cond. inlet
```

```
                                    TEMPS(6) =        air cond. inlet
                                    TEMPS(9) =        water cond. outlet
                                    TEMPS(8) =        air cond. outlet
                                    TEMPS(9) =        receiver inlet
                                    TEMPS(10)=        receiver outlet
                                    TEMPS(11)=        expansion valve inlet
                                    TEMPS(12)=        evaporator inlet
                                    TEMPS(13)=        evaporator outlet
                                    TEMPS(14)= Water  cond. inlet
                                    TEMPS(15)=        cond. outlet
                                    TEMPS(16)=        primary inlet
                                    TEMPS(17)=        primary outlet
                                    TEMPS(18)= compressor head
                                    TEMPS(19)= milk inlet
                                    TEMPS(20)= vat temp
                                    TEMPS(21)= ambient */
2 FREONTOT(2) FIXED DECIMAL(5), /* running totals of freon vol. (2 flow meters)*/
2 TIME FIXED DECIMAL(4),        /* total time since start*/
2 ENTHALPIES,
  3 COMPH FIXED DECIMAL(6,3),   /* enthalpy change during compression*/
  3 DSH FIXED DECIMAL(6,3),     /* enthalpy level of Delivery SuperHeat*/
  3 CONDH FIXED DECIMAL(6,3),   /* enthalpy change during CONDensation*/
  3 SCCOND_RECH FIXED DECIMAL(6,3),/* SubCooled CONDenser RECeiver enthalpy difference*/
  3 REC_EXPH FIXED DECIMAL(6,3),/* RECeiver EXPansion valve enthalpy difference*/
  3 EVAPH FIXED DECIMAL(6,3),   /* enthalpy change across the evaporator = R.E.*/
  3 CEVAPH FIXED DECIMAL(6,3),  /* Compressor - EVAPorator outlet enthalpy difference*/
2 COP(2) FIXED DECIMAL(6,4),    /* actual and theoretical C.O.P.s*/
2 COMP_EFF FIXED DECIMAL(4,3),  /* compressor efficiency*/
2 PRIMEXCHANGER,
  3 EXCHQ(2) FIXED DECIMAL(6,3),/* EXCHanger heat flows(Q)  1)water Q
                                                            2)refrigerant Q */
  3 LMTD FIXED DECIMAL(6,3),    /* Log Mean Temperature Difference */
  3 UA FIXED DECIMAL(7,4),      /* overall thermal conductance (U) x Area */
  3 CWAT FIXED DECIMAL(7,5),    /* specific heat (C) x mass flow of WATer */
  3 CREF FIXED DECIMAL(7,5),    /* specific heat (C) x mass flow of REFrigerant */
```

```
        3 NTU FIXED DECIMAL(6,3),       /* Number of Transfer Units */
        3 EC FIXED DECIMAL(6,4);        /* Effectiveness Coefficient*/

DCL STOP_TIME(4) FIXED DECIMAL(3,0);  /* stop times for 7oC and 4oC cooling*/
DCL KWH7(4) FIXED DECIMAL(5,2);       /* comp kW h and fan kW h for 7oC stop times*/
DCL KWH4(4) FIXED DECIMAL(5,2);       /* comp kW h and fan kW h for 4oC stop times*/
DCL FILENAME(8) CHAR(80) VAR;         /* array for interactive naming of files*/
DCL FREONTOTS(8) FIXED DECIMAL(3,0);  /* freon totals (2 flow meters) for each stop time*/
DCL FILLTIMES(2) FIXED DECIMAL(3,0);  /* vat loading times(800 and 1200 litres)*/
DCL FIRSTKWH(4) FIXED DECIMAL(5,2);   /* initial kW h meter readings for each start up*/
DCL (I,J,Q1,Q2,C,CMIN,SCH) FIXED DECIMAL(7,5);  /* loop counters and constants etc. */
DCL NROWS FIXED DECIMAL;              /* Number of ROWS of data for each run*/

                    /******************************
                    *                             *
                    *   DECLARATION OF FILES      *
                    *                             *
                    ******************************/

DCL TABLE FILE STREAM OUTPUT;         /* table of processed results*/
DCL RESULTS FILE STREAM OUTPUT;       /* combined file of the next 5 files-checking*/
DCL #1 FILE STREAM OUTPUT;            /* output file for MINITAB input*/
DCL #2 FILE STREAM OUTPUT;            /* output file for MINITAB input*/
DCL #3 FILE STREAM OUTPUT;            /* output file for MINITAB input*/
DCL #4 FILE STREAM OUTPUT;            /* output file for MINITAB input*/
DCL #5 FILE STREAM OUTPUT;            /* output file for MINITAB input*/
DCL DATAIN FILE STREAM INPUT;          /* raw data input file*/
DCL SCREEN FILE STREAM INPUT;         /* terminal input file*/
DCL CONSOLE FILE STREAM OUTPUT;       /* terminal output file*/

DCL FINISHED BIT(1) STATIC INIT('0'); /* input file end flag*/

                    /******************************
                    *                             *
```

```
*    DECLARATION OF FUNCTIONS  *
*                              *
*     Based on equations in    *
*     Appendix A4:4.2          *
*******************************/


          /***********************************
          *                                  *
          *   SATURATION PRESSURE FUNCTION   *
          *                                  *
          *    Calculates saturation         *
          *    pressure for corresponding    *
          *    temperature (oK)              *
          *                                  *
          ***********************************/

SPRESS:PROCEDURE(SAT_TEMP) RETURNS(FIXED DECIMAL(6,4));

       DCL SAT_TEMP FIXED DECIMAL(5,2);
       DCL SAT_PRESS FIXED DECIMAL(6,4);
       SAT_PRESS=((SAT_TEMP-157.564)/85.69)**3.8226;

       RETURN(SAT_PRESS);

END SPRESS;

          /***********************************
          *                                  *
          *   SATURATION TEMPERATURE FUNCTION  *
          *                                  *
          *    Calculates saturation temperature *
          *   from corresponding saturation  *
          *   pressure                       *
          *                                  *
```

```
          ****************************************/

STEMP:PROCEDURE(SH_PRESS) RETURNS(FIXED DECIMAL(5,2));

      DCL SH_PRESS FIXED DECIMAL(6,4);
      DCL SAT_TEMP FIXED DECIMAL(5,2);

      SAT_TEMP=85.963*(SH_PRESS)**.2616+157.564;

      RETURN (SAT_TEMP);

END STEMP;

          /****************************************
          *                                       *
          *  VAPOUR SPECIFIC HEAT FUNCTION        *
          *                                       *
          *   Calculates the specific heat of     *
          *   the superheated vapour.Requires     *
          *   the superheat pressure.             *
          *                                       *
          ****************************************/
CVAP:PROCEDURE(SH_PRESS) RETURNS(FIXED DECIMAL(6,4));

     DCL SH_PRESS FIXED DECIMAL(6,4);
     DCL VAP_C FIXED DECIMAL(5,4);

     VAP_C=0.61244*10**(0.008849*SH_PRESS);

     RETURN(VAP_C);

END CVAP;

          /****************************************
```

```
         *                                       *
         *  LIQUID SPECIFIC HEAT FUNCTION        *
         *                                       *
         *   Calculates the specific heat of     *
         *   the liquid.Requires the saturation *
         *   temperature and uses the            *
         *   SATURATION PRESSURE FUNCTION.       *
         *                                       *
         ****************************************/

CF:PROCEDURE(SAT_TEMP) RETURNS(FLOAT);

   DCL SAT_TEMP FIXED DECIMAL(5,2);
   DCL LIQ_C FLOAT DECIMAL;
   LIQ_C=0.888*10**(0.0066*SPRESS(SAT_TEMP));


   RETURN(LIQ_C);

END CF;


         /****************************************
         *                                       *
         *  LIQUID SPECIFIC VOLUME FUNCTION      *
         *                                       *
         *   Calculates the specific volume of  *
         *   the liquid.Requires the saturation *
         *   temperature.                        *
         *                                       *
         ****************************************/

LVOL:PROCEDURE(SAT_TEMP) RETURNS(FLOAT DECIMAL);

     DCL SAT_TEMP FIXED DECIMAL(5,2);
```

```
    DCL LIQ_VOL FLOAT DECIMAL(5);
    DCL TEMP FIXED DECIMAL(5,2);

    TEMP=388.3-SAT_TEMP;
    LIQ_VOL=1/(558.085+(0.777*TEMP)+(17.943*(TEMP**0.5))+
            (117.436*(TEMP**0.333))-(3.40204E-4*(TEMP**2)));

    RETURN(LIQ_VOL);

END LVOL;


      /****************************************
      *                                       *
      *   LIQUID ENTHALPY FUNCTION            *
      *                                       *
      *    Calculates the enthalpy of the     *
      *    liquid.Requires temperature and    *
      *    uses LIQUID SPECIFIC HEAT,LIQUID   *
      *    SPECIFIC VOLUME and SATURATION     *
      *    PRESSURE FUNCTIONS.                *
      *                                       *
      ****************************************/

HF:PROCEDURE(SAT_TEMP) RETURNS(FIXED DECIMAL(6,3));

   DCL SAT_TEMP FIXED DECIMAL(5,2);
   DCL LIQ_ENTH FIXED DECIMAL(6,3);

   LIQ_ENTH=(0.888*10**(0.0021*SPRESS(SAT_TEMP)))*
            (SAT_TEMP-233)+(LVOL(SAT_TEMP)+
            6.556E-4)/2*(SPRESS(SAT_TEMP)-0.6417);
```

```
   RETURN(LIQ_ENTH);

END HF;

         /****************************************
         *                                       *
         *   LIQUID/VAPOUR ENTHALPY FUNCTION     *
         *                                       *
         *    Calculates the enthalpy change     *
         *    for vaporization. Requires the     *
         *    SATURATION PRESSURE FUNCTION and   *
         *    the saturation temperature.        *
         *                                       *
         ****************************************/

HFG:PROCEDURE(SAT_TEMP) RETURNS(FIXED DECIMAL(6,3));

    DCL SAT_TEMP FIXED DECIMAL(5,2);
    DCL FG_ENTH FIXED DECIMAL(6,3);

    FG_ENTH=165.063*10**(-0.011*SPRESS(SAT_TEMP));

    RETURN(FG_ENTH);

END HFG;
         /****************************************
         *                                       *
         *   SUPERHEAT ENTHALPY FUNCTION         *
         *                                       *
         *    Calculates the enthalpy of the     *
         *    superheated vapour.Requires the    *
         *    LIQUID ENTAHLPY,LIQUID/VAPOUR      *
         *    and LIQUID SPECIFIC HEAT           *
         *    FUNCTIONS.  Superheat temperature  *
```

```
     *    and pressure are also required.    *
     *                                       *
     *****************************************/

HSH:PROCEDURE(SH_TEMP,SH_PRESS)RETURNS(FIXED DECIMAL(6,3));

   DCL SH_TEMP FIXED DECIMAL(5,2);
   DCL SH_PRESS FIXED DECIMAL(6,4);
   DCL SH_ENTH FIXED DECIMAL(6,3);
   DCL SAT_TEMP FIXED DECIMAL(5,2);

   SAT_TEMP= STEMP(SH_PRESS);

   SH_ENTH=HF(SAT_TEMP)+HFG(SAT_TEMP)+(SH_TEMP-SAT_TEMP)
           *CVAP(SH_PRESS);

   RETURN(SH_ENTH);

END HSH;

     /*****************************************
     *                                       *
     *   SUBCOOLED ENTHALPY FUNCTION         *
     *                                       *
     *    Calculates the enthalpy of the     *
     *    subcooled liquid.Requires LIQUID   *
     *    SPECIFIC HEAT and SATURATION       *
     *    TEMPERATURE FUNCTIONS and          *
     *    subcooled temperature and          *
     *    pressure.                          *
     *                                       *
     *****************************************/

HSC:PROCEDURE(SC_TEMP,SC_PRESS)RETURNS(FIXED DECIMAL(6,3));
```

```
   DCL SAT_TEMP FIXED DECIMAL(5,2);
   DCL SC_PRESS  FIXED DECIMAL(6,4);
   DCL SC_TEMP  FIXED DECIMAL(5,2);
   DCL SC_ENTH FIXED DECIMAL(6,3);

   SAT_TEMP=STEMP(SC_PRESS);

   SC_ENTH=HF(SAT_TEMP)-(SAT_TEMP-SC_TEMP)*CF(SAT_TEMP);

   RETURN(SC_ENTH);

END HSC;
                              /*****************************
                              *                             *
                              *        MAIN PROGRAM         *
                              *                             *
                              *****************************/

/*Obtain file names from terminal*/
OPEN FILE(SCREEN) TITLE('TTY -DEVICE');
OPEN FILE(CONSOLE) TITLE('TTY -DEVICE');
GET FILE(SCREEN) LIST(FILENAME);
PUT FILE(CONSOLE) LIST('File names are:',FILENAME);
CLOSE FILE(CONSOLE);
CLOSE FILE(SCREEN);

/*Open remaining files using filenames from screen for titles*/
OPEN FILE(DATAIN) TITLE(FILENAME(1));
OPEN FILE(RESULTS) TITLE(FILENAME(2));
OPEN FILE(TABLE) TITLE(FILENAME(3)) PRINT PAGESIZE(60) LINESIZE(132);
OPEN FILE(#1) TITLE(FILENAME(4));
OPEN FILE(#2) TITLE(FILENAME(5));
OPEN FILE(#3) TITLE(FILENAME(6));
OPEN FILE(#4) TITLE(FILENAME(7));
OPEN FILE(#5) TITLE(FILENAME(8));
```

```
/*Set end of file condition*/
ON ENDFILE(DATAIN) FINISHED='1'B;


/*Read in data structure while file is not empty*/
DO WHILE(^FINISHED);

   GET FILE(DATAIN) LIST(FIRSTKWH,KWH7,KWH4,FREONTOTS,
                         FILLTIMES,STOP_TIME,NROWS);

   GET FILE(DATAIN) LIST((COMPKWH(J),WATT(J),AMP(J),
                        VOLT(J),STWATT(J),PF(J),FANKWH(J),TIME(J)
                        DO J=1 TO NROWS));

   GET FILE(DATAIN) LIST(((PRESSURE(I,J) DO J=1 TO 6),(FLOWS(I,J)
                        DO J=1 TO 5),FREONTOT(I,1),FREONTOT(I,2)
                        DO I= 1 TO NROWS));

   GET FILE(DATAIN) LIST(((TEMPS(I,J) DO J=1 TO 15)
                         DO I=1 TO NROWS));

   GET FILE(DATAIN) LIST(((TEMPS(I,J) DO J=16 TO 21)
                         DO I=1 TO NROWS));
   GET FILE(DATAIN) SKIP(2);

END;


/*Calculate  power totals from kW h meter readings at each stop time */
/*compressor power X 3 for 1 phase to 3 phase                        */
KWH7(1)=(KWH7(1)-FIRSTKWH(1))*3;
KWH4(1)=(KWH4(1)-FIRSTKWH(1))*3;
KWH4(3)=(KWH4(3)-FIRSTKWH(3))*3;
```

```
KWH7(2)=(KWH7(2)-FIRSTKWH(2));
KWH4(2)=(KWH4(2)-FIRSTKWH(2));
KWH4(4)=(KWH4(4)-FIRSTKWH(4));


/*Repeat all calculations for */
/*each line of data.          */
DO I=1 TO NROWS;

   /*Convert mV to oK */
   DO J=1 TO 21;
      TEMPS(I,J)=TEMPS(I,J)*23.364+273;
   END;

   /*Convert kPa to bar abs. */
   DO J=1 TO 6;
      PRESSURE(I,J)=PRESSURE(I,J)/100.0+1.0;
   END;

   /*Calculate compressor and fan kWh */
   /*running totals                   */
   COMPKWH(I)=(COMPKWH(I)-FIRSTKWH(1))*3;
   FANKWH(I)=FANKWH(I)-FIRSTKWH(2);

   /*Calculate instantaneous compressor */
   /*power consumption.                 */
   WATT_CAL(I)=(3*VOLT(I)*AMP(I)*PF(I))/(SQRT(3)*1000);

   /*Calculate enthalpies */
   COMPH(I)=HSH(TEMPS(I,2),PRESSURE(I,2))-HSH(TEMPS(I,1),
            PRESSURE(I,1));

   DSH(I)=HSH(TEMPS(I,2),PRESSURE(I,2))-(HF(STEMP
          (PRESSURE(I,3)))+HFG(STEMP(PRESSURE(I,3))));
```

```
/*Use TEMPS(I,7) for water cooled system instead of T8 */
CONDH(I)=HF(STEMP(PRESSURE(I,3)))+HFG(STEMP(PRESSURE(I,3)
        ))-HSC(TEMPS(I,8),PRESSURE(1,3));

/*Use TEMPS(I,7) for water cooled system instead of TEMPS(I,8) */
SCH=HF(TEMPS(I,8));

SCCOND_RECH(I)=SCH-HF(TEMPS(I,10));
REC_EXPH(I)=HF(TEMPS(I,10))-HF(TEMPS(I,11));

EVAPH(I)=HSH(TEMPS(I,13),PRESSURE(I,6))-
        HF(TEMPS(I,11));

CEVAPH(I)=HSH(TEMPS(I,1),PRESSURE(I,1))-(EVAPH(I)+SCH);


/*Convert flow Hz to l/min. */
/*or kg/s .                  */
FLOWS(I,1)=FLOWS(I,1)*0.030321;
FLOWS(I,3)=FLOWS(I,3)*0.06608+0.001122;
FLOWS(I,4)=(FLOWS(I,1)/6E4)/LVOL(TEMPS(I,9));
FLOWS(I,5)=FLOWS(I,5)*0.02868;
FLOWS(I,6)=FLOWS(I,5)/6E4/LVOL(TEMPS(I,10));

/*Calculate overall and thermodynamic */
/*COPs and compressor/motor efficiency*/
COP(I,1)=FLOWS(I,6)*EVAPH(I)/WATT_CAL(I);
COP(I,2)=EVAPH(I)/COMPH(I);
COMP_EFF(I)=COP(I,1)/COP(I,2);

/*Calculate heat loss form refrigerant */
/*and heat gain by water.              */
EXCHQ(I,1)=FLOWS(I,3)/60*(TEMPS(I,17)-TEMPS(I,16))*4.186;
EXCHQ(I,2)=FLOWS(I,4)*(HSH(TEMPS(I,3),PRESSURE(I,2))-HSH
```

```
                        (TEMPS(I,4),PRESSURE(I,3)));


/*Calculate Log Mean Temperature Difference */
Q1= (TEMPS(I,3)-TEMPS(I,17));
Q2= (TEMPS(I,4)-TEMPS(I,16));
IF Q1>Q2
THEN LMTD(I)=(Q1-Q2)/LOG((Q1/Q2));
ELSE IF Q1<Q2
     THEN LMTD(I)=(Q2-Q1)/LOG((Q2/Q1));
     ELSE LMTD(I)=Q1;


/*Calculate overall thermal conductance x */
/*area product                            */
UA(I)=EXCHQ(I,2)/LMTD(I);

/*Calculate Ec  */
CREF(I)=FLOWS(I,4)*((CVAP(PRESSURE(I,2))+CVAP(PRESSURE(I,3)
                                                    ))/2);
CWAT(I)=FLOWS(I,3)/60*4.186;

IF CWAT(I) >= CREF(I)
THEN DO;
       C=CREF(I)/CWAT(I);
       CMIN=CREF(I);
     END;
ELSE DO;
       C=CWAT(I)/CREF(I);
       CMIN=CWAT(I);
     END;

NTU(I)=UA(I)/CMIN;

EC(I)=(1-EXP(-NTU(I)*(1-C)))/(1-C*EXP(-NTU(I)
```

```
          *(1-C)));

END;

/*Convert oK to oC for printing */
DO I=1 TO NROWS;
   DO J=1 TO 21;
      TEMPS(I,J)=TEMPS(I,J)-273;
    END;
END;


/*Output results to MINITAB files and */
/*RESULTS file.                        */

/*Output to RESULTS and MINITAB #1 files */
 PUT FILE(RESULTS) EDIT((COMPKWH(J),WATT(J),WATT_CAL(J),AMP(J),VOLT(J),
                      STWATT(J),PF(J),FANKWH(J),(PRESSURE(J,I)
                                               DO I=1 TO 6)
                                               DO J=1 TO NROWS))
                     (COL(1),4 F(5,2),2 F(6,1),F(5,2),
                      F(5,2),F(5,2),2 F(6,2),3 F(5,2));

 PUT FILE(#1) EDIT((COMPKWH(J),WATT(J),WATT_CAL(J),AMP(J),VOLT(J),
                 STWATT(J),PF(J),FANKWH(J),(PRESSURE(J,I)
                                               DO I=1 TO 6)
                                          DO J=1 TO NROWS))
                     (COL(2),4 F(5,2),2 F(6,1),3 F(5,2),
                      2 F(6,2),3 F(5,2));

/*Output to RESULTS and MINITAB #2 files */
 PUT FILE(RESULTS) EDIT(((TEMPS(I,J)  DO J=1 TO 10)
                                      DO I=1 TO NROWS))
                     (COL(1),F(5,2),2 F(7,2),7 F(6,2));
```

```
 PUT FILE(#2) EDIT(((TEMPS(I,J) DO J=1 TO 10)
                               DO I=1 TO NROWS))
                 ( COL(2),F(5,2),2 F(7,2),7 F(6,2));

/*Output to RESULTS and MINITAB #3 files */
 PUT FILE(RESULTS) EDIT(((TEMPS(I,J) DO J=11 TO 21)
                                    DO I=1 TO NROWS))
                      ( COL(1),11 F(6,2));

 PUT FILE(#3) EDIT(((TEMPS(I,J) DO J=11 TO 21)
                               DO I=1 TO NROWS))
                 ( COL(2),11 F(6,2));

/*Output to RESULTS and MINITAB #4 files */
 PUT FILE(RESULTS) EDIT(((FLOWS(I,J) DO J=1 TO 6),ENTHALPIES(I)
                                    DO I=1 TO NROWS))
                      ( COL(1),F(5,2),F(6,2),F(5,2),
                  F(7,4),F(5,2),F(7,4),2 F(6,2),F(8,3),2 F(7,3),
                  F(8,3),F(7,3));

 PUT FILE(#4) EDIT(((FLOWS(I,J) DO J=1 TO 6),ENTHALPIES(I)
                               DO I=1 TO NROWS))
                 ( COL(2),F(5,2),F(6,2),F(5,2),
                  F(7,4),F(5,2),F(7,4),2 F(6,2),F(8,3),2 F(7,3),
                  F(8,3),F(7,3));

/*Output to RESULTS and MINITAB #5 files */
 PUT FILE(RESULTS) EDIT((FREONTOT(I,1),FREONTOT(I,2),TIME(I),(COP(I,J)
                        DO J=1 TO 2),COMP_EFF(I),
                        PRIMEXCHANGER(I) DO I=1 TO NROWS))
                      ( COL(1),2 F(4),F(4),3 F(5,2),
                        2 F(6,3),F(7,3),5 F(6,3));

 PUT FILE(#5) EDIT((FREONTOT(I,1),FREONTOT(I,2),TIME(I),COP(I,1),COP(I,2),
                   COMP_EFF(I),PRIMEXCHANGER(I) DO I=1 TO NROWS))
```

```
                    (COL(2),2 F(4),F(4),3 F(5,2),
                     2 F(6,3),F(7,3),5 F(6,3));

/*Print table of all results with headings */

/*Page 1*/
PUT FILE(TABLE) EDIT('NUMBER OF COLUMNS IS',NROWS)(SKIP,COL(1),
            A,F(3));

/*Page 2*/
 PUT FILE(TABLE) PAGE;
 PUT FILE(TABLE) EDIT('COMP KWH','WATTS','WATTS_CAL','AMPS','VOLTS',
                      'STWATTS','PF','FAN KWH','NOMINUTES')
                     (SKIP,COL(1),A,COL(14),A,COL(24),A,
                     COL(38),A,COL(50),A,COL(60),A,
                     COL(75),A,COL(85),A,COL(96),A);
 PUT FILE(TABLE) EDIT((POWER(I),TIME(I) DO I=1 TO NROWS))
                      (SKIP(2),COL(1),9 (F(6,2),X(6)));

/*Page 3*/
 PUT FILE(TABLE) PAGE;
 PUT FILE(TABLE) EDIT('COM IN','COM OUT','COND',
                      'RECEIV','EVAP IN','EVAP OUT')
                     (SKIP(2),COL(3),A,COL(18),A,
                      COL(35),A,COL(51),A,COL(67),A,
                      COL(83),A);
 PUT FILE(TABLE) EDIT(((PRESSURE(I,J) DO J=1 TO 6) DO I=1 TO NROWS))
                     (SKIP(2),COL(1),6 (F(6,2),X(10)));

/*Page 4*/
 PUT FILE(TABLE) PAGE;
 PUT FILE(TABLE) EDIT('COI','COO','PRI','PRO','WRI',
                      'ARI','WRO','ARO','REI','REO',
                      'EXI','EVI','EVO','WWI','WWO',
                      'PWI','PWO','HED','MKI','VAT',
```

```
                'MOT')(SKIP,COL(1),21 (X(3),A));
 PUT FILE(TABLE) EDIT(((TEMPS(I,J) DO J=1 TO 21) DO I=1 TO NROWS))
                (SKIP(2),COL(1),21 (F(6,2)));

/*Page 5*/
 PUT FILE(TABLE) PAGE;
 PUT FILE(TABLE) EDIT('REFRIG FLOW','CONDENSER WATER FLOW',
                'HOT WATER FLOW','FREON MASS FLOW',
                'FREON TOT1','FREON TOT2','COP','COPT','%')
                (SKIP(2),COL(3),7 (A,X(5)),A,X(3),A);
 PUT FILE(TABLE) EDIT((FLOWS(I,1),FLOWS(I,5),(FLOWS(I,J)
                                    DO J=2 TO 4),
                 FLOWS(I,6),FREONTOT(I,1),FREONTOT(I,2),
                 (COP(I,J) DO J=1 TO 2),
                 COMP_EFF(I) DO I=1 TO NROWS))
               (SKIP(2),COL(1),2 F(6,3),COL(22),F(6,3),
                COL(47),F(6,3),COL(63),2 F(7,4),COL(86),
                F(5),COL(101),F(5),COL(112),F(5,3),COL(120),F(5,3),
                COL(129),F(4,2));

/*Page 6*/
 PUT FILE(TABLE) PAGE;
 PUT FILE(TABLE) EDIT('ENTHALPIES','PRIMARY EXCHANGER')
               (COL(25),A,COL(100),A);
 PUT FILE(TABLE) EDIT('COMPH','DSH','CONDH','SCCOND_RECH',
                'REC_EXPH',
                 'EVAPH','CEVAPH','EXCHQW','EXCHQR',
                 'LMTD','UA','CWAT','CREF','NTU','EC')
                (SKIP(2),7 (A,X(4)),COL(73),8 (A,X(4)));
 PUT FILE(TABLE) EDIT((ENTHALPIES(I),PRIMEXCHANGER(I)
                 DO I=1 TO NROWS))(SKIP(2),F(7,3),
                  3 F(8,3),2 F(12,3),F(10,3),COL(73),
                  F(6,3),2 F(9,3),5 F(7,3));

/*Page 7*/
```

```
PUT FILE(TABLE) PAGE;
PUT FILE(TABLE) EDIT('TIMING STATISTICS','COOLING 800 LITRES',
                'COOLING 2000 LITRES')(SKIP,COL(50),A,
                 SKIP(2),COL(21),A,COL(73),A);
PUT FILE(TABLE) EDIT('COOLING TO 7 OC','COMPRESSOR KWH',
                 KWH7(1))(SKIP(2),A,SKIP,
                 COL(3),A,COL(33),F(5,2));
PUT FILE(TABLE) EDIT('FAN KWH',KWH7(2))(SKIP,COL(3),
                 A,COL(33),F(5,2));
PUT FILE(TABLE) EDIT('FILL TIME (MIN)',FILLTIMES(1)
                )(SKIP,COL(3),A,COL(33),F(5));
PUT FILE(TABLE) EDIT('COOLING TIMES (MIN)','FROM START',
                 STOP_TIME(1),'FROM END OF'
                !!' FILLING',STOP_TIME(1)-FILLTIMES(1)
                 )(SKIP,COL(3),
                 A,2 (SKIP,COL(5),A,COL(33),F(5)
                 ));
PUT FILE(TABLE) EDIT('FREON TOTAL (LITRES)',FREONTOTS(1),FREONTOTS(2))
                  (SKIP,COL(3),A,COL(33),2 F(5));

PUT FILE(TABLE) EDIT('COOLING TO 4 OC','COMPRESSOR KWH',
                 KWH4(1),KWH4(3))(SKIP(2),A,SKIP,
                 COL(3),A,COL(33),F(5,2),COL(81),F(5,2));
PUT FILE(TABLE) EDIT('FAN KWH',KWH4(2),KWH4(4))(SKIP,COL(3),
                 A,COL(33),F(5,2),COL(81),F(5,2));
PUT FILE(TABLE) EDIT('FILL TIME (MIN)',FILLTIMES(1),
                FILLTIMES(2))(SKIP,COL(3),A,COL(33),F(5),
                COL(81),F(5));
PUT FILE(TABLE) EDIT('COOLING TIMES (MIN)','FROM START',
                 STOP_TIME(2),STOP_TIME(4),'FROM END OF'
                !!' FILLING',STOP_TIME(2)-FILLTIMES(1),
                 STOP_TIME(4)-FILLTIMES(2))(SKIP,COL(3),
                 A,2 (SKIP,COL(5),A,COL(33),F(5),COL(81),
                 F(5)));
PUT FILE(TABLE) EDIT('FREON TOTAL (LITRES)',FREONTOTS(5),
```

```
                        FREONTOTS(6),FREONTOTS(7),
                        FREONTOTS(8))(SKIP,COL(3),A,COL(33),
                        2 F(5),COL(81),2 F(5));

/*Close all output  files*/
  CLOSE FILE(RESULTS);
  CLOSE FILE(TABLE);
  CLOSE FILE(#1);
  CLOSE FILE(#2);
  CLOSE FILE(#3);
  CLOSE FILE(#4);
  CLOSE FILE(#5);

END ANALYSIS;
```

A4:5 DEVELOPMENT OF EQUATIONS AND ERROR ANALYSIS

A4:5.1 Development of Refrigerant Equations

A4:5.1.1 Saturation Pressure (P_{SAT})

The relationship between saturation pressure in bar abs (P_{SAT}) and saturation temperature in ^{o}K (T_{SAT}) is presented in Figure A4:5 with the relationship between dT_{SAT}/dP_{SAT} and P_{SAT} presented in Figure A4:6.

The equation for the line in Figure A4:6 (determined from a least squares analysis) is:-

$$\log \frac{dT_{SAT}}{dP_{SAT}} = -0.7384 \log P_{SAT} + \log 22.488 \qquad \text{.....Eqn A4:1}$$

Transposing Eqn A4:1 and rearranging gives:-

$$dT_{SAT} = 22.488\, P_{SAT}^{-0.7348} . dP_{SAT} \qquad \text{.....Eqn A4:2}$$

and after integration gives:-

$$T_{SAT} = \frac{22.488}{0.2616} P_{SAT}^{0.2616} + C \qquad \text{.....Eqn A4:3}$$

where C = constant

The constant C ($157.564^{o}K$) was evaluated by substituting a value of P_{SAT} (3.0861 bar abs) and the corresponding value of T_{SAT} ($273^{o}K$) from Freon 12 tables into Eqn A4:3.

Substituting the value for C into Eqn A4:3 and rearranging gives the final expression for P_{SAT} of:-

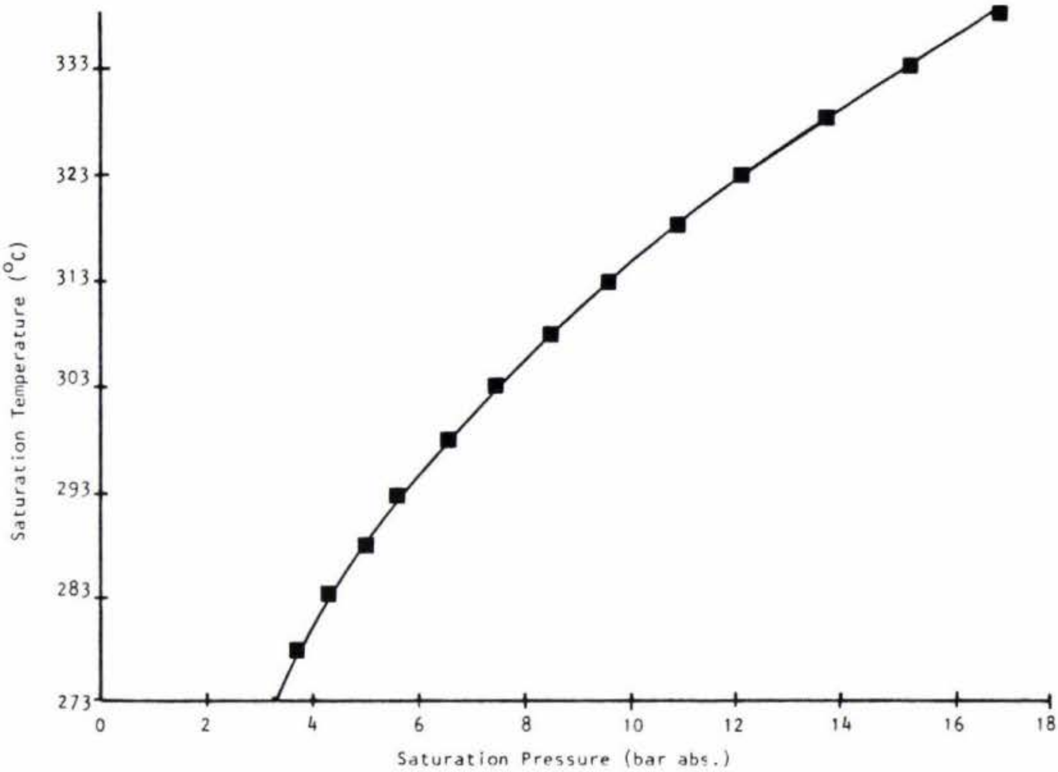


FIGURE A4:5

Relationship between saturation temperature and pressure for Freon 12

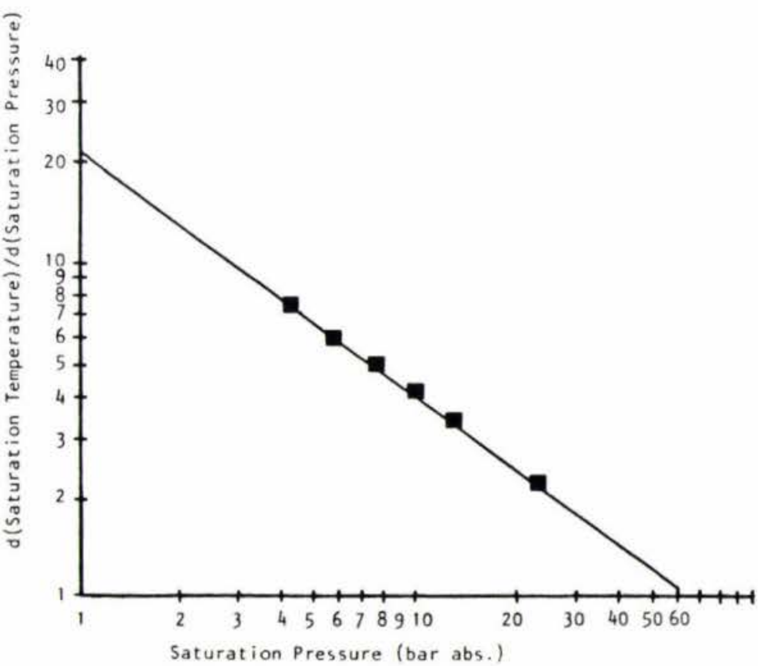


FIGURE A4:6

Relationship between the slope of FIGURE A4:5 and saturation pressure

$$P_{SAT} = \left(\frac{T_{SAT} - 157.564}{85.963}\right)^{3.823} \qquad \text{.....Eqn A4:4}$$

A comparison of the values of P_{SAT} from Eqn A4;4 and the Freon tables is presented in Table A4:5, from which the mean error was calculated to be $\pm$ 0.3%.

A4:5.1.2 Specific Heat of the Superheated Vapour (C_{VAP})

Analysis of the data for C_{VAP} in $kJ.kg^{-1}.^{o}K^{-1}$ and suction superheat pressure in bar abs. (P_{SH}) produced the equation for the straight line of C_{VAP} against P_{SH} in Figure A4:7 of :-

$$C_{VAP} = 0.61244 \times 10^{(0.008849 \times P_{SH})} \qquad \text{.....Eqn A4:5}$$

The difference between values of C_{VAP} obtained from Eqn A4:5 and Freon tables for a range of pressures is presented in Table A4:6. The mean error in C_{VAP} was calculated to be $\pm$ 2%.

A4:5.1.3 Specific Heat of the Liquid (C_f)

A plot of C_f against P_{SAT} is presented in Figure A4:8, from which the equation developed using the least squares method was:-

$$C_f = 0.888 \times 10^{(0.0061\ P_{SAT})} \qquad \text{.....Eqn A4:6}$$

Table A4:7 presents the differences between values obtained from Eqn A4:6 and the Freon tables. From this table the mean error for Eqn A4:6 was calculated to be $\pm$ 0.3%.

TABLE A4:5

Differences between table and calculated values of P_{SAT}

T_{SAT} (°K)	P_{SAT} (bar abs)		Difference (bar abs)	% Difference
	Table	Equation		
253	1.5093	1.4912	-0.0181	-1.20
263	2.1912	2.1826	-0.0086	-0.39
273	3.0861	3.0860	-0.0001	0.00
283	4.2330	4.2396	0.0066	0.15
293	5.6729	5.6841	0.0112	0.20
303	7.4490	7.4632	0.0142	0.19
313	9.6065	9.6233	0.0168	0.17
323	12.193	12.213	0.020	0.16
333	15.259	15.285	0.026	0.17

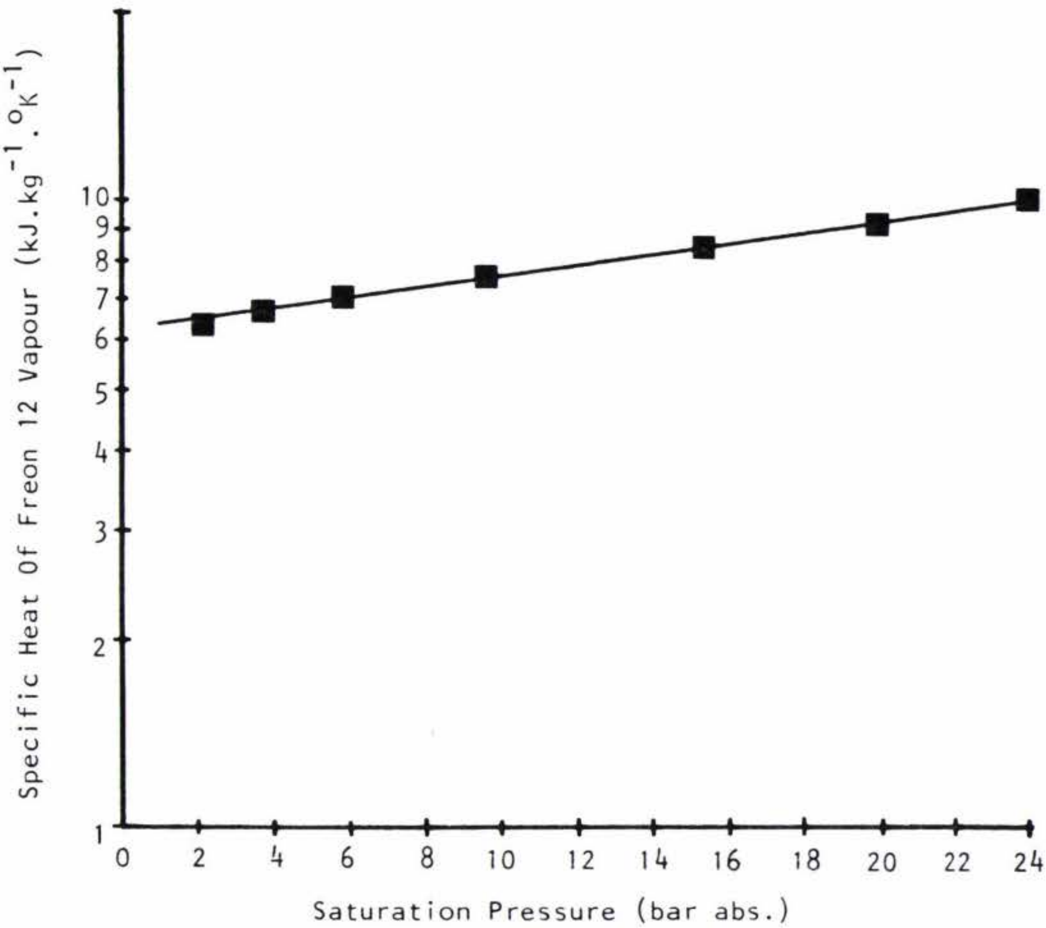


FIGURE A4:7

Relationship between the specific heat of Freon 12 vapour and saturation pressure.

TABLE A4:6

Differences between equation and Freon table values for C_{VAP}

P_{SH} (bar abs)	C_{VAP} ($kJ.kg^{-1}.^{o}K^{-1}$)		Difference ($kJ.kg^{-1}.^{o}K^{-1}$)	% Difference
	Table	Equation		
2.1912	0.626	0.640	0.014	2.23
3.0861	0.647	0.652	0.005	0.77
3.6255	0.660	0.659	-0.001	-0.15
6.8782	0.716	0.705	-0.011	-1.54
7.4490	0.727	0.713	-0.014	-1.93
9.6065	0.759	0.745	-0.014	-1.84
12.1932	0.792	0.785	-0.007	-0.88
15.2592	0.839	0.836	-0.003	-0.36

Table values were calculated from:-

$$C_{VAP} = \frac{h_{SH} - h_{SAT}}{30}$$

where h_{SH} = specific enthalpy at 30°C superheat for P_{SH} (average for range of values expected from experimental results) ($kJ.kg^{-1}$)

h_{SAT} = specific enthalpy at saturation for P_{SH} ($kJ.kg^{-1}$)

P_{SH} = superheat pressure (bar abs)

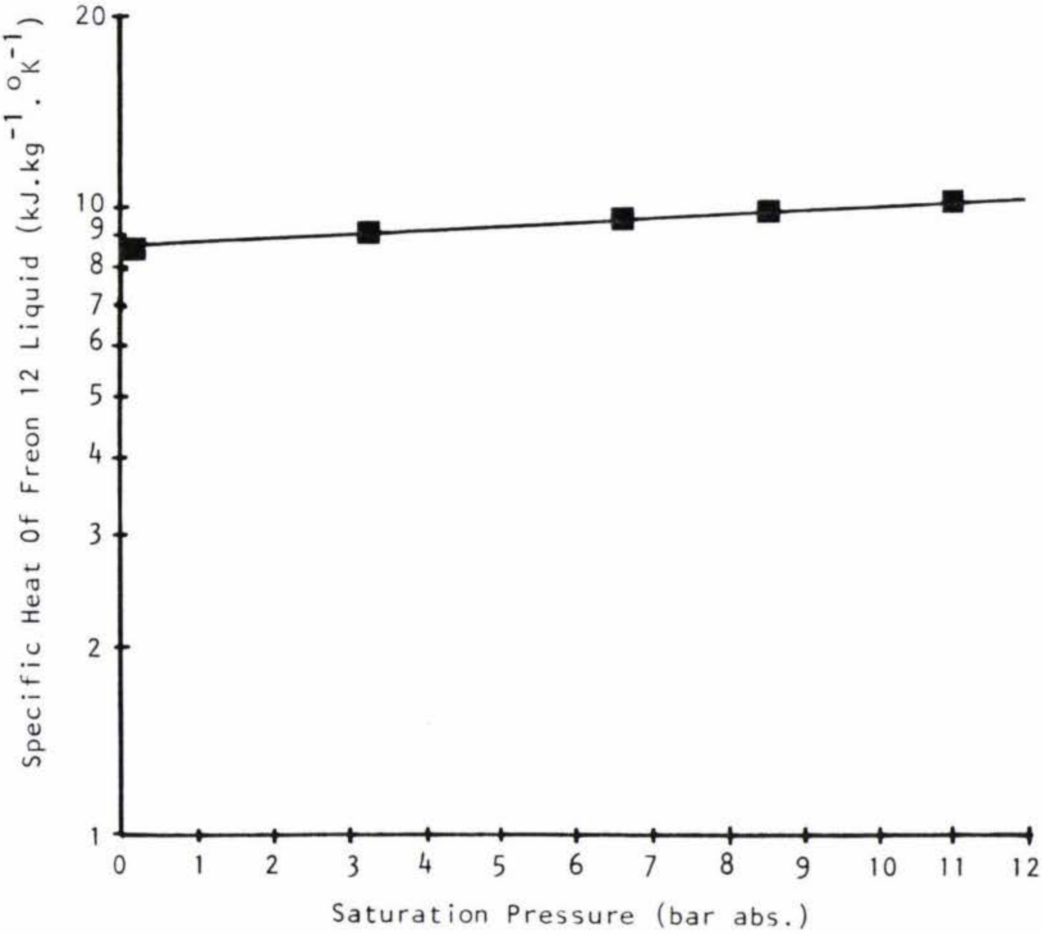


FIGURE A4:8

Relationship between the specific heat of Freon 12 liquid and saturation pressure.

TABLE A4:7

Differences between table and calculated values for C_f

P_{SAT} (bar abs)	C_f ($kJ.kg^{-1}.^oK^{-1}$) Table	Equation	Difference ($kJ.kg^{-1}.^oK^{-1}$)	% Difference
1.5093	0.901	0.905	0.004	0.44
2.1912	0.912	0.914	0.002	0.22
3.0861	0.925	0.927	0.002	0.22
4.2330	0.941	0.943	0.002	0.21
5.6729	0.962	0.964	0.002	0.21
7.4490	0.986	0.990	0.004	0.41
9.6065	1.017	1.022	0.005	0.49
12.193	1.058	1.063	0.005	0.47
15.259	1.111	1.113	0.002	0.18

A4:5.1.4 Specific Volume of the Liquid (V_f)

Metricating the equation for V_f from McHarness et al. (1955) gives:-

$$\frac{1}{V_f} = 558.025 + 0.777(388.3T) + 17.943(388.3-T)^{1/2}$$
$$+ 117.436(388.3-T)^{1/3}$$
$$- 3.40204 \times 10^{-4}(388.3-T)^2 \quad .....\text{Eqn A4:7}$$

where T = temperature in ^{o}K

The differences between the values obtained from Eqn A4:7 and the Freon table are presented in Table A4:8, from which the mean error was calculated to be -0.98%.

A4:5.1.5 Specific Enthalpy of the Liquid (h_f)

The general form of the equation for h_f from McHarness et al. (1955) was:-

$$h_f = \int_{233}^{T} C_f \, . \, dT + \int_{0.6417}^{P_{SAT}} \left(\frac{V_{f1} + V_{f2}}{2}\right) . \, dP_{SAT}$$

.....Eqn A4:8

The integration of this equation by substituting Eqn 4:5 (C_f) and Eqn 4:6 (V_f) resulted in Eqn 4:7. However, an alternative method using a modified version of Eqn A4:6 and Eqn A4:7 was selected for the computer programme as a result of the limits for Eqn 4:5. The equation used in the computer programme was:-

TABLE A4:8

Differences between table and calculated values for V_f

T (^oK)	V_f ($m^3 \times 10^{-3}.kg^{-1}$)		Difference ($m^3 \times 10^{-3}.kg^{-1}$)	% Difference
	Table	Equation		
273	0.71590	0.71053	0.00537	-0.75
283	0.73327	0.72737	0.00590	-0.80
293	0.75246	0.74592	0.00654	-0.87
303	0.97386	0.76654	0.00732	-0.95
313	0.79802	0.78972	0.00830	-1.04
323	0.82573	0.81615	0.00958	-1.16
333	0.85814	0.84684	0.01130	-1.32

$$h_f = (0.888 \times 10^{(0.0021 \times P_{SAT})})(T - 233)$$

$$+ \left(\frac{V_{f1} + 6.556 \times 10^{-4}}{2}\right)(P_{SAT} - 0.6417) \qquad \text{.....Eqn A4:9}$$

where T = liquid temperature ^oK

233 = temperature (^oK) for h_f = 0 $kJ.kg^{-1}$

6.556×10^{-4} = V_f ($m^3.kg^{-1}$) for h_f = 0 $kJ.kg^{-1}$

0.6417 = P (bar abs) for h_f = 0 $kJ.kg^{-1}$

Table A4:9 presents the differences between the table values and those calculated from Eqn A4:9. The mean error calculated from Table A4:9 was $\pm$ 0.2%. Calculations of h_f, to allow for the 0.5% error in temperature (^oC) resulted in an overall mean error of 0.3%.

A4:5.1.6 Specific Enthalpy of Vaporisation (h_{fg})

Data for h_{fg} in $kJ.kg^{-1}$ and P_{SAT} in bar abs is plotted in Figure A4:9, from which the equation of the straight line is:-

$$h_{fg} = 165.063 \times 10^{(-0.011025\ P_{SAT})} \qquad \text{.....Eqn A4:10}$$

The difference table (Table A4:10) resulted in a mean error of $\pm$ 1.3%.

A4:5.1.7 Specific Enthalpy of the Superheat Vapour (h_{SH})

The specific enthalpy of the superheat vapour was calculated from the sum of h_f, h_{fg} and change in enthalpy in the superheated vapour region, i.e.,

TABLE A4:9

Differences between table and calculated values for h_f

T (^oK)	P (bar abs)	h_f ($kJ.kg^{-1}$) Table	h_f ($kJ.kg^{-1}$) Equation	Difference ($kJ.kg^{-1}$)	% Difference
253	1.0593	17.816	17.889	0.073	0.41
263	2.1912	26.674	26.924	0.050	0.19
273	3.0861	36.052	36.056	-0.004	0.01
283	4.2330	45.357	45.322	-0.035	0.08
293	5.6729	54.873	54.768	-0.105	-0.19
303	7.4490	64.592	64.449	-0.143	-0.22
313	9.6065	74.587	74.430	-0.157	-0.21
323	12.193	84.936	84.790	-0.146	-0.17
333	15.259	95.742	95.623	-0.119	-0.12

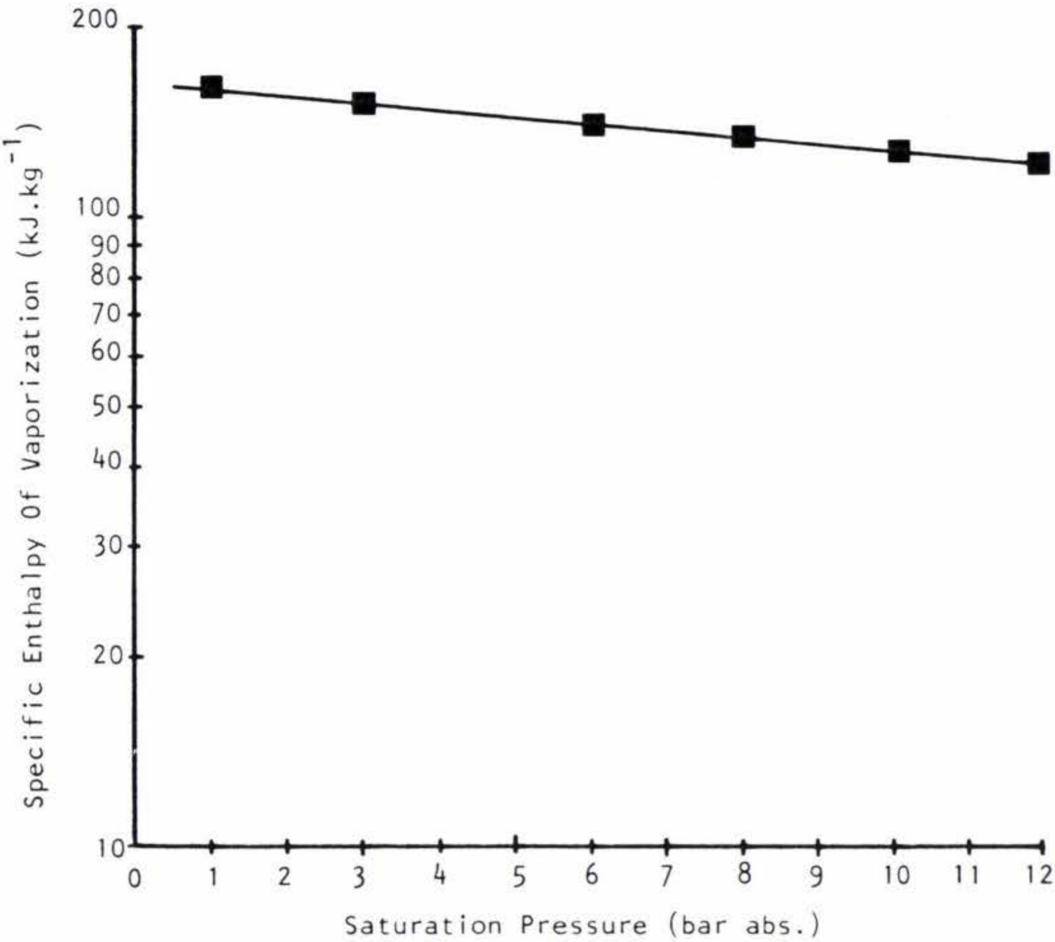


FIGURE A4:9

Relationship between specific enthalpy of vaporization for Freon 12 and saturation pressure

TABLE A4:10

Differences between table and calculated values of h_{fg}

P_{SAT} (bar abs)	h_{fg} ($kJ.kg^{-1}$) Table	Equation	Difference ($kJ.kg^{-1}$)	% Difference
1.5093	160.918	158.945	-1.973	-1.23
2.1912	156.312	156.186	-0.126	-0.08
3.0661	151.477	152.652	1.175	0.78
4.2330	146.363	148.257	1.894	1.29
5.6729	140.907	142.930	2.023	1.44
7.4490	135.026	136.633	1.607	1.19
9.6065	128.611	129.358	0.747	0.58
12.193	121.512	121.144	-0.368	-0.30
15.259	113.519	112.076	-1.443	-1.27

$$h_{SH} = h_f + h_{fg} + C_{VAP}(T_{SH} - T_{SAT})$$

.....Eqn A4:11

where T_{SH} = temperature of superheat vapour (^oK)

C_{VAP} = specific heat of the vapour for the super heat pressure (P_{SH}) ($kJ.kg^{-1}.^oK^{-1}$)

Table and calculated values of h_{SH} for 20^{o}K of superheating are presented in Table A4:11, from which the mean error was calculated to be $\pm$ 0.7%. The effect of the 0.5% error in temperature was calculated to be $\pm$ 0.03%, and for the $\pm$ 0.05 bar error in pressure, $\pm$ 0.13%. Combining these errors results in an overall mean error of $\pm$ 0.7%.

A4:5.2 Errors for Calculated System Variables

A4:5.2.1 Error in Refrigeration Effect (R.E.)

The equation for R.E. (Eqn 2:5) was taken from Section 2:3.1.6 where:-

$$R.E. = h_{SH} - h_f \quad (kJ.kg^{-1})$$

Using typical values of 200 $kJ.kg^{-1}$ for h_{SH} and 60 $kJ.kg^{-1}$ for h_f, and the respective errors from Sections A4:5.1.5 and A4:5.1.7, then:-

$$h_{SH} = 200\ kJ.kg^{-1} \pm 0.7\% = 200 \pm 1.4\ kJ.kg^{-1}$$

$$h_f = 60\ kJ.kg^{-1} \pm 0.3\% = 60 \pm 0.18\ kJ.kg^{-1}$$

Applying these values to the above equation results in:-

TABLE A4:11

Differences between table and calculated values of h_{SH}

T_{SH} (^oK)	P_{SH} (bar abs)	h_{SH} ($kJ.kg^{-1}$) Table	h_{SH} ($kJ.kg^{-1}$) Equation	Difference ($kJ.kg^{-1}$)	% Difference
283	2.1912	195.672	195.774	0.102	0.052
293	3.0861	200.488	201.639	1.151	0.574
303					
313					
323	7.4490	214.254	215.403	1.149	0.536
333	9.6065	218.527	218.841	0.314	0.144
343	12.193	222.583	221.910	-0.673	-0.302
353	15.259	226.375	224.845	-1.530	-0.676

T_{SH} = 20^oK of superheating (typical value)

$$R.E. = 200 \pm 1.4\ kJ.kg^{-1} - 60 \pm 0.18\ kJ.kg^{-1}$$

$$= 140 \pm (1.4^2 + 0.18^2)^{1/2}\ kJ.kg^{-1}$$

$$= 140 \pm 1.4\ kJ.kg^{-1}$$

$$= 140\ kJ.kg^{-1} \pm 1.0\%$$

A4:5.2.2 Error in Calculated Instantaneous Compressor Power Consumption in kW (I.C.P.C.)

The equation of instantaneous compressor power consumption is:-

$$I.C.P.C. = \frac{3 \times Volts \times Amps/phase \times P.F.}{\sqrt{3} \times 1000} \quad \text{.....Eqn A4:12}$$

Where the error in volt meter reading	=	$\pm$ 0.25 V
Error in volts (meter reading x 4)	=	$\pm$ 1.0 V
Error in amps/phase meter reading	=	$\pm$ 0.025 A
Error in amps/phase (meter reading x 5)	=	$\pm$ 0.125 A
Error in P.F.	=	$\pm$ 0.01

Taking typical examples of

$$Volts = 410 \pm 1.0\ V$$

$$Amps/phase = 6.00 \pm 1.25\ A$$

$$P.F. = 0.60 \pm 0.01$$

then:-

$$I.C.P.C. = 2.556 \pm [(\frac{1}{410})^2 + (\frac{0.15}{6.00})^2 + (\frac{0.01}{0.60})^2]^{1/2}$$

$$= 2.556 \pm 2.7\%$$

A4:5.2.3 Error in Coefficient of Performance

In Section 4:4.2.2 the equation for C.O.P. (Eqn 4:11) was:-

$$COP = \frac{R.E. \times M_r}{I.C.P.C.}$$

Taking typical values for

$$R.E. = 140 \text{ kJ.kg}^{-1} \pm 1.0\% \quad \text{(Eqn 2:5)}$$

$$M_r = \frac{2.03 \text{ l.min}^{-1} \pm 1.7\% \times 1322 \text{ m}^3.\text{kg}^{-1} \pm 0.98\%}{60000}$$

(Eqn 4:10)

$$= 0.0447 \pm 2.0\%$$

$$I.C.P.C. = 2.556 \text{ kW} \pm 2.7\%$$

results in a typical value for C.O.P. of:-

$$COP = \frac{140 \times 0.0447}{2.556} \pm (0.01^2 + 0.02^2 + 0.027^2)^{1/2}$$

$$= 2.45 \pm 3.5\%$$

A4:5.2.4 Error in Heat Recovery Rate

From Section 4:4.1 Eqn 4:2 was:-

$$EXCHQ = M_w \times Sp\ Ht_w \times \Delta t$$

where M_w = mass flow rate of water ($kg.s^{-1}$)
(error = $\pm$ 1.0% Section A4:4.3)

$Sp\ Ht_w$ = specific heat of water ($kJ.kg^{-1}.{}^oC^{-1}$)
= ($4.186\ kJ.kg^{-1}.{}^oC^{-1}$)

Δt = temperature rise (^{o}C)

= (two $\pm$0.5% - twi $\pm$ 0.5%)

Taking typical values of

M_W = 0.01042 $kg.s^{-1}$ $\pm$ 1.0%

Δt = (55^{o}C $\pm$ 0.5% - 15^{o}C $\pm$ 0.5%)

= 40^{o}C $\pm$ 0.7%

then:-

$$EXCHQ = 0.01042 \times 4.186 \times 40 \pm (0.01^2 + 0.007^2)^{1/2}$$

$$= 1.74 \text{ kW} \pm 1.22\%$$

A4:5.2.5 Error in Overall Thermal Conductance (U)

The calculation of U, in $kW.m^{-2}.^{o}C^{-1}$, is given by:-

$$U = \frac{Q}{\Delta t_m A} \qquad \text{(Eqn 4:1)}$$

where Q = heat flow into the water
= EXCHQ

A = heat exchanger area (m^2)

Δt_m = log mean temperature difference (^{o}C)

$$= \frac{(T_3 - T_{17}) - (T_4 - T_{16})}{\ln\left(\frac{T_3 - T_{17}}{T_4 - T_{16}}\right)}$$

Taking typical values for each variable and their errors gives:-

EXCHQ = 1.74 kW $\pm$ 1.22% (Section A4:5.2.5)

A = 0.84 m^2 $\pm$ 5.7% (area of primary heat exchanger for which fin height and tube diameter were measured to an accuracy of 0.5 mm, and length to an accuracy of $\pm$ 5mm)

$$T_3 = 90^{\circ}C \pm 0.5\% = 90 \pm 0.45^{\circ}C$$

$$T_4 = 35^{\circ}C \pm 0.5\% = 35 \pm 0.18^{\circ}C$$

$$T_{16} = 15^{\circ}C \pm 0.5\% = 15 \pm 0.08^{\circ}C$$

$$T_{17} = 55^{\circ}C \pm 0.5\% = 55 \pm 0.23^{\circ}C$$

from which:-

$$T_3 - T_{17} = 35 \pm 0.51^{\circ}C = 35^{\circ}C \pm 1.5\%$$

$$T_4 - T_{16} = 20 \pm 0.20^{\circ}C = 20^{\circ}C \pm 1.0\%$$

$$(T_3 - T_{17}) - (T_4 - T_{16}) = 15 \pm 0.55^{\circ}C = 15^{\circ}C \pm 3.7\%$$

$$\ln \left(\frac{T_3 - T_{17}}{T_4 - T_{16}} \right) = 0.56 \pm 1.8\%$$

$$\Delta t_m = 26.78^{\circ}C \pm 3.8\%$$

Evaluation of these variables resulted in a value for overall thermal conductance of:-

$$U = \frac{1.74}{0.84 \times 26.78}$$

$$\pm (0.012^2 + 0.057^2 + 0.038^2)^{1/2}$$

$$= 0.077 \text{ kW.m}^{-2}.^{\circ}C^{-1} \pm 7.0\%$$

APPENDIX A6

EXPERIMENT II DATA TABLES AND HEAT EXCHANGER MODEL

A6:1 DATA TABLES

(See following pages)

TABLE A6:1

Refrigerant flow rates ($l.min^{-1}$) data summary

800 LITRE LOAD

	W 12	W 10	W 7.5	W 6.5	A 12	A 10	A 7.5	A 6.5
MAX	2.35 $\pm$0.03	2.47 $\pm$0.03	2.41 $\pm$0.03	2.37 $\pm$0.03	2.26 $\pm$0.03	2.30 $\pm$0.03	2.33 $\pm$0.03	2.31 $\pm$0.03
MIN	1.91 $\pm$0.02	1.88 $\pm$0.02	1.90 $\pm$0.02	1.83 $\pm$0.02	1.81 $\pm$0.02	1.81 $\pm$0.02	1.86 $\pm$0.02	1.97 $\pm$0.02
MEAN	2.16 $\pm$0.04	2.18 $\pm$0.04	2.18 $\pm$0.04	2.16 $\pm$0.04	2.07 $\pm$0.04	2.10 $\pm$0.04	2.14 $\pm$0.04	2.18 $\pm$0.04

1200 LITRE LOAD

	W 12	W 10	W 7.5	W 6.5	A 12	A 10	A 7.5	A 6.5
MAX	2.06 $\pm$0.02	2.11 $\pm$0.03	2.19 $\pm$0.03	2.15 $\pm$0.03	2.08 $\pm$0.02	2.10 $\pm$0.03	2.13 $\pm$0.03	2.13 $\pm$0.03
MIN	1.76 $\pm$0.02	1.80 $\pm$0.02	1.87 $\pm$0.02	1.86 $\pm$0.02	1.78 $\pm$0.02	1.83 $\pm$0.02	1.76 $\pm$0.02	1.80 $\pm$0.02
MEAN	1.95 $\pm$0.04	1.98 $\pm$0.04	2.04 $\pm$0.04	2.02 $\pm$0.04	1.93 $\pm$0.04	1.99 $\pm$0.04	1.98 $\pm$0.04	2.01 $\pm$0.04

TABLE A6:2

Refrigerant flow ($1.min^{-1}$) data summary for 23°C and 18°C milk inlet temperature and 4°C and 7°C final milk temperature

800 LITRE LOAD

	W7.5	W7.5/7	W7.5/7/18	W7.5/18	A7.5	A7.5/18	W10	W10/18	A10	A10/18
MAX	2.41±0.03	2.33±0.03	2.16±0.03	2.18±0.03	2.33±0.03	2.14±0.03	2.47±0.03	2.23±0.03	2.30±0.03	2.13±0.03
MIN	1.90±0.02	2.01±0.02	2.03±0.02	1.89±0.02	1.86±0.02	1.82±0.02	1.88±0.02	1.85±0.02	1.81±0.02	1.78±0.02
MEAN	2.18±0.04	2.21±0.04	2.12±0.04	2.06±0.04	2.14±0.04	2.02±0.04	2.18±0.04	2.05±0.04	2.10±0.04	1.99±0.04

1200 LITRE LOAD

	W7.5	W7.5/7	W7.5/7/18	W7.5/18	A7.5	A7.5/18	W10	W10/18	A10	A10/18
MAX	2.19±0.03	2.18±0.03	2.04±0.02	2.01±0.02	2.13±0.03	2.01±0.02	2.11±0.03	1.93±0.02	2.10±0.03	1.95±0.02
MIN	1.87±0.02	2.01±0.02	1.95±0.02	1.87±0.02	1.76±0.02	1.71±0.02	1.80±0.02	1.78±0.02	1.83±0.02	1.72±0.02
MEAN	2.04±0.04	2.11±0.04	2.01±0.04	1.97±0.04	1.98±0.04	1.90±0.04	1.98±0.04	1.89±0.04	1.99±0.04	1.85±0.04

TABLE A6:3

Refrigeration effect ($kJ.kg^{-1}$) data summary
(Error = 2.52 $kJ.kg^{-1}$)

800 LITRE LOAD

	W 12	W 10	W 7.5	W 6.5	A 12	A 10	A 7.5	A 6.5
MAX	144.87	146.44	148.02	147.49	144.77	146.34	147.03	148.28
MIN	140.14	141.24	141.89	142.25	139.77	141.23	141.30	144.49
MEAN	142.91	144.03	145.30	145.28	142.61	144.35	144.44	146.76

1200 LITRE LOAD

	W 12	W 10	W 7.5	W 6.5	A 12	A 10	A 7.5	A 6.5
MAX	141.72	143.67	144.63	144.85	142.17	142.49	144.04	145.79
MIN	138.79	139.82	141.89	141.96	139.98	140.49	141.40	142.80
MEAN	140.42	141.89	143.34	143.54	141.28	141.55	142.95	144.58

TABLE A6:4

Refrigeration effect (kJ.kg^{-1}) data summary for various milk inlet and final temperatures
(Error = $\pm$2.52 kJ.kg^{-1})

800 LITRE LOAD

	W 7.5	W 7.5/7	W 7.5/7/18	W 7.5/18	A 7.5	A 7.5/18	W 10	W 10/18	A 10	A 10/18
MAX	148.02	146.72	144.63	147.74	147.03	145.79	146.44	143.36	146.34	143.90
MIN	141.89	143.20	142.82	141.95	141.30	143.49	141.24	140.42	141.23	140.78
MEAN	145.30	145.33	143.95	144.45	144.44	146.65	144.03	142.22	144.35	142.83

1200 LITRE LOAD

	W 7.5	W 7.5/7	W 7.5/7/18	W 7.5/18	A 7.5	A 7.5/18	W 10	W 10/18	A 10	A 10/18
MAX	144.63	145.01	143.57	143.20	144.04	144.08	144.63	141.72	142.49	142.97
MIN	141.89	143.05	142.52	141.74	141.40	141.13	139.82	139.76	140.49	140.26
MEAN	143.34	144.17	143.14	142.54	142.95	142.90	141.89	140.69	141.55	142.06

TABLE A6:5

Instantaneous power consumption data summary
(kW)

800 LITRE LOAD

	W 12	W 10	W 7.5	W 6.5	A 12	A 10	A 7.5	A 6.5
MAX	2.95 ±0.08	2.85 ±0.06	2.59 ±0.07	2.44 ±0.07	2.98 ±0.08	2.81 ±0.08	2.55 ±0.07	2.39 ±0.06
MIN	2.62 ±0.07	2.44 ±0.07	2.26 ±0.06	2.13 ±0.06	2.56 ±0.07	2.42 ±0.07	2.28 ±0.06	2.20 ±0.06
MEAN	2.82 ±0.07	2.65 ±0.07	2.45 ±0.07	2.30 ±0.07	2.77 ±0.07	2.63 ±0.07	2.43 ±0.07	2.32 ±0.07

1200 LITRE LOAD

	W 12	W 10	W 7.5	W 6.5	A 12	A 10	A 7.5	A 6.5
MAX	2.84 ±0.08	2.65 ±0.07	2.55 ±0.07	2.33 ±0.06	2.85 ±0.08	2.71 ±0.07	2.45 ±0.07	2.32 ±0.06
MIN	2.56 ±0.07	2.43 ±0.07	2.28 ±0.06	2.21 ±0.06	2.65 ±0.07	2.52 ±0.06	2.16 ±0.06	2.15 ±0.06
MEAN	2.70 ±0.07	2.56 ±0.07	2.39 ±0.07	2.28 ±0.07	2.73 ±0.07	2.61 ±0.07	2.34 ±0.07	2.24 ±0.07

TABLE A6.6

Instantaneous power consumption data summary for various milk inlet and final temperatures

800 LITRE LOAD

	W7.5	W7.5/7	W7.5/7/18	W7.5/18	A7.5	A7.5/18	W10	W10/18	A10	A10/18
MAX	2.59±0.07	2.58±0.07	2.44±0.07	2.39±0.06	2.55±0.07	2.43±0.06	2.85±0.08	2.65±0.07	2.81±0.08	2.68±0.07
MIN	2.26±0.06	2.37±0.06	2.38±0.06	2.22±0.06	2.28±0.06	2.22±0.06	2.44±0.07	2.44±0.07	2.42±0.07	2.43±0.07
MEAN	2.45±0.07	2.42±0.07	2.41±0.07	2.33±0.07	2.43±0.07	2.37±0.06	2.65±0.07	2.56±0.07	2.63±0.07	2.56±0.07

1200 LITRE LOAD

	W7.5	W7.5/7	W7.5/7/18	W7.5/18	A7.5	A7.5/18	W10	W10/18	A10	A10/18
MAX	2.55±0.07	2.47±0.07	2.42±0.07	2.40±0.06	2.45±0.07	2.40±0.06	2.65±0.07	2.55±0.07	2.71±0.07	2.58±0.07
MIN	2.28±0.06	2.35±0.06	2.34±0.06	2.21±0.06	2.16±0.06	2.19±0.06	2.43±0.06	2.33±0.06	2.52±0.06	2.29±0.06
MEAN	2.39±0.07	2.42±0.07	2.39±0.07	2.33±0.07	2.34±0.07	2.38±0.07	2.56±0.07	2.47±0.07	2.61±0.07	2.49±0.07

TABLE A6.7

COP data Summary

800 LITRE LOAD

	W 12	W 10	W 7.5	W 6.5	A 12	A 10	A 7.5	A 6.5
MAX	2.57 ±0.10	2.84 ±0.11	3.11 ±0.12	3.24 ±0.12	2.47 ±0.10	2.71 ±0.11	3.05 ±0.12	3.27 ±0.12
MIN	2.26 ±0.09	2.44 ±0.10	2.70 ±0.11	2.76 ±0.11	2.22 ±0.09	2.38 ±0.10	2.59 ±0.10	2.95 ±0.12
MEAN	2.43 ±0.10	2.66 ±0.11	2.92 ±0.12	3.08 ±0.12	2.39 ±0.10	2.59 ±0.10	2.88 ±0.11	3.14 +0.12

1200 LITRE LOAD

	W 12	W 10	W 7.5	W 6.5	A 12	A 10	A 7.5	A 6.5
MAX	2.35 ±0.10	2.56 ±0.10	2.82 ±0.11	2.97 ±0.12	2.34 ±0.09	2.52 ±0.10	2.84 ±0.11	3.04 ±0.12
MIN	2.13 ±0.09	2.33 ±0.09	2.63 ±0.11	2.70 ±0.11	2.10 ±0.08	2.29 ±0.09	2.60 ±0.10	2.71 ±0.11
MEAN	2.26 ±0.09	2.46 ±0.10	2.76 ±0.11	2.88 ±0.11	2.25 ±0.09	2.42 ±0.10	2.73 ±0.11	2.94 ±0.12

COP data summary for various milk inlet and final temperatures

800 LITRE LOAD

	W7.5	W7.5/7	W7.5/7/18	W7.5/18	A7.5	A7.5/18	W10	W10/18	A10	A10/18
MAX	3.11±0.12	3.02±0.12	2.82±0.11	3.06±0.12	3.05±0.12	2.92±0.11	2.84±0.11	2.69±0.11	2.71±0.11	2.61±0.10
MIN	2.70±0.11	2.73±0.11	2.73±0.11	2.73±0.11	2.59±0.10	2.67±0.11	2.44±0.10	2.39±0.10	2.38±0.10	2.33±0.09
MEAN	2.92±0.11	2.92±0.11	2.79±0.11	2.88±0.11	2.88±0.11	2.82±0.11	2.66±0.11	2.56±0.10	2.59±0.11	2.51±0.10

1200 LITRE LOAD

	W7.5	W7.5/7	W7.5/7/18	W7.5/18	A7.5	A7.5/18	W10	W10/18	A10	A10/18
MAX	2.82±0.11	2.88±0.11	2.72±0.11	2.97±0.12	2.84±0.11	2.77±0.11	2.56±0.10	2.46±0.10	2.52±0.10	2.47±0.10
MIN	2.63±0.11	2.74±0.11	2.67±0.11	2.69±0.11	2.60±0.10	2.49±0.10	2.33±0.09	2.37±0.09	2.29±0.09	2.29±0.09
MEAN	2.76±0.11	2.82±0.11	2.70±0.11	2.71±0.11	2.73±0.11	2.68±0.11	2.46±0.10	2.41±0.10	2.42±0.10	2.38±0.09

TABLE A6:9

COP data summary for with and without primary heat exchanger

800 LITRE LOAD

	A 12	A 12/NP	A 10	A 10/NP	A 7.5	A 7.5/NP
MAX	2.47 ±0.10	2.72 ±0.11	2.71 ±0.11	2.93 ±0.12	3.05 ±0.12	3.28 ±0.13
MIN	2.22 ±0.09	2.32 ±0.09	2.38 ±0.10	2.48 ±0.10	2.59 ±0.10	2.97 ±0.12
MEAN	2.39 ±0.10	2.55 ±0.10	2.59 ±0.10	2.74 ±0.11	2.88 ±0.11	3.16 ±0.13

1200 LITRE LOAD

	A 12	A 12/NP	A 10	A 10/NP	A 7.5	A 7.5/NP
MAX	2.34 ±0.09	3.08 ±0.12	2.52 ±0.10	2.70 ±0.11	2.84 ±0.11	3.08 ±0.12
MIN	2.10 ±0.08	2.27 ±0.09	2.29 ±0.09	2.39 ±0.09	2.60 ±0.10	2.91 ±0.11
MEAN	2.25 ±0.09	2.42 ±0.10	2.42 ±0.10	2.57 ±0.10	2.73 ±0.11	3.00 ±0.12

Heat recovery rate (kW) data summary

800 LITRE LOAD

	W 12	W 10	W 7.5	W 6.5	A 12	A 10	A 7.5	A 6.5
	(Model Results)							
MAX	2.11	1.79	1.40	1.19	1.74 ±0.02	1.55 ±0.02	1.27 ±0.02	1.15 ±0.02
MIN	2.01	1.73	1.32	1.14	1.66 ±0.02	1.38 ±0.02	1.15 ±0.02	1.05 ±0.01
MEAN	2.08	1.76	1.31	1.17	1.70 ±0.02	1.47 ±0.02	1.22 ±0.02	1.12 ±0.02

1200 LITRE LOAD

	W 12	W 10	W 7.5	W 6.5	A 12	A 10	A 7.5	A 6.5
	(Model Results)							
MAX	2.16	1.80	1.39	1.23	1.75 ±0.02	1.53 ±0.02	1.23 ±0.02	1.14 ±0.02
MIN	2.01	1.75	1.32	1.18	1.55 ±0.02	1.37 ±0.02	1.10 ±0.02	0.98 ±0.02
MEAN	2.13	1.78	1.36	1.21	1.65 ±0.02	1.46 ±0.02	1.18 ±0.02	1.09 ±0.02

6:2 DEVELOPMENT OF PRIMARY HEAT EXCHANGER MATHEMATICAL MODEL

A6:2.1 Aim

The aim of this model was to determine the water outlet temperature and heat flow for a heat exchanger, of known dimensions, transferring heat from refrigerant vapour to water, given the refrigerant vapour and water inlet temperatures and flow rates, as well as the refrigerant inlet and outlet pressures.

A6:2.2 Theory

A6:2.2.1 Determination of Film Coefficients and U Values

The change in water temperatures and the rate of heat flow in a heat exchanger is dependent upon the heat exchange area, overall thermal conductance (U) and the temperature regime.

The U value for any heat exchanger can be expressed as:-

$$U^{-1} = f_1^{-1} + C^{-1} + f_2^{-1} \qquad \text{.....Eqn A6:1}$$

where U = overall thermal conductance ($kW.m^{-2}.^{o}C^{-1}$)

f_1 & f_2 = film coefficients ($kW.m^{-2}.^{o}C^{-1}$)

C = the thermal conductance of the material separating the two films ($kW.m^{-2}.^{o}C^{-1}$)

In practice C is large relative to f and therefore can be neglected.

Rewriting Eqn A6:1 for a heat exchanger using superheated refrigerant vapour and water gives:-

$$U^{-1} = f_{wat}^{-1} + f_{ref}^{-1} \qquad \text{.....Eqn A6:2}$$

where f_{ref} = refrigerant film coefficient ($kW.m^{-2}.{}^{o}C^{-1}$)

f_{wat} = water film coefficient ($kW.m^{-2}.{}^{o}C^{-1}$)

Standard texts (knudsen and Katz, 1958) suggest that for an annulus:

$$f_{wat} = 0.023 \times \frac{k}{d_e} \times \left(\frac{\rho . d_e . V}{\mu}\right)^{0.8}$$

$$\times \left(\frac{C_p . \mu}{k}\right)^{0.4} \times \left(\frac{d_2}{d_1}\right)^{0.45} \qquad \text{.....Eqn A6:3}$$

where k = thermal conductivity ($kW.m^{-2}.{}^{0}C^{-1}$)

d_e = equivalent dimension = $d_2 - d_1$ for an annulus (m)

ρ = density ($kg.m^{-3}$)

V = velocity ($m.s^{-1}$)

μ = viscosity ($N.S.m^{-2}$)

Cp = specific heat ($kJ.kg^{-1}.{}^{o}C^{-1}$)

For a particular heat exchanger using water

$$V = \frac{M_w}{A}$$

where M_w = flow rate of water ($m^3.s^{-1}$)

A = cross sectional area of flow (m^2)

Then equation A6:3 can be rewritten as:-

$$f_{wat} = b^{-1}.M_w^{0.8} \qquad .....\text{Eqn A6:4}$$

where b^{-1} equals a constant dependent upon water properties and physical characteristics of the heat exchanger.

Substituting into Eqn A6:2 gives:-

$$U^{-1} = b.M_w^{-0.8} + f_{ref}^{-1} \qquad .....\text{Eqn A6:5}$$

This equation has the form of the straight line equation of $y = bx + c$. When $M_w = \infty$, $f_{ref}^{-1} = U^{-1}$.

On this basis it is possible to estimate f_{ref} under a particular set of refrigerant conditions by plotting U as a function of $M_w^{-0.8}$, and determining the intercepts. All other values must be kept constant. This is the method derived by Wilson (1915) and is known as the Wilson plot technique.

In general, variations in refrigerant flow rates and inlet temperatures will result in different f_{ref} values. For each refrigerant flow rate (M_r) the value of f_{ref} will be a function of the heat exchanger temperature regime since temperature affects vapour viscosity, specific heat and thermal conductivity. Thus values of f_{ref} for different refrigerant flow rates and inlet temperatures can be found by plotting the results of a series of experiments.

An attempt was made to predict values of f_{ref} using a simple linear model. It was assumed that f_{ref} was given by:-

$$f_{ref}^{-1} = k_1.t_m + k_2.M_r^{-1} + C \qquad .....\text{Eqn A6:7}$$

where k_1, k_2 and C are constants

t_m = mean inlet temperature

$$= \frac{(tri + twi)}{2}$$

M_r = refrigerant flow rate

Thus by plotting graphs of f_{ref}^{-1} against t_m at a constant flow rate k_1 can be determined, and likewise k_2 can be found from plots of f_{ref} against M_r^{-1} using values of f_{ref}^{-1} for $t_m = 0$ (i.e., values of the intercepts with the f_{ref} axis).

For the water side, the value of b in Eqn A6:4 will also vary with the temperature regime due to changes in viscosity and thermal conductivity. It was assumed that these variations in b can be expressed as:-

$$b = B.t_m + D \qquad .....\text{Eqn A6:8}$$

where B and D are constants.

Values of b may be obtained from the slope of U^{-1} against $M_w^{-0.8}$ plots.

Combining Eqns A6:5, A6:7 and A6:8,

$$U^{-1} = (B.t_m + D)\, M_w^{-0.8} + (k_1.t_m + k_2.M_r^{-1} + C)$$

.....Eqn A6:9

A6:2.2.2 Calculation of Effectiveness Ratio of Outlet Temperatures

The determination of U from experimentally determined f_{ref} values, at specified water flow rates, refrigerant and water inlet temperatures, allows the calculation of the Effectiveness Ratio of the heat exchanger (E_c) from the equation:-

$$Ec = \frac{1 - Exp(-NTU\ [1-C])}{1 - C \times Exp(-NTU\ [1-C])} \qquad \text{.....Eqn A6:10}$$

where NTU = number of transfer units

$$= \frac{UA}{C_{min}}$$

A = heat transfer area (m^2)

$$C = \frac{C_{min}}{C_{max}}$$

C_{min} = minimum specific heat X mass flow rate product ($kW.^oC^{-1}$)

C_{max} = maximum specific heat X mass flow rate product ($kW.^oC^{-1}$)

$Exp\ (n) = e^n$

Depending upon the mass flow rates either the water or the refrigernat vapour mass flow rate X specific heat product will be minimum while the other is a maximum. E_c can be used to find the ratio of the actual temperature change of one of the fluids to the maximum possible temperature change since

$$E_c = \frac{tri - tro}{tri - twi} \qquad \text{.....Eqn A6.11}$$

where tri = refrigerant inlet temperature (^{o}C)

tro = refrigerant outlet temperature (^{o}C)

twi = water inlet temperature (^{o}C)

rearranging Eqn A6:11 gives:-

$$tro = tri - E_c \,(tri - twi) \qquad \text{.....Eqn A6:12}$$

Once the refrigerant outlet temperature has been determined (Eqn A6:12), the heat flow from the superheated vapour can be calculated using the inlet temperature, refrigerant pressures and the superheat enthalpy equation (Eqn 4:9), i.e.,

$$Q_{ref} = M_r \,(h_1 - h_2) \qquad \text{.....Eqn A6:13}$$

where Q_{ref} = refrigerant heat flow (kW)

M_r = mass flow rate of the refrigerant ($kg.s^{-1}$)

h_1 = inlet specific enthalpy ($kJ.kg^{-1}$)

h_2 = outlet specific enthalpy ($kJ.kg^{-1}$)

Assuming the heat exchanger is insulated then the heat lost from the vapour will be absorbed by the water, from which the water outlet temperatures can be determined using the equation:-

$$two = twi + \frac{Q_{ref}}{M_w \times Sp\ Ht} \qquad \text{.....Eqn A6:14}$$

where two = water outlet temperature (^{o}C)

twi = water inlet temperature (^{o}C)

Sp Ht = specific heat of water ($kJ.kg^{-1}.^{o}C^{-1}$)

Q_{ref} = heat flow (kW)

M_w = water mass flow rate ($kg.s^{-1}$)

A6:2.3 Materials and Method

The experimental equipment described in Section 4:1 was used with the heat exchanger designed in Section 5:5.

Four water flow rates ranging from 0.8 $l.min^{-1}$ to 10 $l.min^{-1}$ were used with each of the three refrigerant flow rates (1.8, 2.0 and 2.3 $l.min^{-1}$) at four condenser pressures (6.5, 7.5, 10 and 12 bar). Water inlet temperatures were controlled so that condensation would not occur at high water flow rates.

The results of U were plotted against values of $M_w^{-0.8}$ from which the twelve values of b and f_{ref} were obtained using Equation A6:5. The values of b and f_{ref} were plotted to obtain relationships in terms of t_m (average of the two inlet temperatures), refrigerant and water flow rates. Once the full relationship for U was determined a computer program to determine water outlet temperatures and heat flows was developed using Eqns A6:10 to A6:14.

A6:2.4 Results

Results of M_r, $M_w^{-0.8}$, U^{-1}, h_{ref}^{-1}, t_m and b are presented in Table A6:11 and graphically in Figures A6:1 to A6:3.

TABLE A6:11

Data summary for Wilson plot test runs

Condenser Pressure (Bar)	M_r $(l.min^{-1})$	$M_w^{-0.8}$ $(min.l^{-1})$	t_m (^oC)	U^{-1} $(m^2.^oC.kW^{-1})$	h_{ref}^{-1} $(m^2.^oC.kW^{-1})$	k $(kJ.min^{-1}.m^{-2}.^oC^{-1})$
6.5	2.25	0.155-0.629	51.5	9.48-11.05	8.66	5.76
	2.03	0.155-1.064	51.5	9.95-15.83	8.99	6.43
	1.81	0.157-1.064	52.0	10.80-16.29	9.97	6.07
7.5	2.25	0.153-1.120	58.5	9.70-14.71	8.75	5.36
	2.03	0.198-1.010	56.5	10.32-14.35	9.13	5.65
	1.78	0.172-0.648	55.0	10.73-13.63	9.68	6.10
10.0	2.30	0.192-1.510	73.0	10.02-14.90	9.27	3.70
	2.03	0.185-0.911	68.0	10.79-13.48	10.10	3.70
	1.82	0.188-0.870	67.0	11.13-13.79	10.28	3.94
12.0	2.30	0.212-1.120	77.0	10.47-13.31	9.77	3.16
	2.00	0.212-1.064	75.0	10.64-12.90	10.42	2.88
	1.78	0.210-0.833	73.0	11.62-13.10	11.03	2.38

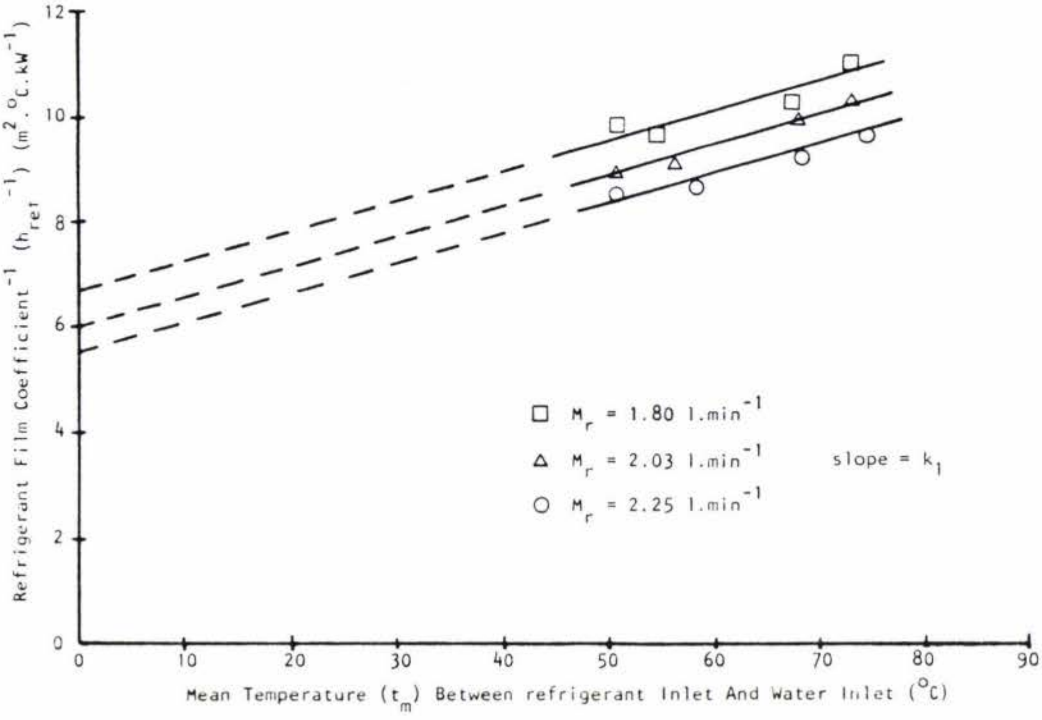


FIGURE A6:1

Hand fitted relationship between h_{ref}^{-1} and mean inlet temperature

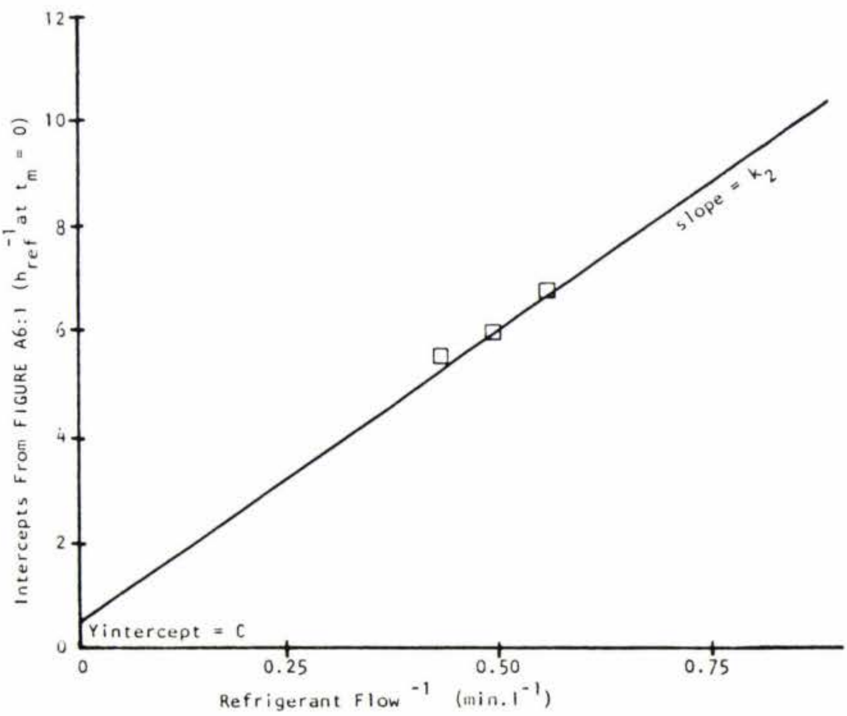


FIGURE A6:2

Relationship bewteen h_{ref}^{-1} at t_m=0 and refrigerant flow^{-1}

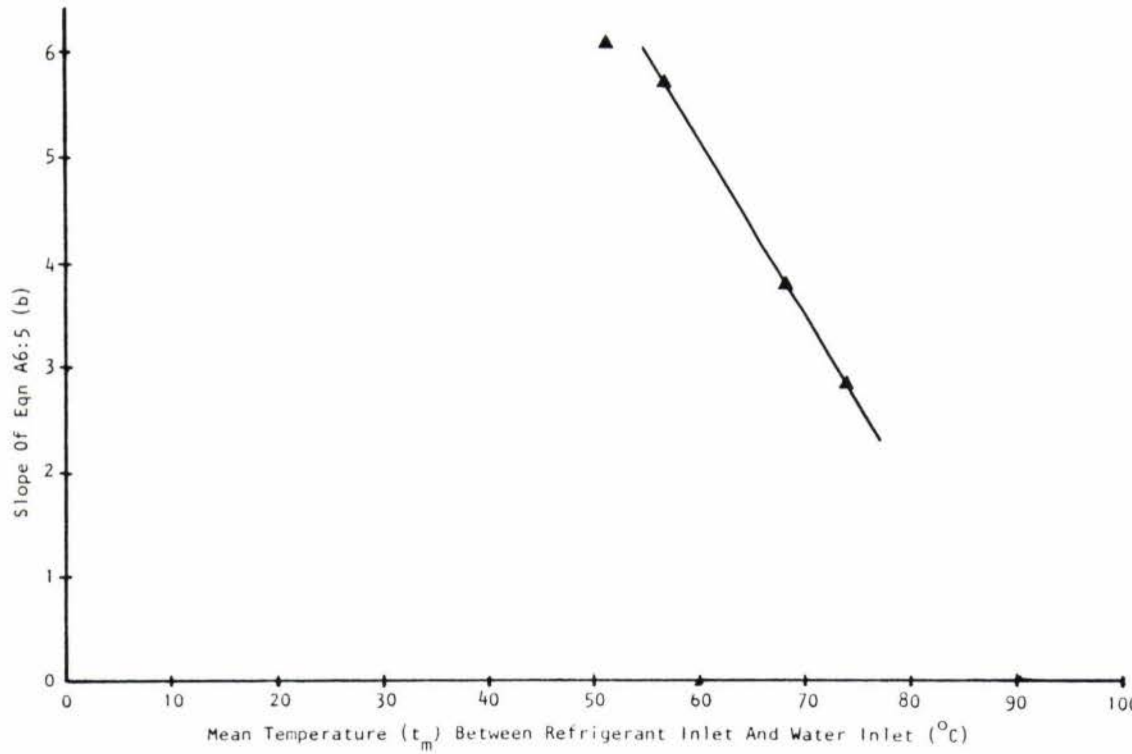


FIGURE A6:3

Hand fitted relationship between the slope of Eqn A6:5 and mean inlet temperature

Fom the slope of the graph of f_{ref}^{-1} against t_m (Figure A6:1), the value of k_1 was found to be 0.0575 (Eqn A6:7). Similarly the plot of f_{ref}^{-1} against M_r^{-1} gave values of $k_2 = 11.03$ and $C = 0.465$. The values of f_{ref}^{-1} were obtained from the intercept of the lines in Figure 6:1 and are thus the values of f_{ref}^{-1} when $t_m = 0$. Thus Eqn A6:7 may be written as:-

$$f_{ref}^{-1} = 0.0575\ t_m + 11.03\ M_r^{-1} + 0.465 \quad \text{.....Eqn A6:15}$$

The constants B and D in Eqn A6:8 were determined from plots of b against t_m (Figure 6:3) where the values of b were obtained from the plot of U^{-1} against $M_w^{-0.8}$. From the graph, B = -0.167 and D = 15.6. Substituting into Eqn A6:8 gives:-

$$b = 15.6 - 0.167\ t_m \quad \text{.....Eqn A6:16}$$

Thus substituting in Eqn A6:10 gives:-

$$U^{-1} = [(15.6 - 0.167t_m)\ M_w^{-0.8}]$$

$$+ [\ 0.0575t_m + (0.465 + 11.03M_r^{-1})] \quad \text{.....Eqn A6:17}$$

Calculation of water outlet temperatures and heat flows follows on from Eqn A6:10.

A6:2.5 Discussion

The computer program (Appendix A6:3) was tested using the data from all the water cooled system runs specified in Table 3:2. The results of the testing gave good agreement between the model and the experiment results for the water flow rate data used, with:-

$$two\ (model) = two\ (experimental) \pm 2.5^{\circ}C$$

$$Q_{ref}\ (model) = Q_{ref}\ (experimental) \pm 0.1\ kW$$

```
MODEL:PROCEDURE;
/***********************************************************************
*                                                                      *
*                  PRIMARY  HEAT  EXCHANGER  MODEL                     *
*                                                                      *
*   Aim:-      The aim of this program is to calculate heat exchanger  *
*              heat flows and water outlet temperature for a constant  *
*              water flow rate.                                        *
*                                                                      *
*   Input:-    Data for the water cooled condenser system runs of:     *
*                   1) Head,condenser and receiver pressure            *
*                   2) Water inlet temperature                         *
*                   3) Refrigerant inlet temperature                   *
*                   4) Refrigerant flow rate(receiver)                 *
*              was used.                                               *
*                                                                      *
*   Method:-   Using the model calculations developed in Appendix A6:1, *
*              an estimate of overall thermal conductance was made     *
*              from which values for Effectiveness coefficient (Ec),   *
*              refrigerant outlet temperature,heat flow and,finally,   *
*              water outlet temperature were determined for each row of *
*              data.                                                   *
*                                                                      *
*   Output:-   Results for both the original data and the model were   *
*              printed to a table file(MODRES) and to an input file    *
*              for MINITAB (MODMINI).                                  *
*                                                                      *
***********************************************************************/

                 /*****************************
                 *                           *
                 *  DECLARATION OF VARIABLES *
                 *                           *
                 *****************************/
```

```
DCL 1 DATA(20),                            /* Data structure - holds all the data */
    2 PRESSURE(3) FIXED DECIMAL(8,4),      /* pressures
                                              PRESSURE(1) = comp. outlet
                                              PRESSURE(2) = condenser
                                              PRESSURE(3) = receiver*/
    2 FLOWS(3) FLOAT DECIMAL(7),           /* flow rates
                                              FLOWS(1) = refrig. flow (l/min)
                                              FLOWS(2) = water flow (l/min)
                                              FLOWS(3) = refrig. flow (kg/s) */
    2 TEMPS(7) FLOAT DECIMAL(7),           /* temperatures
                                              TEMPS(1) = refrig. prim. inlet
                                              TEMPS(2) = refrig. prim. outlet
                                              TEMPS(3) = receiver
                                              TEMPS(4) = water prim. inlet
                                              TEMPS(5) = water prim. outlet
                                              TEMPS(6) = refrig. prim. outlet (model)
                                              TEMPS(7) = water prim. outlet (model) */
    2 PRIMEXCHANGER,
      3 EXCHQ(2) FIXED DECIMAL(6,3),       /* EXCHanger heat flows(Q) 1) from refrig.
                                                                      2) to water */
      3 LMTD FIXED DECIMAL(6,3),           /* Log Mean Temperature Difference */
      3 UA FIXED DECIMAL(7, 4),            /* overall thermal conductance(U) x area(A) */
      3 UAMOD FLOAT DECIMAL(7),            /* UA from model */
      3 CWAT FIXED DECIMAL(7,5),           /* specific heat(C) x mass WATer flow rate */
      3 CREF FIXED DECIMAL(7,5),           /* specific heat(C) x mass REFrig. flow rate */
      3 CMIN FIXED DECIMAL(7,5);           /* = CWAT if CWAT<CREF or CREF if CREF<CWAT */
      3 C FIXED DECIMAL(7,5);              /* C=CMIN/CMAX */
      3 NTU FIXED DECIMAL(6,3),            /* Number of Transfer Units */
      3 EC FIXED DECIMAL(6,4) ;            /* Effectiveness Coefficient */
DCL CMIN FIXED DECIMAL(7,5);               /* = CWAT if CWAT<CREF or CREF if CREF<CWAT */
DCL C FIXED DECIMAL(7,5);                  /* C=CMIN/CMAX */
DCL FILENAME(2) CHAR(80) VAR;              /* array for interactive naming of files */
DCL (Q1,Q2,
     FLOWIN,NROWS) FIXED DECIMAL(7,5);     /* Temp. diffs(Q1,Q2) water flow, number data rows */
DCL (I,J) FIXED;                           /* loop counters */
```

```
                         /****************************
                         *                           *
                         *   DECLARATION OF FILES    *
                         *                           *
                         ****************************/

DCL #1 FILE STREAM INPUT;                /* data input files */
DCL #2 FILE STREAM INPUT;                /* data input files */
DCL #3 FILE STREAM INPUT;                /* data input files */
DCL #4 FILE STREAM INPUT;                /* data input files */
DCL #5 FILE STREAM INPUT;                /* data input files */
DCL SCREEN FILE STREAM INPUT;            /* terminal input file */
DCL CONSOLE FILE STREAM OUTPUT;          /* terminal output file */
DCL MODRES FILE STREAM OUTPUT;           /* model results file (printer) */
DCL MODMINI FILE STREAM OUTPUT;          /* model results file (MINITAB) */

DCL FINISHED BIT(1) STATIC INIT('0');    /* end of file flag */

                         /****************************
                         *                           *
                         *   DECLARATION OF FUNCTIONS *
                         *                           *
                         *    Based on equations in  *
                         *    Appendix A4:4.1        *
                         ****************************/


          /**************************************
          *                                     *
          *   SATURATION PRESSURE FUNCTION      *
          *                                     *
          *    Calculates saturation            *
          *    pressure for corresponding       *
```

```
         *    temperature (oK)                     *
         *                                         *
         ******************************************/

SPRESS:PROCEDURE(SAT_TEMP) RETURNS(FIXED DECIMAL(6,4));

       DCL SAT_TEMP FIXED DECIMAL(5,2);
       DCL SAT_PRESS FIXED DECIMAL(6,4);
       SAT_PRESS=((SAT_TEMP-157.564)/85.69)**3.8226;

       RETURN(SAT_PRESS);

END SPRESS;

         /******************************************
         *                                         *
         *   SATURATION TEMPERATURE FUNCTION       *
         *                                         *
         *    Calculates saturation temperature    *
         *   from corresponding saturation         *
         *   pressure                              *
         *                                         *
         ******************************************/

STEMP:PROCEDURE(SH_PRESS) RETURNS(FIXED DECIMAL(5,2));

      DCL SH_PRESS FIXED DECIMAL(6,4);
      DCL SAT_TEMP FIXED DECIMAL(5,2);

      SAT_TEMP=85.963*(SH_PRESS)**.2616+157.564;

      RETURN (SAT_TEMP);

END STEMP;
```

```
/****************************************
*                                       *
*  VAPOUR SPECIFIC HEAT FUNCTION        *
*                                       *
*   Calculates the specific heat of     *
*   the superheated vapour.Requires     *
*   the superheat pressure.             *
*                                       *
****************************************/
CVAP:PROCEDURE(SH_PRESS) RETURNS(FIXED DECIMAL(6,4));

     DCL SH_PRESS FIXED DECIMAL(6,4);
     DCL VAP_C FIXED DECIMAL(5,4);

     VAP_C=0.61244*10**(0.008849*SH_PRESS);

     RETURN(VAP_C);

END CVAP;

/****************************************
*                                       *
*  LIQUID SPECIFIC HEAT FUNCTION        *
*                                       *
*   Calculates the specific heat of     *
*   the liquid.Requires the saturation  *
*   temperature and uses the            *
*   SATURATION PRESSURE FUNCTION.       *
*                                       *
****************************************/

CF:PROCEDURE(SAT_TEMP) RETURNS(FLOAT);

   DCL SAT_TEMP FIXED DECIMAL(5,2);
```

```
   DCL LIQ_C FLOAT DECIMAL;
   LIQ_C=0.888*10**(0.0066*SPRESS(SAT_TEMP));


   RETURN(LIQ_C);

END CF;

         /****************************************
         *                                       *
         *  LIQUID SPECIFIC VOLUME FUNCTION      *
         *                                       *
         *   Calculates the specific volume of   *
         *   the liquid.Requires the saturation  *
         *   temperature.                        *
         *                                       *
         ****************************************/

LVOL:PROCEDURE(SAT_TEMP) RETURNS(FLOAT DECIMAL);

     DCL SAT_TEMP FIXED DECIMAL(5,2);
     DCL LIQ_VOL FLOAT DECIMAL(5);
     DCL TEMP FIXED DECIMAL(5,2);

     TEMP=388.3-SAT_TEMP;
     LIQ_VOL=1/(558.085+(0.777*TEMP)+(17.943*(TEMP**0.5))+
             (117.436*(TEMP**0.333))-(3.40204E-4*(TEMP**2)));

     RETURN(LIQ_VOL);

END LVOL;
```

```
/****************************************
*                                       *
*   LIQUID ENTHALPY FUNCTION            *
*                                       *
*    Calculates the enthalpy of the     *
*    liquid.Requires temperature and    *
*    uses LIQUID SPECIFIC HEAT,LIQUID   *
*    SPECIFIC VOLUME and SATURATION     *
*    PRESSURE FUNCTIONS.                *
*                                       *
****************************************/

HF:PROCEDURE(SAT_TEMP) RETURNS(FIXED DECIMAL(6,3));

   DCL SAT_TEMP FIXED DECIMAL(5,2);
   DCL LIQ_ENTH FIXED DECIMAL(6,3);

   LIQ_ENTH=0.888*10**(0.0021*SPRESS(SAT_TEMP)))*(SAT_TEMP-233)+
            (LVOL(SAT_TEMP)+6.556E-4)/2*(SPRESS(SAT_TEMP)-0.6417);

   RETURN(LIQ_ENTH);

END HF;

/****************************************
*                                       *
*   LIQUID/VAPOUR ENTHALPY FUNCTION     *
*                                       *
*    Calculates the enthalpy change     *
*    for vaporization. Requires the     *
*    SATURATION PRESSURE FUNCTION and   *
*    the saturation temperature.        *
*                                       *
```

```
        *****************************************/

HFG:PROCEDURE(SAT_TEMP) RETURNS(FIXED DECIMAL(6,3));

    DCL SAT_TEMP FIXED DECIMAL(5,2);
    DCL FG_ENTH FIXED DECIMAL(6,3);

    FG_ENTH=165.063*10**(-0.011*SPRESS(SAT_TEMP));

    RETURN(FG_ENTH);

END HFG;
        /****************************************
        *                                       *
        *   SUPERHEAT ENTHALPY FUNCTION         *
        *                                       *
        *    Calculates the enthalpy of the     *
        *    superheated vapour.Requires the    *
        *    LIQUID ENTAHLPY,LIQUID/VAPOUR      *
        *    and LIQUID SPECIFIC HEAT           *
        *    FUNCTIONS.  Superheat temperature  *
        *    and pressure are also required.    *
        *                                       *
        *****************************************/

HSH:PROCEDURE(SH_TEMP,SH_PRESS)RETURNS(FIXED DECIMAL(6,3));

   DCL SH_TEMP FIXED DECIMAL(5,2);
   DCL SH_PRESS FIXED DECIMAL(6,4);
   DCL SH_ENTH FIXED DECIMAL(6,3);
   DCL SAT_TEMP FIXED DECIMAL(5,2);

   SAT_TEMP= STEMP(SH_PRESS);

   SH_ENTH=HF(SAT_TEMP)+HFG(SAT_TEMP)+(SH_TEMP-SAT_TEMP)
```

```
            *CVAP(SH_PRESS);

   RETURN(SH_ENTH);

END HSH;

        /****************************************
        *                                       *
        *   SUBCOOLED ENTHALPY FUNCTION         *
        *                                       *
        *    Calculates the enthalpy of the     *
        *    subcooled liquid.Requires LIQUID   *
        *    SPECIFIC HEAT and SATURATION       *
        *    TEMPERATURE FUNCTIONS and          *
        *    subcooled temperature and          *
        *    pressure.                          *
        *                                       *
        ****************************************/

HSC:PROCEDURE(SC_TEMP,SC_PRESS)RETURNS(FIXED DECIMAL(6,3));

   DCL SAT_TEMP FIXED DECIMAL(5,2);
   DCL SC_PRESS  FIXED DECIMAL(6,4);
   DCL SC_TEMP  FIXED DECIMAL(5,2);
   DCL SC_ENTH FIXED DECIMAL(6,3);

   SAT_TEMP=STEMP(SC_PRESS);

   SC_ENTH=HF(SAT_TEMP)-(SAT_TEMP-SC_TEMP)*CF(SAT_TEMP);

   RETURN(SC_ENTH);

END HSC;
                           /******************************
                           *                             *
```

```
                          *       MAIN PROGRAM          *
                          *                             *
                          *******************************/

/*Obtain file names from terminal*/
OPEN FILE(SCREEN) TITLE('TTY -DEVICE');
OPEN FILE(CONSOLE) TITLE('TTY -DEVICE');
PUT FILE(CONSOLE) LIST('Enter filenames ,design flows and number of rows');
GET FILE(SCREEN) SKIP LIST(FILENAME,FLOWIN,NROWS);
PUT FILE(CONSOLE) LIST('File names are:',FILENAME);
CLOSE FILE(CONSOLE);
CLOSE FILE(SCREEN);

/*Open remaining files */
OPEN FILE(MODRES) TITLE(FILENAME(1)) PRINT PAGESIZE(60) LINESIZE(132);
OPEN FILE(#1) TITLE(FILENAME(2));
OPEN FILE(#2) TITLE(FILENAME(3));
OPEN FILE(#3) TITLE(FILENAME(4));
OPEN FILE(#4) TITLE(FILENAME(5));
OPEN FILE(#5) TITLE(FILENAME(6));
OPEN FILE(MODMINI) TITLE(FILENAME(7));


/*Set end of file condition */
ON ENDFILE(#5) FINISHED='1'B;


/*Read in data while files not empty */
DO WHILE(^FINISHED);
   GET FILE(#1) EDIT((PRESSURE(I,1),PRESSURE(I,2),PRESSURE(I,3)
                    DO   I=1 TO NROWS))
                    (X(48),2 F(6,2),F(5,2),SKIP);

   GET FILE(#2) EDIT((TEMPS(I,1),TEMPS(I,2),TEMPS(I,3) DO I=1 TO NROWS))
                    (X(14),2 F(6,2),X(24),F(6,2),SKIP);
```

```
   GET FILE(#3) EDIT((TEMPS(I,4),TEMPS(I,5) DO I=1 TO NROWS))
                    (X(31),2 F(6,2),SKIP);

   GET FILE(#4) EDIT(((FLOWS(I,J) DO J=1 TO 3)DO I=1 TO NROWS))
                    (F(6,2),X(7),F(4,2),F(7,4),SKIP);

   GET FILE(#5) EDIT((UA(I),CREF(I) DO I=1 TO NROWS))
                    (X(48),F(5,3),X(7),F(5,3),SKIP);

   GET FILE(#5) SKIP(2);

END;

/*Repeat model calculations for each row of data */
DO I=1 TO NROWS;

   /*Set water flow rate */
   FLOWS(I,2)=FLOWIN;

   /*Calculate UA model */
   UAMOD(I)=(1/((((0.465+11.03*(1/FLOWS(I,1)))+
                (0.0575*((TEMPS(I,1)+TEMPS(I,4))/2)))+
                ((15.16-0.167*((TEMPS(I,1)+TEMPS(I,4))/2))*
                 FLOWS(I,2)**-0.8))*0.84);

   /*Calculate CMIN and C  and then NTU and EC */
   CWAT(I)=(FLOWS(I,2)/60.0*4.186);

   IF CWAT(I) >= CREF(I)
   THEN DO;
          C=(CREF(I)/CWAT(I));
          CMIN=CREF(I);
        END;
   ELSE DO;
```

```
          C=(CWAT(I)/CREF(I));
          CMIN=CWAT(I);
        END;

   NTU(I)=(UAMOD(I)/CMIN);

   EC(I)=((1-EXP(-NTU(I)*(1-C)))/(1-C*EXP(-NTU(I)
         *(1-C))));

   /*Calculate refrig. prim. outlet temperature */
   TEMPS(I,6)=(TEMPS(I,1)-EC(I)*(TEMPS(I,1)-TEMPS(I,4)));

   /*Calculate heat flow in primary */
   EXCHQ(I,2)=(FLOWS(I,3)*(HSH((TEMPS(I,1)+273),PRESSURE(I,1))-HSH
                          ((TEMPS(I,6)+273),PRESSURE(I,2))));

   /*Calculate water outlet temperature (model) */
   TEMPS(I,7)=(EXCHQ(I,2)/4.186/FLOWS(I,2)*60.0 + TEMPS(I,4));

   /*Calculate water heat flow (check) */
   EXCHQ(I,1)=(FLOWS(I,2)/60.0*4.186*(TEMPS(I,7) - TEMPS(I,4)));

   /*Calculate Log Mean Temperature Difference */
   Q1= (TEMPS(I,1)-TEMPS(I,7));
   Q2= (TEMPS(I,6)-TEMPS(I,4));
   IF (Q1>Q2) & (Q1>0 & Q2>0)
   THEN LMTD(I)=(Q1-Q2)/LOG((Q1/Q2));
   ELSE IF (Q1<Q2) & (Q1>0 & Q2>0)
        THEN LMTD(I)=(Q2-Q1)/LOG((Q2/Q1));
        ELSE LMTD(I)=Q1;


END;

/*Print results to output files (MODRES & MODMINI) */
```

```
/*Page 1 MODRES */
PUT FILE(MODRES) PAGE;
PUT FILE(MODRES) EDIT('PRESSURES','REF IN','REF OUT','RECEIVER')
                   (SKIP(2),COL(25),A,SKIP(2),COL(5),
                    X(4),A,X(13),A,X(12),A);
PUT FILE(MODRES) EDIT(((PRESSURE(I,J) DO J=1 TO 3)
                                   DO I=1 TO NROWS))
                   (SKIP(2),COL(5),3 (F(10,2),X(10)));

/*Page 2 MODRES */
PUT FILE(MODRES) PAGE;
PUT FILE(MODRES) EDIT('FLOWS','TEMPERATURES')(SKIP(2),COL(10),
                                             A,COL(80),A);
PUT FILE(MODRES) EDIT('REF FLOW','HOT WAT FLOW','REF MASS FLOW',
                    'REFIN','REFOUT','RECEIV','WATIN','WATOUT',
                    'REFOUTm','WATOUTm')(SKIP(5),COL(1),
                     A,X(5),A,X(3),A,X(6),A,X(7),A,X(7),A,X(8),A,
                     X(7),A,2 (X(6),A));

PUT FILE(MODRES) EDIT(((FLOWS(I,J) DO J=1 TO 3),(TEMPS(I,J)
                        DO J=1 TO 7) DO I=1 TO NROWS))
          (SKIP(2),10 (X(3),F(10,4)));

/*Page 3 MODRES */
PUT FILE(MODRES) PAGE;
PUT FILE(MODRES) EDIT('HEAT FLOW','LMTDm','UA','UAm','CWAT','CREF',
           'NTUm','ECm')(SKIP(5),COL(1),
            X(6),A,X(10),A,X(13),A,X(12),A,3 (X(11),A),
            X(12),A);
PUT FILE(MODRES) EDIT((EXCHQ(I,2),LMTD(I),UA(I),UAMOD(I),CWAT(I),
            CREF(I),NTU(I),EC(I) DO I=1 TO NROWS))
          (SKIP(2),COL(5),8 (F(10,4),X(5)));
```

```
/*MINITAB file */
PUT FILE(MODMINI) EDIT(((TEMPS(I,J) DO J=1 TO 7),UAMOD(I),EC(I),
             EXCHQ(I,1),LMTD(I) DO I=1 TO NROWS))
           (COL(1),7 F(7,2),3 F(7,4),F(7,3),SKIP);

/*Close output files */
CLOSE FILE(MODRES);
CLOSE FILE(MODMINI);


END MODEL;
```

APPENDIX A7

EXPERIMENT III DATA TABLES

TABLE A7:1

Vat cylinder water heating power consumption (3 x 1 week)

Day	Volume of Water Entering Cylinder ($l.day^{-1}$)	Heating Load			Standing Load		Total Power (kWh)	% Standing
		Time (h)	Power (kWh)	$l.(kWh)^{-1}$	Time (h)	Power (kWh)		
4 *	3	-	-	-	2.20	5.72	5.72	100.0
5 **	39	4.60	9.60	4.1	2.06	4.30	13.90	30.9
6 *	5	-	-	-	2.00	5.52	5.52	100.0
7	155	5.13	11.59	13.4	1.70	3.84	15.43	24.9
8	177	5.86	13.02	13.6	2.20	4.89	17.91	27.3
9	146	4.87	10.57	13.8	2.26	4.90	15.47	31.7
10 *	10	-	-	-	2.23	5.13	5.13	100.0
25	158	4.60	9.60	16.5	2.00	4.17	13.77	30.3
26	54	1.66	3.77	14.3	2.30	5.22	8.99	58.1
27 ***	20294	14.86	34.18	593.7	0.90	2.07	36.25	5.7
28	0	4.73	9.79	-	2.40	4.97	14.67	33.7
29	0	-	-	-	2.26	6.16	6.16	100.0
30	128	4.26	9.50	13.5	2.16	4.82	14.32	37.1
31	0	-	-	-	2.30	5.21	5.21	100.0
39	126	2.53	6.74	18.7	2.20	5.86	12.60	46.5
40	54	1.67	3.98	13.6	2.57	6.12	10.10	60.0
41	42	1.40	3.07	13.7	2.33	5.12	8.19	62.5
42	80	2.67	5.83	13.7	2.33	5.08	10.91	46.6
43	0	-	-	-	1.70	4.33	4.33	100.0
44	71	2.53	5.42	13.1	2.03	4.35	9.77	44.5
45 **	43	0.73	1.72	25.0	1.87	4.41	6.13	71.9

* Assumed as zero volume

** Rejected on the basis of $l.(kWh)^{-1}$ less than 10

*** Rejected on the basis of $l.(kWh)^{-1}$ greater than 20

TABLE A7:2

Machine cylinder water heating power consumption (3 x 1 week)

Day	Volume of Water Entering Cylinder (litres)		Heating Load						Standing Load		Total Power	% Standing
			Morning			Evening						
	Morning	Evening	Time (h)	Power (kWh)	$1.(kWh)^{-1}$	Time (h)	Power (kWh)	$1.(kWh)^{-1}$	Time (h)	Power (kWh)	(kWh)	
4	183	98	8.20	18.60	9.8	4.20	9.53	10.3	1.56	3.54	31.61	11.2
5	109	133	4.93	11.20	9.7	4.80	10.90	12.2	1.83	4.16	26.26	15.8
6	115	121	6.13	14.30	8.0	5.40	12.59	9.6	1.83	4.27	31.16	13.7
7	110	100	4.73	11.74	9.4	4.40	10.92	9.2	1.90	4.71	27.37	17.2
8	98	104	5.13	10.86	9.0	4.86	10.92	10.1	2.20	4.66	25.80	18.0
9	104	79	4.60	10.31	10.1	3.66	8.21	9.6	2.40	5.38	23.90	22.5
10	116	101	4.10	9.92	11.7	4.73	11.33	8.9	2.13	5.10	26.25	19.4
25	119	145	5.33	12.50	9.5	6.13	14.37	10.1	1.63	3.82	30.69	12.5
26	87	88	4.00	9.19	9.5	4.06	9.32	9.4	2.13	4.89	23.40	20.9
27	97	90	4.40	10.21	9.5	4.00	9.28	9.7	2.26	5.24	24.73	21.2
28	156	104	5.80	13.12	11.9	4.80	10.86	9.6	1.86	4.21	28.18	14.9
29	104	92	4.80	11.12	9.4	4.26	9.87	9.3	1.80	4.17	25.16	16.6
30	101	203	4.67	10.45	9.7	6.40	14.33	14.2	1.80	4.03	28.81	14.0
31	-	36	-	-	-	1.50	3.79	9.4	2.40	6.07	9.86	61.5
39	119	89	5.47	12.66	9.4	3.80	8.80	10.1	2.07	4.79	26.25	18.3
40	88	140	4.00	9.13	9.6	6.00	13.69	10.2	1.87	4.27	27.08	15.8
41	125	158	5.27	12.03	10.4	5.67	12.94	12.2	1.80	4.11	29.07	14.1
42 *	263	162	5.80	13.29	19.8	7.67	17.57	9.2	1.40	3.21	34.06	9.4
43 *	124	362	5.53	12.68	9.8	7.33	16.80	21.5	1.23	2.82	32.30	8.7
44	114	107	5.13	11.78	9.7	4.67	10.72	10.0	1.77	4.06	26.57	15.3
45	102	95	4.53	10.44	9.8	4.33	9.98	9.5	2.07	4.77	25.19	18.9

* Rejected on the basis of $1.(kWh)^{-1}$ greater than 15

TABLE A7:3

Summary of total, water heating and refrigeration power consumption (kWh) for a 50 day monitoring period

Day	Total Farm	Water Heating Consumption				Refrigeration Consumption			
		Machine	Vat	Total	%Farm Total	3HP	5HP	Total	%Farm Total
1	69.3	34.07	10.32	44.39	64.1	2.37	15.21	17.58	25.4
2	100.8	25.19	4.59	29.78	29.5	13.5	-	13.50	13.4
3	129.9	32.02	14.68	46.70	36.2	12.87	3.36	16.23	12.5
4	103.2	31.67	5.72	37.39	36.2	15.69	3.69	19.38	18.8
5	135.6	26.26	13.91	40.17	29.6	8.67	6.84	15.15	11.2
6	128.7	31.16	5.52	36.68	28.5	11.79	-	11.79	9.2
7	86.1	27.37	15.43	42.80	49.7	12.39	5.07	17.46	20.3
8	92.4	25.80	17.91	43.71	47.3	11.91	4.56	16.47	17.8
9	120.6	23.90	15.47	39.37	32.6	8.52	5.34	13.86	11.5
10	67.8	26.25	5.13	31.38	46.3	8.07	6.12	14.19	20.9
11	102.0	24.58	10.71	35.29	34.6	1.41	13.29	14.70	14.4
12	64.8	25.82	5.14	30.96	47.7	-	14.97	14.97	23.1
13	68.7	26.01	11.61	37.62	54.7	13.11	-	13.11	19.1
14	98.7	29.53	10.52	40.05	40.6	11.40	-	11.40	11.6
15	101.1	26.51	5.75	32.26	31.9	-	15.18	15.18	15.0
16	96.9	24.31	13.11	37.42	38.6	9.90	-	9.90	10.2
17	53.4	27.97	4.13	32.10	60.1	8.13	5.94	14.07	26.3
18	106.2	21.87	9.78	31.65	29.8	1.53	13.44	14.97	14.1
19	107.4	34.57	17.15	51.72	48.2	11.52	1.32	12.84	12.0
20	68.7	26.10	10.33	36.43	53.0	15.72	2.19	17.91	26.0
21	101.4	27.25	4.07	31.32	30.1	14.13	-	14.13	13.9
22	-	-	-	-	-	-	-	-	-
23	-	-	-	-	-	-	-	-	-
24	63.3	16.11	13.49	29.60	46.8	-	14.97	14.97	23.6
25	99.0	30.69	13.77	44.46	44.9	11.31	-	11.31	11.4
26	69.6	23.40	8.99	32.39	46.5	-	12.18	12.18	17.5
27	91.5	24.73	36.25	60.98	66.6	-	12.84	12.84	14.0
28	103.8	28.18	14.76	42.94	41.4	9.96	-	9.96	9.6
29	68.7	25.16	6.61	31.77	46.2	-	14.70	14.70	21.4
30	99.0	28.81	14.32	43.13	43.6	9.84	-	9.84	9.9
31	67.2	9.86	5.21	15.07	22.4	-	8.10	8.10	12.1
32	103.8	30.80	10.57	41.37	39.9	-	13.47	13.47	13.0
33	93.6	32.39	6.50	38.89	41.5	11.76	-	11.76	12.6
34	101.4	26.51	10.69	37.20	36.7	-	14.58	14.58	14.4
35	99.0	28.13	12.34	40.47	40.8	-	12.63	12.63	12.6
36	67.5	31.20	4.68	35.88	53.1	8.82	-	8.82	13.1
37	67.5	31.22	3.55	34.77	51.5	7.35	4.35	11.70	17.3
38	101.7	20.18	10.79	30.97	30.4	6.69	6.48	13.17	12.9
39	90.0	26.25	12.60	38.85	43.2	-	12.69	12.69	14.1
40	103.2	27.08	10.10	37.18	36.0	-	13.11	13.11	12.7
41	-	29.07	8.19	37.26	-	-	16.56	16.56	-
42	96.9	34.06	10.91	44.97	46.4	-	9.03	9.03	9.3
43	66.3	32.30	4.33	36.63	55.2	13.11	-	13.11	19.8
44	101.1	26.57	9.77	36.34	35.9	11.79	-	11.79	11.7
45	63.9	25.19	6.13	31.32	49.0	-	13.53	13.53	21.2
46	92.9	26.30	7.70	34.00	36.5	-	12.21	12.21	13.1
47	132.6	29.65	15.48	45.13	34.0	-	14.91	14.91	11.2
48	64.2	25.27	7.45	32.72	51.0	-	15.51	15.51	24.2
49	68.7	25.72	7.94	33.66	49.0	-	14.25	14.25	20.7
50	69.0	23.69	21.84	45.53	66.0	-	14.19	14.19	20.6

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