DEVELOPMENT OF AN ELECTRIC HEAT PUMP FOR DOMESTIC USE

by

DAVID ROBERT CROSSMAN

Doctor of Philosophy

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SUMMARY

This thesis records the design and development of an electrically driven, air to water, vapour compression heat pump of nominally 6kW heat output, for residential space heating. The study was carried out on behalf of GEC Research Ltd through the Interdisciplinary Higher Degrees Scheme at Aston University.

A computer based mathematical model of the vapour compression cycle was produced as a design aid, to enable the effects of component design changes or variations in operating conditions to be predicted. This model is supported by performance testing of the major components, which revealed that improvements in the compressor isentropic efficiency offer the greatest potential for further increases in cycle COP_h.

The evaporator was designed from first principles, and is based on wire- wound heat transfer tubing. Two evaporators, of air side area 10.27 and 16.24m^2, were tested in a temperature and humidity controlled environment, demonstrating that the benefits of the large coil are greater heat pump heat output and lower noise levels. A systematic study of frost growth rates suggested that this problem is most severe at the conditions of saturated air at 0°C combined with low condenser water temperature.

A dynamic simulation model was developed to predict the in-service performance of the heat pump. This study confirmed the importance of an adequate radiator area for heat pump installations.

A prototype heat pump was designed and manufactured, consisting of a hermetic reciprocating compressor, a coaxial tube condenser and a helically coiled evaporator, using Refrigerant 22. The prototype was field tested in a domestic environment for 1½ years. The installation included a comprehensive monitoring system. Initial problems were encountered with defrosting and compressor noise, both of which were solved. The unit then operated throughout the 1985/86 heating season without further attention, producing a COP_h of 2.34.

KEY WORDS: HEAT PUMPS, RESIDENTIAL SPACE HEATING, DESIGN, MODELLING, FIELD TESTING.

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NOTATION

Abbreviations

CEC	Commission of the European Communities
COP_h	heat pump coefficient performance
CTR	Component Test Rig
dB(A)	decibels, A weighting
DSI	Dynamic Simulation of the Interaction Between the Heat Pump, Heat Distribution System and the Building (computer program)
ERC	Engineering Research Centre
ETR	Evaporator Test Rig
GEC	General Electric Company
HP	high pressure
IOL	internal overload (in compressor motor)
LP	low pressure
PID	proportional, integral and derivative
PRT	platinum resistance thermometer
RCCB	residual current circuit breaker
TEV	thermostatic expansion valve
VCCM	Vapour Compression Cycle Model (computer program)

Symbols

a	Numerical constant in the empirical equation of the compressor volumetric efficiency characteristic
A	area
A_{ef}	effective total area of a finned tube
$A_{f,m}$	mean face area of evaporator coil
$A_{f,o}$	outside face area of evaporator coil
A_m	mean area of tube wall
A_p	primary heat transfer area on the refrigerant side of the condenser, i.e. not including fin area.
A_s	fin area
$A_{t,m}$	mean transverse air flow area of the evaporator
A_x	fin cross-sectional area

b	numerical constant in the empirical equation of the compressor volumetric efficiency characteristic
$B'_{a,f}$	evaporator air side blockage area per unit length of tube
Bo	Boiling number
c	numerical constant in the empirical equation of the compressor isentropic efficiency characteristic
C	numerical constant used in transducer calibration equations, see for example table 4.9
C_1	constant in the Nusselt equation
C	heat capacity
Co	Convection number
Cp	isobaric specific heat capacity
d	numerical constant in the empirical equation of the compressor isentropic efficiency characteristic or diameter
d_1 to d_4	evaporator; see figure 4.1 or condenser; see figure 6.4
d_e	effective diameter of finned condensing surface
E	heat exchanger effectiveness
f	friction factor
Fr_l	Froude number of liquid phase
g	acceleration due to gravity
h	height of evaporator or specific enthalpy or heat transfer coefficient
h_{2s}	specific enthalpy at compressor discharge following isentropic, adiabatic compression
h_f	condenser water side fouling coefficient
H_a	power of immersion heaters in the heat source circuit of the CTR
I	thermodynamic irreversibility or electrical current
k	thermal conductivity
l	length or heat exchanger pass length
L_{mf}	mean length of fin
m	fin height

$\dot{m}$	mass flow rate
M	numerical constant used in transducer calibration equations, see for example table 4.9 or mass
N_c	number of refrigerant circuits in the evaporator
N_l	number of fin loops per revolution of the evaporator heat transfer tubing
NTU	number of heat transfer units
Nu	Nusselt number
p	fin perimeter
P	pressure
Pr	Prandlt number
$\dot{q}$	heat transfer rate
$\dot{q}_b$	heat output rate from supplementary heater
$\dot{q}_c$	heat output rate from the condenser
$\dot{q}_{c,c}$	heat transfer rate across the two phase region of the condenser
$\dot{q}_{c,d}$	heat transfer rate across the desuperheater region of the condenser
$\dot{q}_e$	heat transfer rate across the evaporator
$\dot{q}_{e,1}$	heat transfer rate across the two phase region of the evaporator
$\dot{q}_{e,2}$	heat transfer rate across the superheating region of the evaporator
Q	quantity of heat
R	electric resistance
R_{ref}	resistance of reference resistor
Re	Reynolds number
R_f	relative roughness, i.e. the ratio of the actual friction factor to that of a plain, smooth tube of the same diameter
s	specific entropy or tube spacing
S	entropy
t	time or thickness
T	temperature (in°C or K depending on context)
T_a	ambient air temperature

T_{a1}, T_{a2}	air temperature at the evaporator intake and discharge, respectively
T_{ar}	room air temperature
T_1 to T_4	refrigerant temperatures at the four state points
T_c	condensing temperature
T_d	dew point temperature
T_e	evaporating temperature
T_s	surface temperature
TSAT(P)	saturated temperature corresponding to pressure P
T_{w1} to T_{w3}	water temperatures at condenser inlet, condenser outlet and supplementary heater outlet respectively
u	velocity
u_f	evaporator air side mean face velocity
u_t	evaporator air side transverse velocity
U	overall heat transfer coefficient
$U_{c,i}$	overall heat transfer coefficient for the two phase region in the condenser, referred to the water side area.
$U_{d,i}$	overall heat transfer coefficient for the desuperheating region in the condenser, referred to the water side area
v	specific volume
$\dot{V}$	volumetric flow rate
$\dot{V}_{sw}$	compressor displacement rate
V	transducer voltage signal. Table 4.9 refers
w_1, w_2	fin pitch for evaporator tubing. Figure 4.1 refers
w	refrigerant fraction in the refrigerant/oil liquid mixture
W	electricity consumed by the heat pump, including compressor and fan ($\dot{W}$ = electrical power absorbed)
W_b	electricity consumed by the flow boiler
W_c	electricity consumed by the compressor motor
W_f	electricity consumed by the fan motor
W_h	electricity consumed by the compressor and fan with the heat pump in heating mode
x	thermodynamic quality
X	oil fraction in the total mixture
Z	liquid fraction in the total mixture or elevation

β	variable used in fin efficiency calculation
$\delta\epsilon$	temperature offset between two matched thermometers
μ	dynamic viscosity
Δ	difference
ΔP	pressure drop
ΔP_{a1}	air side pressure drop through evaporator
ΔP_{a2}	air side pressure drop through ductwork
ΔT_{LM}	Log mean temperature difference
η	efficiency or effectiveness
η_a	surface effectiveness
η_f	fin efficiency
η_{fm}	combined efficiency of fan and fan motor
ρ	density
Ψ	two phase evaporator heat transfer enhancement factor

Subscripts

a	air
b	supplementary heater or building
c	condenser, condensing or reversed Carnot cycle
d	desuperheating or heating requirement
e	evaporator, evaporating
f	face of evaporator heat exchanger or fin
g	enthanediol solution or heat gain
i	internal
is	isentropic
l	liquid refrigerant or loss
liq	liquid
mix	mixture
o	outside
p	pipework

r	refrigerant or radiator
s	steel or surface
sw	swept volume
v	refrigerant vapour or volumetric
w	water or tube wall
1-4	thermodynamic state points in the vapour compression cycle, see figure 2.3
1	evaporator outlet/compressor suction
2	compressor discharge/condenser inlet
3	condenser outlet/expansion valve inlet
4	expansion valve outlet/evaporator inlet
5	evaporator inlet when high pressure drop distributor tubes connect the expansion valve to the evaporator . State point 5 is used for temperature and pressure only.

Superscripts

$\cdot$	first derivative with respect to time, e.g. $\dot{q}$ = rate of heat transfer
'	per unit length, e.g. A' = surface area per unit length of tubing, expressed in m^2/m, or saturation properties
"	per unit area, e.g. $\dot{m}''$ = mass flow rate per unit area, or mass flux.

CHAPTER 1 INTRODUCTION

1. ORGANISATIONAL BACKGROUND

This thesis records the research undertaken on behalf of GEC Research Ltd. through the Interdisciplinary Higher Degrees Scheme at the University of Aston in Birmingham. In common with many IHD projects, the work was carried out at the company's premises, in this case at the Energy Systems Group of the Engineering Research Centre (ERC) at Whetstone, Leicester. The Energy Systems Group was formed to accelerate the development and exploitation of new energy technologies, and has expertise in the areas of renewable energy systems, combined heat and power and turbomachinery design.

The initial six months of the research were spent surveying GEC companies in the domestic products sector in order to identify a useful and suitable project along the lines of the development of a domestic energy saving product. A project was conceived and gained the immediate support of Redring Electric Ltd. of Peterborough and the later support of GEC Xpelair Ltd. of Birmingham. The objective was to develop an electrically driven, vapour compression heat pump for domestic space heating. The principal UK market was seen as the replacement of oil fired central heating boilers. At the time this work began, in early 1982, the ERC (then the Mechanical Engineering Laboratory of GEC Power Engineering Ltd.) was completing the development of a product aimed at the same market sector, a water cooled, off-peak electric storage boiler (1). This product, the 'Nightstor 100', has subsequently achieved success in the market place.

2. PRODUCT SPECIFICATION

Discussions with the product companies led to the following specification:

a) Heat source to be the outside air. Compared with other possible environmental sources, e.g. subsoil, ground or surface water, air has the two disadvantages of the coincidence of low temperature with high building heating requirement and it contains water vapour which can lead to frost blockage of the evaporator. However, air has the overriding advantage of widespread availabilty and was adopted to ensure the largest possible market for the product.

b) Heat sink to be a conventional water filled radiator system, again for commercial reasons. In 1982 some 71 per cent of all central heating systems in the UK were the boiler/radiator type (2), and practically all oil fired systems used water as the heat distribution medium. Thus, to be compatible with current UK heating practice, the heat pump must be water cooled.

c) Nominal heat output capacity of 6 kW. It was recognised that the market would require a range of machines and that the 6 kW version would represent the smallest capacity within the range. It was decided to develop the 6 kW unit because this corresponds to the output available from a heat pump using the largest available compressor operating on a single phase electrical supply. The larger heat pumps would require either multiple combinations of single phase compressors or larger, three phase machines.

d) Electrical supplementary heater. Although the heat pump could be operated in parallel with the existing oil fired boiler, and the boiler used to supplement the heat output at sub-balance point ambient temperatures, it was decided to incorporate an electrical resistance heater in the heat pump package to provide a stand-alone heating source. It was anticipated that in the majority of installations the oil boiler would be near the end of its working life and so would be removed at the time of the installation of the heat pump. The capacity of the supplementary heater was to be the same as the nominal heat output of the heat pump, and located immediately downstream of the condenser.

e) Space heating only. Owing to the considerations of possible capacity shortfall with the 6 kW unit and of the higher output temperatures associated with the production of domestic hot water, the heat pump is primarily aimed at providing only space heating. The heat pump could be marketed as part of a package which included the installation of a 50 gallon off-peak electric hot water storage cylinder plus any economically feasible enhancement of the thermal insulation of the house, as is the current practice with the Nightstor 100 product. The heat pump is not required to produce summer air conditioning.

f) Self-contained, unitary design for outside siting. To keep installation costs down all the major components are to be contained in a single, transportable, weather-proofed construction. The refrigerant circuit to be charged and commissioned during manufacture. Site work to be restricted to the provision of suitable hardstanding and condensate drain, connection and thermal insulation of the water flow and return piping, connection of the electrical supply and the installation of the indoor user control box. The electrical supplementary heater to be included in the heat pump compartment. The central heating circulating pump to remain installed at a convenient site within the building, in accordance with current practice.

g) Wherever possible, select standard, commercially available components. This applies particularly to the compressor and control valves. It was appreciated, however, that the major heat exchangers may be non-standard.

h) Hot gas bypass defrost system, in which the high temperature vapour discharged from the compressor is fed directly into the frosted evaporator, is preferred to reverse cycle defrosting, because of the losses and unreliability associated with the four way reversing valve (3).

3. THE MAJOR EMPHASES OF THE RESEARCH PROJECT

At the time this work began there was a widespread interest in the UK in the application of the heat pump to domestic space heating, mainly because of an expectation of a continuing rise in the price of heating oil. A small number of indigenious manufacturers were developing and selling their heat pumps, although most of the units sold were imported from France, Germany, USA and Japan.

An effort was made to establish the deficiencies in the existing designs and the Electricity Council were helpful in this respect. The major problem areas were summarized as follows:

a) Evaporator frosting difficulties; rapid frost accumulation rate, reduction in COP_h under frosting conditions. This is a particular problem for the UK due to the generally high ambient humidity combined with the fairly mild air temperatures, and it was thought that the problem was made worse by the tendency to use small evaporators.

b) Unreliability. Split systems, for which refrigeration work was required on site to link the evaporator to the condensing unit, were seen as being particularly prone to malfunction caused by inadequate refrigerant charge, refrigerant leakage and the superheat calibration of the expansion valve. The minor electrical controls, for example pressure trips and the control of the supplementary heater, were also regarded as unreliable.

c) Installation practices. The criteria for matching the heat pump to the building heating load and the selection of radiator sizes were not established.

d) High starting current. The very high starting currents of the single phase compressor motors could lead to an objectionable voltage drop in the supplies of the owner and of his neighbours.

e) Noise levels, particularly associated with the movement of air through the evaporator.

f) Inadequate condenser water flow rate, due to the high pressure drop in the condenser.

g) High installation cost.

h) Low achieved COP_h in actual installations. Many of the heat pumps sold were not designed specifically for the UK climatic conditions.

A product development exercise was designed to address the important deficiencies in the existing heat pumps. The role of the collaborating product companies was to address the production engineering aspects of the design, with a view to minimizing the manufacturing cost, and to investigate the commercial background to the product, for example market size, distribution outlets etc. The role of the ERC was to design the vapour compression unit and select the ancillary components to bring the development to the stage of a demonstration of in-service performance of a prototype heat pump.

Within the scope of the product design and development exercise a number of specific areas of research were identified:

a) Accurate matching of the capacities of the major components to optimize the heating performance of the heat pump.

b) Evaporator design with reference to heat pump COP_h, frost deposition and noise levels.

c) Study of the effects of the characteristics of the building and the heat distribution system on the seasonal performance of the heat pump.

d) The development of a control algorithm to minimize running costs.

Item a) was approached as a combined analytical and experimental study in which the behaviour of the major components was numerically modelled and the results of the models were compared with data collected from tests. Once the analytical methods were validated in this way the model was used to indicate the optimum component selection. Item b), the evaporator design, was approached in a similar fashion, although the investigation into the behaviour of the evaporator under frosting conditions was primarily an experimental investigation. The study into the effects of the building and heat distribution system, item c), was mainly analytical, although the results were validated against the experimental data generated by the field trial of a prototype heat pump. The field trial also provided the means of developing and investigating the control strategy, item d). During the trial all logical control over the heat pump was effected by a micro computer, providing an expedient means of making corrections and adjustments to the control algorithm.

Within the company context, the author's role was as the 'Engineer Responsible' for the original definition of the project, the development and clarification of the specification, the design, procurement, assembly and operation of all experimental apparatus and the prototype unit, and for the preparation and execution of the computer programs. Project management, encompassing administration, financial control and technical guidance was provided by a senior engineer, the 'Project Officer'. Additional assistance was provided by the Company in the form of drawing office and laboratory technician support.

4. ORGANISATION OF THE THESIS

Besides meeting academic requirements this thesis is designed also to record the results and conclusions from a series of experimental and analytical studies of the design and application of the domestic heat pump, for the use of the Company in pursuing the product development exercise, and for other workers in this field.

The thesis is organised as two introductory chapters, followed by the five core chapters, 3 to 7, which form the technical contribution of this study, and draws to a close with a discussion of the commercial prospects of the product and the presentation of the main conclusions. The first two chapters set out the nature and scope of the investigation, survey the historical development of the domestic heat pump and consider the thermodynamic principles of operation of the vapour compression heat pump. Chapter 3 contains the methods and results from two mathematical models; an analysis of the vapour compression cycle which is used to optimize the component design, and a numerical simulation study to predict the behaviour of a heat pump as the heating source of a building central heating system. The design, analysis and experimental evaluation of the evaporator is presented in Chapter 4. The methods of selecting the other major components are given in Chapter 5. Details of the design of the prototype heat pump and its control system are presented in Chapter 6, along with an account of the field trial installation and the data collection and analysis methods. The results and findings from the field trial are presented and discussed in Chapter 7. Chapter 8 is a discussion of the market prospects of the domestic heat pump. The main conclusions are presented in Chapter 9 and recommendations for further work are given in Chapter 10.

A short Bibliography of general texts in this field is included for the benefit of non-specialists requiring a more detailed introduction to the various topics.

5. THE HISTORICAL DEVELOPMENT OF THE HEAT PUMP

Lord Kelvin is generally acclaimed as the originator of the concept of the heat pump as a heating device, and produced a design in 1852 (4), although it is known that the invention of mechanical refrigeration predates this (5, 6). Lord Kelvin's heat pump was not constructed, and it was left to Haldane to demonstrate the first practical example, some 75 years later. This was a dual heat source device, extracting heat from either the surrounding air or a nearby river and delivering it to his own house in Scotland (7). This aroused considerable interest, particularly in Switzerland and the USA, and by 1940 some 50 heat pumps were in operation in the USA (5). Interest in Switzerland was related to the lack of indigenous fossil fuel reserves and the preponderance of hydro-electricity; between 1939 and 1945 35 large scale heat pumps were installed, mainly to heat municipal buildings, universities and swimming pools (5).

The first UK large scale installation was the Norwich heat pump, designed and built by Sumner in 1945 (8). This was a water to water vapour compression machine of 29 kW output, using sulphur dioxide as the refrigerant and river water as the heat source. A second major scheme was commissioned in 1952 to heat and cool the Festival Hall, London by extracting heat from the River Thames (9). A centrifugal compressor was driven by a modified Rolls Royce Merlin engine fuelled by town gas, and this powerful drive resulted in a heat output of 2.7 MW. Unfortunately the building heating requirement was overestimated and the heat pump was not well matched to the load, leading to operating difficulties.

An early example of a UK domestic heat pump is Sumner's ground source device applied to the heating of his own bungalow in 1950 (10). Some years later the Lucas Company developed an air to air domestic heat pump and launched the product during the 1960's (11). The relatively low fuel costs of that period, however, militated against the commercial success of the device and the product was discontinued.

Post war development of reversible heat pump/air conditioners in the USA followed an interesting growth curve. The 1950's showed a rapid growth from 2000 installed units in 1954 to 10000 in 1957, and by 1963 some 76000 units were in operation (5). The design emphasis of these machines is the air conditioning capability. Stagnation of sales followed due to reliability problems, and expansion restarted after 1972. In 1976 it is estimated that 1.6 million units were in operation in the USA (3).

Residential heat pumps have been most successful in those regions where both heating and cooling are required, e.g. Australia, Japan and the USA (12). Where the cooling loads are negligible the heat pump must compete against existing, primarily fossil fuel fired heating systems, and this is broadly the case for Western Europe in general and the UK in particular. A resurgence of interest in Europe occurred during the 1970's as a consequence of the 1973/74 oil crisis. Major research initiatives were supported by the EEC Energy R&D Programme (13) and a number of manufacturers developed products to serve the residential sector. Within the UK market in 1984 BSIRA were predicting the overall heat pump market to nearly double in five years, from the estimated total sales of £24.5 M in 1983 (14).

CHAPTER 2 THERMODYNAMIC CONSIDERATIONS

This chapter lays downs the fundamental thermodynamic principles of heat pump operation. The ideal cycle, the reversed Carnot cycle is described, followed by a discussion on how the practical version of this cycle, the vapour compression cycle, differs from the ideal case. The Second Law of Thermodynamics is invoked to demonstrate the irreversibilites which occur in actual systems, the analysis being based on measured data from a laboratory test rig.

1. THE IDEAL HEAT PUMP CYCLE

A heat pump is a device which transfers heat from a low temperature reservoir to a higher temperature reservoir by the application of mechanical work, figure 2.1 a). The First Law of Thermodynamics requires that, in the absence of heat losses, the heat transferred to the higher temperature reservoir is the sum of the heat extracted from the low temperature source and the work absorbed by the device.

The consequences of the Second Law of Thermodynamics are that the work absorbed must be greater than zero and that the ratio of the heat rejected to the work absorbed cannot exceed a certain value for any combination of heat source and heat sink temperatures. This ratio is termed the coefficient of performance of the device when operating as a heat pump, COP_h. The limiting value is given by (see for example, Gosney (15)):

$$COP_{h,c} = \frac{T_c}{T_c - T_e}$$

The subscript c denotes the maximum possible value of the COP_h, which would be achieved by executing the hypothetical cycle of ideal operations proposed by Carnot. The absolute temperatures must be used in the above equation. The reversed Carnot cycle is described with reference to figure 2.1 b):

a) Reversible, adiabatic compression from state 1 until the vapour temperature reaches T_c.

b) Reversible, isothermal compression until the pressure of the vapour reaches the saturated pressure corresponding to T_c.

c) Reversible, isothermal condensation at constant pressure.

d) Reversible, adiabatic expansion to the saturated pressure corresponding to T_e, in which work is done by the fluid.

e) Reversible, isothermal evaporation at constant pressure, after which the fluid is restored to its original state.

The reversed Carnot cycle is the sequence of the ideal processes against which the performance of a practical device may be measured.

2. THE VAPOUR COMPRESSION CYCLE

The most widespread thermodynamic cycle for refrigeration, air conditioning and heat pump duties is the vapour compression cycle. This cycle, in its ideal form, deviates from the reversed Carnot cycle in two respects, see figure 2.1 c):

a) Compression proceeds directly to P_c, which involves an increase in the work of compression and this is reflected in the superheating of the vapour to state 2.

b) The isentropic expansion process 3 to 4 is replaced by an irreversible, isenthalpic throttling process which represents a loss of the work available from the expansion. This work is sacrificed to simplify the cycle.

The practical cycle is shown in figure 2.1 d) and the major components in figure 2.2. The practical cycle has the following attributes:

a) The suction vapour is slightly superheated, 1' to 1, to ensure the fluid entering the compressor is completely dry. Wet vapour may lead to mechanical damage to the compressor.

b) The compression is neither reversible nor adiabatic. The heat removed from the fluid during compression is, however, usually small compared to the work of compression.

c) Frictional pressure drop results in a gradual reduction in the saturated condensing and evaporating temperatures as the fluid flows through the heat exchangers.

d) The cycle may include some liquid subcooling, 3' to 3, which increases the heat transferred to the heat sink for the same work of compression.

Finally, unlike the reversed Carnot cycle, temperature differences must be established across the heat exchangers to effect the transfer of heat. Thus the condensing temperature must exceed the heat sink temperature, and the evaporating temperature must be less than the heat source temperature and this leads to a reduction in COP_h.

3. THERMODYNAMIC ANALYSIS OF THE VAPOUR COMPRESSION CYCLE

The purpose of the analysis is to highlight the irreversibilities both of the heat pump system as a whole and within individual components of the system. It follows the procedures given by Trip for refrigeration systems, and more recently, by McMullan and Morgan for heat pumps (16, 6). In order to illustrate the performance of actual heat pump components, the analysis is presented as a case study based on experimental results from the Evaporator Test Rig, which is a complete air to water heat pump. The details of the experiment are given in Chapter 4. The particular test results are chosen to represent the typical operating conditions of a domestic heat pump; the air temperature is 3.2°C and the condenser water inlet temperature is 50°C. The thermodynamic state point data are presented in table 2.1, and the cycle is plotted on the T-s plane in figure 2.3 a) and on the log P-h plane in figure 2.3 b). A straight, dashed line is drawn for the compression process because the details of the pathway are not known. The expansion is shown dashed to emphasize the irreversible nature of this process.

Thermodynamic irreversibility is defined as the amount of available energy which is rendered unavailable during a process. The available energy in a system is that energy which is available for the production of mechanical work. If T_a is the temperature of the coldest, naturally available heat sink, then the maximum efficiency with which a quantity of heat Q at temperature T could be converted into work is $(T - T_a)/T$. Thus the available and unavailable components of Q are given by:

$$\text{available energy} = \frac{Q\,(T - T_a)}{T}$$

$$\text{unavailable energy} = \frac{Q\,T_a}{T}$$

For an isolated system:

$$\Delta S_e = -\frac{Q_e}{T_a}, \qquad \Delta S_c = \frac{Q_c}{T_w}$$

The sign convention for heat and work transfers adopted here is contrary to normal thermodynamic practice, in that heat transferred <u>from</u> the system and work done <u>on</u> the system are positive. This is convenient as it results in the two energy transfers of prime importance, Q_c and W, being positive.

The total irreversibility in the system is given by:

$$I = T_a\,(\Delta S_e + \Delta S_c)$$

where T_a is the temperature of the heat source, assumed to be also the coldest naturally occurring heat sink, Q_e is extracted from this source, and Q_c is delivered to the water coolant at temperature T_w.

The heat and work transfers across the cycle, all expressed per kg of refrigerant flowing through the circuit, are given as:

$$Q_e = h_1 - h_4 = 140.71 \text{ kJ/kg}$$
$$Q_c = h_2 - h_3 = 214.21 \text{ kJ/kg}$$
$$W = Q_c - Q_e = 73.50 \text{ kJ/kg}$$
$$COP_h = \frac{Q_c}{W} = 2.91$$

In fact the actual COP_h is lower than this, since it is based on the electrical power absorbed by the fan and the compressor, the real compressor is not adiabatic and some heat is lost to the surroundings.

The heat source temperature is 276.2 K and the water coolant temperature is 323 K, hence the $COP_{h,c}$ and work input for a reversed Carnot cycle heat pump are

$$COP_{h,c} = \frac{T_w}{T_w - T_a} = 6.90$$

$$W_{ideal} = \frac{Q_c}{COP_{h,c}} = 31.04 \text{ kJ/kg}$$

The additional work input to drive the actual cycle of 42.46 kJ/kg represents the total irreversibility in the system, which may be independently calculated from (16):

$$I_{total} = T_a \left\{ \frac{Q_c}{T_w} - \frac{Q_e}{T_a} \right\} = 42.46 \text{ kJ/kg}$$

To investigate the irreversibilites of the individual components, it is necessary to add the irreversibilites of the working fluid which result from the operation of the component to those of the surroundings:

$$I_{ref} = T_a (s_{in} - s_{out})$$

$$I_{surroundings} = T_a \frac{(\text{heat transferred})}{T}$$

where T is the temperature at which the heat is transferred.

For convenience, it is assumed that the compressor and expansion valve are adiabatic.

Compressor : $I = T_a (s_2 - s_1)$ $= 21.38$ kJ/kg

Condenser : $I = T_a \left\{ s_3 - s_2 + \frac{Q_c}{T_w} \right\}$ $= 8.14$ kJ/kg

Expansion valve : $I = T_a (s_4 - s_3)$ $= 8.20$ kJ/kg

Evaporator : $I = T_a \left\{ s_1 - s_4 - \frac{Q_e}{T_a} \right\}$ $= 4.74$ kJ/kg

TOTAL $I = 42.46$ kJ/kg

The irreversibilites occurring in the minor components and in the connecting pipework are effectively lumped together with the major components in the above analysis. Interestingly, the compressor has the greatest irreversibility, and the departure from isentropic compression is indicated on the T-s diagram of figure 2.3 a). The next chapter examines the performance consequences of improving the isentropic efficiency of the compressor. The condenser shows a higher irreversibility than the evaporator because of the high temperature difference which occurs in the desuperheating region, whereas the evaporator is practically isothermal. In their analysis of a vapour compression system using R12 and an open, rotary vane compressor, McMullan and Morgan found that the greatest irreversibility loss was attributable to the expansion process (6).

4. CONCLUDING REMARKS

In conclusion, this chapter has described the vapour compression cycle and compared it to the ideal, the reversed Carnot cycle. An assessment of the thermodynamic irreversibilities in the measured operating cycle of an experimental heat pump has shown the greatest losses occur in the compressor.

State Point	Pressure P (bar, absolute)	Temperature T (°C)	Specific Volume v (m^3/kg)	Specific Enthalpy h (kJ/kg)	Specific Entropy s (kJ/kgK)	Quality x
1	4.22	1.4	0.057	252.56	0.9507	1.02
2	22.51	122.6	0.015	326.06	1.0281	1.44
3	21.95	52.4	0.0009	111.85	0.3944	-0.02
4	4.54	-2.8	0.0081	111.85	0.4241	0.34
Saturated Properties						
1'	4.22	-4.9	0.055	248.11	0.9343	1.00
2'	22.51	56.6	0.0098	262.73	0.8524	1.00
3'	21.95	55.4	0.0009	115.50	0.4060	0.00
4'	4.54	-2.8	0.0008	41.33	0.1632	0.00

Air temperature = 3.2°C
Water inlet temperature = 50.0°C

TABLE 2.1: THERMODYNAMIC CYCLE FOR EVAPORATOR TEST RIG
RUN C-30-7
SOURCE: Chapter 4

CHAPTER 3 SYSTEM MODELLING

1. INTRODUCTION AND OBJECTIVES

This chapter reports the details of, and important results from mathematical modelling of the performance of a heat pump carried out at two levels. The first is a detailed examination of the behaviour of the major components of the heat pump. The objective is to predict the effect on the steady-state performance of the heat pump of changes to the design of the components and to variations in operating conditions. The solution to the equations governing the component behaviour is found by the iterative computer program VCCM, standing for 'Vapour Compressor Cycle Model'. This program is written in the style of a design tool, to enable the heat pump designer to perform numerical explorations on his proposed design. With this goal in mind, it is the ability of the model to accurately predict the trends in performance, rather than absolute values, which is important. This model is described in section 2 and some pertinent results are given in section 3. The model of the evaporator, which is a major element of the cycle model, is described in Chapter 4.

The second level of modelling is concerned with the prediction of the behaviour of the heat pump when heating a house. The aim is to identify and quantify those factors which have a significant bearing on the achieved performance, for example the radiator area, building heating requirement, and the control strategy. This model takes the form of a computer simulation of the dynamic interactions between the building, the heat distribution system and the heat pump. The model is described in section 4 and results are given in section 5.

2. VAPOUR COMPRESSION CYCLE MODELLING

2.1 Introduction

The motivation to develop a mathematical model of the vapour compression cycle is to facilitate the design of the heat pump system. The model should be able to predict the performance of a given heat pump design from the inputs of the temperature and flow rate of the ambient air and cooling water streams. If the model is also able to investigate evaporator dehumidification effects, then the ambient relative humidity must be supplied as an input. To produce a convenient design tool, then ease of variation in the design of the major heat pump components is required.

Several computer based mathematical models of the vapour compression cycle reported in the literature have influenced this present work. These include the steady-state heat pump models by McMullan and Morgan (17), by Freeman et al (18) and by Tassou et al (19), as well as the dynamic models of the vapour compression system by James and Marshall (20), by Chi and Didion (21) and by MacArthur (22). A consensus from these approaches is to characterise the compressor volumetric efficiency as a function of compression ratio, although the nature of the function used varies. A number of approaches are adopted to describe the thermal performance of the compressor, for example the use of the index of compression (19, 20), or the use of the compressor isentropic efficiency characteristics (17, 18). Most of the research groups preferred to determine the numerical coefficients by testing a compressor. The treatment of the heat exchanger modelling differs widely in detail, although the general approach of handling separately the two phase and single phase regions in each heat exchanger finds widespread acceptance.

2.2 Major Assumptions and Simplifications

a) The frictional pressure drop of the working fluid in the condenser and all interconnecting pipework is usually small and so its effects are not considered.

b) Heat transfer from and to the working fluid through the interconnecting pipework is ignored.

c) The working fluid is assumed to be a pure refrigerant, that is the effects of entrained oil on the properties of the fluid are not considered. This is justified on the grounds of the welcome simplification of the analysis, however, there is some experimental evidence to show that the presence of the oil does modify the behaviour of the working fluid, see Chapter 5.

d) Steady-state conditions.

e) Air side dehumidification and frost fouling effects are not considered.

2.3 Compressor

The function of the compressor model is to predict the mass throughput of the refrigerant, the electrical power absorbed by the compressor motor, and the discharge condition of the working fluid, given the suction conditions and the pressure ratio. The approach is to describe the behaviour of a hermetically sealed reciprocating compressor/motor combination by two performance indicators, the volumetric and isentropic efficiencies, both of which can be expressed as simple functions of the pressure ratio.

2.3.1 Volumetric Efficiency

The dependence of the volumetric efficiency of a given reciprocating compressor on pressure ratio can be expressed as:

$$\eta_v = a - b\left(\frac{P_2}{P_1}\right) \qquad \text{where} \quad \eta_v \equiv \frac{\dot{m}_r v_1}{\dot{V}_{sw}}$$

The volumetric efficiency is defined relative to the condition of the fluid immediately upstream of the point of entry to the hermetic case. The constants a and b are found by experiment or by manipulation of the manufacturer's performance data.

These constants clearly depend on the type of compressor; hermetic, semi-hermetic or open, and on the design details of a particular model within a given type. Less obviously, they are also dependent on the dryness of the suction vapour. A rapid fall in volumetric efficiency when wet vapour is compressed is reported by several investigators, see for example (23) and (24). Furthermore, Gosney reports results for dry compression of R12, indicating increasing volumetric efficiency with suction superheat, although the rate of change of η_v is much less for dry compression compared with the rapid change caused by wet compression

(25). Suggested reasons for the dependence of volumetric efficiency on the dryness of the vapour include the effects of entrained liquid droplets within the superheated vapour stream and the effects of condensation taking place on the cylinder walls during compression (15). Liquid present in the clearance volume at the end of the compression stroke may continue to evaporate during the expansion stroke and the next suction stroke, thereby reducing the volume of induced suction vapour. Thus for modelling purposes a and b must be determined at the appropriate dryness; for the heat pump design this is a vapour superheated by 2 to 10 deg C.

For the hermetic compressor chosen for the prototype heat pump the following values were found by experiment, see Chapter 5:

$a = 0.8975 \pm 2\%$
$b = 0.05625 \pm 2\%$

2.3.2 Isentropic Efficiency

For the purpose of modelling a hermetic compressor for heat pump duty, the isentropic efficiency of the compression process may be defined as the ratio of the specific enthalpy change for isentropic, adiabatic compression from state points 1 to 2s, to the specific electrical power consumed by the compressor motor:

$$\eta_{is} \equiv \frac{\dot{m}_r \ (h_{2s} - h_1)}{\dot{W}_c}$$

This definition corresponds to the 'externally measured isentropic efficiency' used in Chapter 5. The equation directly relates the compressor motor power consumption to the refrigerant mass flow rate and the condition of the working fluid in the suction line. This convenient definition in fact encompasses three separate efficiencies; the motor electrical efficiency, the mechanical efficiency of the compressor and what may be termed the true isentropic efficiency of compression. In practice it is difficult to separate these components, particularly for an hermetic compressor, whereas the product of all three may be found from direct measurement of the suction state, the discharge pressure and the motor power consumption, or may be estimated from performance data supplied by the compressor manufacturer.

The isentropic efficiency is modelled as a function of pressure ratio:

$$\eta_{is} = c - d\left(\frac{P_2}{P_1}\right)$$

where the numerical constants c and d are found by experiment or from manufacturer's data.

For the hermetic compressor chosen for the prototype heat pump the following values were found by experiment, see Chapter 5:

$c = 0.540 \pm 2\%$
$d = 0.015 \pm 3\%$

2.3.3 Energy Balance

The steady flow energy equation for the compressor simplifies to:

$$\dot{W}_c - \dot{q}_L = \dot{m}_r (h_2 - h_1)$$

The enthalpy at discharge, h_2, can be determined if q_L, the rate of heat loss from the compressor casing, is known. This will depend on the surface temperature of the casing, the thickness and conductivity of the thermal insulation applied to the compressor, the temperature of the surrounding air and the convective environment. This quantity was estimated for the compressor used in the prototype heat pump to be 150 W for representative operating conditions, and for convenience this figure is regarded as constant.

2.3.4 Summary of Compressor Model

Known : Performance parameters and compressor constants
$a, b, c, d, \dot{V}_{sw}, \dot{q}_L$

Given : (P_1, T_1) and P_2

To find : $\dot{m}_r$, $\dot{W}_c$ and h_2

Step 1 Calculate the mass flow rate:

$$\dot{m}_r = \frac{\dot{V}_{sw}}{v_1} \left\{ a - b \left(\frac{P_2}{P_1}\right) \right\}$$

Step 2 Calculate the motor power:

$$\dot{W}_c = \frac{\dot{m}_r \ (h_{2s} - h_1)}{c - d \left(\frac{P_2}{P_1}\right)}$$

Step 3 Calculate the discharge specific enthalpy:

$$h_2 = h_1 + \frac{(\dot{W}_c - \dot{q}_L)}{\dot{m}_r}$$

2.4 Condenser

The condenser model predicts the rate of heat transfer and temperature difference across the condenser and the water side pressure drop. The coaxial tube arrangement is modelled due to its suitability for domestic heat pump systems. Details of the geometrical arrangement of the coaxial tube condenser are given in Chapter 5.

2.4.1 Water Side Pressure Drop

The water side frictional pressure drop is found from Darcy's well-known formula (26):

$$\frac{\Delta P_w}{\rho_w g} = \frac{4fl}{d_1} \times \frac{u^2}{2g}$$

i.e. $\Delta P_w = \dfrac{32\ f\ l\ \rho_w \dot{V}_w^2}{\pi^2\ d_1^5}$

where the friction factor f depends on the Reynolds number of the flow and on the surface roughness of the tubing. The various types of finning applied to the outer, refrigerant side, of the condenser tubes may give rise to surface features on the bore of the tube which affect the surface roughness. Thus f will vary depending on the condenser design. This is handled by defining a constant, the relative roughness, which expresses the ratio of the actual friction factor to that for a plain, smooth tube, as calculated from Blasius's formula (26):

$$R_f \equiv \frac{f}{0.079\ Re^{-0.25}}$$

R_f can be found by experiment or by manipulation of the manufacturer's data. The tests performed on two candidate coaxial condensers, Wieland KWG 3X and IMI/YIA CK8-20, resulted in R_f values of 2.128 $\pm 5\%$ and 1.021 $\pm 5\%$ respectively, see Chapter 5.

2.4.2 Condenser Heat Transfer Model - Simplifying Assumptions

a) No liquid subcooling at the outlet of the condenser. By the very nature of the coaxial tube condenser it is difficult to achieve significant subcooling as the liquid-vapour interface extends throughout the shell side volume. The assumption of no appreciable liquid subcooling is supported by the experimental evidence given in Chapter 5.

b) Negligible heat loss from the condenser to the surroundings, i.e. the heat transferred from the working fluid is fully recovered by the water side.

2.4.3 Water Side Heat Transfer Coefficient

The Dittus-Boelter correlation is used to determine the water side heat transfer coefficient (27):

$$Nu_w = 0.023\ Re_w^{0.8}\ Pr_w^{0.4}$$

2.4.4 Refrigerant Side Heat Transfer

For the purpose of modelling, the condenser is considered to be two heat exchangers in series; a desuperheater in which the refrigerant vapour is cooled to the saturation temperature, and a true condenser. The water flows in counter flow first through the condenser and then through the desuperheater. The temperature profiles of the refrigerant and water are shown in figure 5.12. The water temperature at the outlet of the true condenser reaches an intermediate value of T_{wi} and reaches the final value of T_{w2} at the outlet of the desuperheater.

The heat transfer rates across the two heat exchangers are found from the enthalpy changes of the refrigerant:

desuperheater $\dot{q}_{c,d} = \dot{m}_r\ (h_2 - h_2')$

true condenser $\dot{q}_{c,c} = \dot{m}_r\ (h_2' - h_3)$

Hence the intermediate and final water temperatures are calculated from:

intermediate $$T_{wi} = T_{w1} + \frac{\dot{q}_{c,c}}{\rho_w \dot{V}_w Cp_w}$$

outlet $$T_{w2} = T_{wi} + \frac{\dot{q}_{c,d}}{\rho_w \dot{V}_w Cp_w}$$

Consider first the heat transfer in the desuperheater. The heat transfer coefficient for the single phase vapour is found from the Dittus-Boelter equation:

$$Nu_v = 0.023\ Re_v^{0.8}\ Pr_v^{0.4}$$

where the properties of the fluid are evaluated at the mean vapour temperature of $(T_2 + T_c)/2$. The overall heat transfer coefficient for the desuperheater, referred to the water side area, is found from:

$$\frac{1}{U_{d,i}\ A_i} = \frac{1}{h_i\ A_i} + \frac{1}{h_f\ A_i} + \frac{t_w}{k_w\ A_m} \quad \frac{1}{h_{d,o}\ (A_p + \eta_f\ A_s)}$$

where h_f is the water side fouling coefficient

t_w is the mean thickness of the tube wall

A_m is the mean area of the tube wall

A_p is the heat transfer area of the outside of the central tube, not including the area of the fins

A_s is the area of the fins

η_f is the fin efficiency

The (t_w/k_wA_m) term is considerably smaller than the other conductances and so is ignored. Also for a new condenser the water side fouling was assumed to be negligible. For short, integral type fins the fin efficiency was taken to be 1.0.

The log mean temperature difference is calculated from:

$$\Delta T_{LM,d} = \frac{(T_2 - T_{w2}) - (T_c - T_{wi})}{\ln \left| \dfrac{T_2 - T_{w2}}{T_c - T_{wi}} \right|}$$

Thus the area required for desuperheating may be calculated from:

$$A_{i,d} = \frac{\dot{q}_{c,d}}{U_{d,i}\ \Delta T_{LM,d}}$$

Now since the combined area of the desuperheater and true condenser is fixed (A_i), the area remaining for condensation can be found from:

$$A_{i,c} = A_i - A_{i,d}$$

Consider now the heat transfer in the true condenser. The refrigerant side heat transfer coefficient is found from the Nusselt equation for film type condensation outside a single, horizontal, finned tube (28):

$$\frac{h_{o,c}\ d_e}{k_l} = C_1 \left\{ \frac{d_e^3\ \rho_l\ (\rho_l - \rho_v)\ g\ (h_2' - h_3)}{k_l\ \mu_l\ (T_c - T_s)} \right\}^{0.25}$$

where C_1 = 0.725 by analysis.

In the application of this equation to fluorocarbon refrigerants, the density of the vapour phase, ρ_v, is usually ignored (29, 30) and a revised value of $C_1 = 0.689$ is supported by ASHRAE (30).

The effective diameter of the finned tubing is calculated from (30, 31):

$$\frac{1}{(d_e)^{0.25}} = 1.30 \frac{A_s \quad \eta_f}{A_{ef} \ (L_{mf})^{0.25}} + \frac{A_p}{A_{ef} \ d_2^{0.25}}$$

where $A_{ef} = \eta_f A_s + A_p$

L_{mf} = mean length of fin = (area of one side of fin)/(outside diameter of fin).

The details of the fin configuration for the IMI/YIA CK8-20 condenser are shown in figure 6.4, for which an effective diameter of 2.40 mm is calculated. Unfortunately, not all the condenser manufacturers supplied information on the fin design. Hence the assumption that the ratio of the effective diameter to the actual root diameter was the same as that for the CK8-20 was applied when more precise information was not available, e.g. for the KWG 3X condenser.

The overall heat transfer coefficient for the true condenser is found from:

$$\frac{1}{U_{c,i} A_i} = \frac{1}{h_i A_i} + \frac{1}{h_{c,o} (A_p + \eta_f A_s)}$$

The tube wall conductance and the water side fouling factor terms are omitted for the reasons given in the development of the desuperheater equations. Since the available area for condensation is known, along with the heat transfer rate and the overall heat transfer coefficient, the Number of Transfer Units/Effectiveness method is applied to calculate an improved value of the condensing temperature (32):

$$T_c = T_{w1} + \frac{(T_{wi} - T_{w1})}{E}$$

where $E = 1 - e^{-NTU}$, the effectiveness of a true condenser (23)

and $$NTU = \frac{U_{c,i} A_{i,c}}{\rho_w \dot{V}_w Cp_w}$$

2.4.5 Summary of the Condenser Model

Known : Condenser geometrical details; d_1, d_2, d_3, d_e, l, A_p, A_s, A_i, η_f.
Water side flow and fouling allowances; R_f, h_f

Given : Inlet conditions T_{w1}, $\dot{V}_w$, $\dot{m}_r$, T_2 and an estimated value of T_c. The condensing pressure, P_2, is taken to be the saturated pressure at T_c.

To find: ΔP_w and an improved estimate for T_c. When the iteration has converged, find $\dot{q}_c$ and T_{w2}.

Step 1 Calculate the water side pressure drop and heat transfer coefficient:

$$\Delta P_w = \frac{2.528 \; R_f \; Re_w^{-0.25} \; l \; \rho_w \; \dot{V}_w^2}{\pi^2 \; d_1^5}$$

$$h_i = 0.023 \; \frac{k_w}{d_1} \; Re_w^{0.8} \; Pr_w^{0.4}$$

Step 2 Calculate the heat transfer rates in the desuperheating and condensing regions:

$$\dot{q}_{c,d} = \dot{m}_r \, (h_2 - h_2')$$
$$\dot{q}_{c,c} = \dot{m}_r \, (h_2' - h_3)$$

Step 3 Determine the intermediate and final water temperatures:

$$T_{wi} = T_{w1} + \frac{\dot{q}_{c,c}}{\rho_w \dot{V}_w Cp_w}$$

$$T_{w2} = T_{wi} + \frac{\dot{q}_{c,d}}{\rho_w V_w Cp_w}$$

Step 4 Calculate the area required for desuperheating:

$$A_{i,d} = \frac{\dot{q}_{c,d}}{U_{d,i} \quad \Delta T_{LM,d}}$$

$$\text{where } U_{i,d} = \left\{ \frac{1}{h_i} + \frac{1}{h_{d,o}} \times \frac{A_i}{(A_p + \eta_f A_s)} \right\}^{-1}$$

$$h_{d,o} = 0.023 \frac{k_v}{d_2} Re_v^{0.8} Pr_v^{0.4}$$

Step 5 Find the area remaining for condensing:

$$A_{i,i} = A_i - A_{i,d}$$

Step 6 Calculate the mean condenser tube temperature from the water temperature:

$$T_s = \frac{(T_{wi} + T_{w2})}{2} + \frac{\dot{q}_{c,c}}{h_i A_{i,c}}$$

Step 7 Calculate the overall heat transfer coefficient in the true condenser:

$$U_{i,c} = \left\{ \frac{1}{h_i} + \frac{1}{h_{o,c}} \times \frac{A_i}{(A_p + \eta_f A_s)} \right\}^{-1}$$

$$\text{where} \quad h_{o,c} = 0.689 \left\{ \frac{\rho_1^2 k_1^3 (h_2' - h_3) g}{\mu_1 (T_c - T_s) d_e} \right\}^{0.25}$$

Step 8 Find an improved estimate of T_c:

$$T_c = T_{w1} + \frac{(T_{wi} - T_{w1})}{E}$$

$$\text{where } E = 1 - e^{-NTU}$$

$$\text{and} \quad NTU = \frac{U_{i,c} \; A_{i,c}}{\rho_w \; V_w \; Cp_w}$$

The iteration between the condenser and compressor models continues until T_c converges. The convergence criterion is that successive estimates T_c differ by less than 0.05 deg C.

This model has been applied to the CK8-20 and KWG 3X condensers which were experimentally evaluated in the component selection study presented in Chapter 5. Table 3.1 summarizes the geometrical and fluid flow data for these condensers. Some items of data for the KWG 3X condenser relating to the fin design were not available and so, as previously described, these were estimated by assuming similar characteristics to the CK8-20 design.

2.5 Expansion Valve

The thermostatic expansion valve has two functions:

a) To reduce the pressure of the working fluid to a suitable level to sustain heat transfer across the evaporator.

b) To meter the refrigerant flow to produce a constant vapour superheat at the outlet of the evaporator.

The pressure is reduced by irreversible expansion through a sharp edge orifice. The mathematical modelling of this component is trivial and may be summarized as:

a) Isenthalpic expansion; $h_4 = h_3$. This is justified by the observations that no useful work is obtained as a result of the expansion and that the area available for heat transfer to or from the working fluid during the process is small.

b) A fixed outlet superheat, ΔT_{sh}, is imposed on the evaporator model.

2.6 Evaporator

The evaporator model is discussed in detail in Chapter 4.

2.7 Solution Procedure

The numerical solution method is an application of the successive substitution technique (33). Initial estimates of the evaporating and condensing temperatures are made. The condensing temperature is then refined by iteration between compressor and condenser models. When T_c has converged, T_e is refined by further iteration between the evaporator and the compressor/condenser models. The process continues until both T_e and T_c have converged. This methodology is illustrated in figure 3.1.

The complete vapour compression cycle model may be summarized as:

Known : Heat exchanger dimensions and compressor and expansion valve characteristics.

Given : External operating conditions, e.g. T_{a1}, $\dot{V}_a$, T_{w1}, $\dot{V}_w$.

To find: The performance of the system : $\dot{q}_c$, $\dot{W}_c$, $\dot{W}_f$, COP_h, T_{w2}, etc.

Step 1 Make initial estimates of the evaporating and condensing temperatures:

$$T_e = T_{a1} - 15$$
$$T_c = T_{w1} + 15$$

Step 2 Compressor model
Calculate mass flow rate, motor power and fluid discharge conditions.

Step 3 Condenser model
Calculate an improved estimate for T_c.
Return the new estimate of T_c to step 2, and iterate between steps 2 and 3 until T_c converges.

Step 4 Evaporator model

Calculate an improved estimate for T_e.

Return the new estimate of T_e to step 2 and continue steps 2, 3 and 4 until T_e converges.

Step 5 Collate results

Summarize the cycle heat and work transfer rates and equilibrium temperatures.

The vapour compression model is contained in the FORTRAN program VCCM, and is shown in flow chart form in figure 3.1. The convergence rate is usually quite rapid; the inner loop, compressor/condenser, requiring typically 5 cycles and the outer loop, compressor/condenser/evaporator, takes 8 cycles. The program is menu driven and designed to interact with the heat pump designer. Operating conditions and the design details of the heat pump may be changed from the keyboard, and the user may set up a loop in which a key variable, e.g. ambient air temperature or condenser length, may be successively varied between the user defined limits. This enables the user to explore the significance of component changes and determine the performance of the proposed system across a wide range of operating conditions.

Refrigerant thermodynamic and transport properties are computed by a suite of programs kindly provided by the New University of Ulster (34). The program suite is designed for refrigerants R11, R12, R21, R22, R502 and R114 and determines thermodynamic properties given pressure and a second (independent) property. Transport properties are calculated for saturated vapour and liquid. A full description and listing of this suite is given in McMullan and Morgan, 1981 (6).

The accuracy of these programs compared favourably with ICI's thermodynamic property tables for R22 across the range of conditions relevant to this study.

2.8 Validation

2.8.1 Compressor

The results of the experimental evaluation of the compressor, as presented in table 5.2, were used to assess the predictive accuracy of the compressor model. Table 3.2 shows the values of the test input conditions of T_1, P_1 and T_c, the results of the model, $\dot{m}_r$, $\dot{W}_c$, T_2, the measured values of these quantities together with an indication of the accuracy of the model. The accuracy of prediction of $\dot{m}_r$ is high, with a discrepancy of less than 1 per cent. The electrical power, $\dot{W}_c$, is predicted to $\pm$ 3 per cent, and the predicted value of T_2 tends to be an overestimate of 2 to 10 deg C.

2.8.2 Condenser

The accuracy of the condenser/compressor iteration loop was tested using data from experiments on two candidate condensers, CK8-20 and KWG 3X, see tables 5.7 and 5.6 respectively. The inputs to the model were the compressor suction state point, T_1, P_1, the water inlet temperature and flow rate. The outputs were the predicted water side pressure drop, water outlet temperature, condensing temperature and the rate of heat transfer to the water. The results of the validation exercise are shown in table 3.3. The prediction accuracy for the CK8-20 condenser is in general higher than that achieved for the KWG 3X unit, which is probably due to the greater knowledge of the geometry of the condensing surface of the CK8-20 design.

3. RESULTS FROM VAPOUR COMPRESSION CYCLE MODEL - TOWARDS AN OPTIMUM DESIGN

The computer model of the heat pump described above was used to determine the selection of the working fluid, examine the influence of the compressor characteristics on heat pump performance and to optimize the condenser size. The following data are the base line inputs to the model and apply throughout this section unless otherwise stated.

a) operating conditions

ambient air temperature,	T_a	4°C
water return temperature,	T_{w1}	49°C
air volumetric flow rate,	$\dot{V}_a$	1.25 m^3/s
water volumetric flow rate,	$\dot{V}_w$	1.0 m^3/h
evaporator exit superheat,	ΔT_{sh}	4 deg C

b) major heat pump components

compressor	Maneurop MT32 JF5
condenser	YIA CK8-20
evaporator	30m, 2 row, 2 circuit helical coil

c) working fluid R22

Details of the heat pump components are given in Chapter 6.

3.1 Performance Comparison Between the Commonly Available Fluorocarbon Refrigerants

The design specification to select standard components where possible restricts the choice of working fluids to those which are approved for use by the compressor manufacturer, namely the refrigerants R12, R22 and R502. The results of three runs of the VCCM program using the above operating conditions and heat pump design, but selecting each working fluid in turn are summarized in table 3.4. R12 results in the highest COP_h of 2.61 but due to its low suction vapour density it produces the lowest condenser heat output rate of 4203 W. R22 produces a higher COP_h and higher heat output rate than R502, and is inferior to R502 only with respect to the very high discharge temperature it produces. Indeed, R502 was initially introduced to replace R22 in applications in which the high discharge temperature leads to equipment unreliability, particularly low temperature operation of hermetic compressors (35, 36). In terms of thermodynamic performance, R22 is a compromise but is favoured over the other two fluids as it produces the highest heat output rate at an acceptable COP_h.

The high discharge temperature developed by R22 naturally begs the question of the thermal stability of the fluid. Barchardt, from Dupont, lists the maximum use temperature of R22 at 249°C, which is the lowest limiting temperature of the common fluorocarbons (28). The decompostion rate at this temperature is about 1 per cent per year, and the limit is considerably higher than the intended operation in this present application. Interestingly, he also reports that R22 has the least tendency to react with lubricating oil in the presence of steel, e.g. at the compressor discharge valve. On balance R22 is selected as the most suitable fluid.

3.2 The Influence of Compressor Characteristics on Heat Pump Performance

The model was used to determine the influence of the volumetric and isentropic efficiencies of the compressor on the overall performance of the heat pump, for the base case input data. Table 3.5 contains the detailed results of a systematic variation in compressor volumetric efficiency. The measured volumetric efficiency of the compressor selected for the prototype heat pump is 0.63 at base case conditions. Increasing the volumetric efficiency increases the refrigerant mass flow rate and hence the heat output rate, as expected. The relationship between heat output rate and volumetric efficiency is shown in figure 3.2 a). Note that the COP_h falls gradually with increased volumetric efficiency; this is due to the decrease in evaporating temperature and increase in condensing temperature resulting from the increased heat transfer rates across the heat exchangers.

Table 3.6 and figure 3.2 b) show the effect of changes in compressor isentropic efficiency. The measured value for the selected compressor is 0.47 at base case conditions. As expected, the COP_h rises with improved isentropic efficiency, reaching a value of 4.0 at an efficiency of 0.94. Heat output rate, however, falls slightly, due to the reduction of heat available in the desuperheater region of the condenser. This effect is illustrated in the figure by the plot of the proportion of heat recovered from the condenser obtained from the desuperheater. This ratio is 32.7 per cent at the base case isentropic efficiency, and falls to 16 per cent at an efficiency of 0.94. This reduction in desuperheating load also implies a lower compressor discharge temperature, T_2, and this is shown in the table.

3.3 Condenser Size and Operating Conditions

Table 3.7 summarizes the output from a series of runs of program VCCM in which the length of the condenser coil was progressively increased from 2 m to 20 m. The length of the CK8-20 condenser is 9.8 m. The principle effect of increasing the surface area of the condenser is of course to reduce the refrigerant condensing temperature. In turn this both increases the specific latent heat of the fluid in the condenser and increases the refrigerant mass flow rate by virtue of the reduction in compressor pressure ratio. The result is an increase in the heat transfer rate across the condenser and an improvement in COP_h. A secondary and minor effect is the slight depression in evaporating temperature, resulting from the general increase in capacity of the system, which tends to slightly reduce the improvement in COP_h. Figure 3.3 illustrates the reduction in T_c and improvement in $\dot{q}_c$ and COP_h with condenser size. Clearly the rate of increase in $\dot{q}_c$ and COP_h flattens out above about 12 m, suggesting further increases in size above this length are not worthwhile, as the performance of the heat pump is now limited by the capacities of the other major components, namely the compressor and evaporator. Table 3.7 also indicates that the water side pressure drop of the condenser may be excessive at the large condenser sizes.

A preliminary payback analysis of successive increases in condenser size is shown in table 3.8. The starting length is taken to be 4m, the analysis compares the marginal increase in capital cost of a larger condenser, assumed to be simply the cost of the extra condenser tubing, with the financial savings accruing from the improved COP_h. This shows that increases to about 10 m have payback times of less than two years and are therefore likely to be attractive. This analysis does not consider the additional benefit of the greater heat output rate available from the heat pump systems equipped with the large condensers, which would further favour the large condensers.

The effect of water flow rate through the condenser is shown in table 3.9. Increasing the water flow rate has a similar effect on system performance as increases in the condenser size, with successive increases producing progressively smaller improvements above a flow rate of about 0.9 m^3/h. Increases in the flow rate have a marked effect on the water side pressure drop, as shown in the table. At very small flow rates the temperature rise of the water is very high, leading to excessive water discharge temperatures. Table 3.10 summarizes the results of a series of runs of program VCCM in which the water inlet temperature was varied in line with the flow rate in order to maintain a constant mean water temperature. Since a building heat distribution system responds primarily to the mean water temperature, this approach maintains equilibrium between the heat pump and the heating system. This presentation tends to improve the COP_h results at low flow rates, compared with the constant inlet temperature approach of table 3.9.

3.4 Summary of Results from the Vapour Compression Cycle Model

The model has been used to aid the selection of the working fluid and to determine the influence of component performance on the behaviour of the heat pump. A summary of the sensitivity of the COP_h and heat output rate of the heat pump to changes in the design or operation of the compressor, condenser and evaporator is presented as table 3.11. The evaporator results are included in the sensitivity analysis for completeness, the detailed results of the evaporator design study are reported in the next chapter. The sensitivity of the performance indicators is expressed as the percentage change from the base case results for a change in the design variable of +10 per cent. The table demonstrates that the greatest further benefit to the COP_h of the base case design arises from an increase in the isentropic efficiency of the compressor. Furthermore, the greatest impact on heat output rate is brought about by improvements in the compressor volumetric efficiency. Hence it is improvements in compressor performance, rather than in the design of the heat exchangers, which are needed to improve the overall performance and hence attractiveness of the heat pump.

4. THE INTERACTION BETWEEN THE HEAT PUMP, HEAT DISTRIBUTION SYSTEM AND THE BUILDING - ANALYSIS

The objective of this analysis is to investigate the relationships between the heat pump, the heat distribution system and the building, in order to predict how the heat pump would perform in actual installations, and to identify the design and operational parameters which significantly affect the achieved performance. The analysis incorporates simplified models of the three main elements, and determines their interaction and transient response to the ambient temperature variation across a typical heating season.

4.1 Simplified Heat Pump Model

A heat pump model simpler than the vapour compression cycle model described earlier is justified by the saving in computer time. The behaviour of the heat pump is described by two equations which determine the steady state heat output and power requirement:

$$\dot{q}_c = A_1 + B_1 (T_{w2} - 55) + C_1 T_a$$

$$\dot{W} = \dot{W}_f + \dot{W}_c$$

where $\dot{W}_c = A_2 + B_2 (T_{w2} - 55) + C_2 T_a$

The coefficients A_1 to C_2 are found from the results of the VCCM runs for a particular heat pump design operating under prescribed water and air volumetric flow rates. The design and flow rates were chosen to represent the prototype heat pump, as described by the base case inputs of section 3. Transient effects within the heat pump itself are not considered because the time constants of the heat distribution system and the building are substantially longer than that of the heat pump. Thus, given the temperatures of the ambient air and the heating system water, the above equations determine the heat output rate and the electrical input power for the heat pump. For convenience the heat output rate may be expressed in terms of the water return temperature rather than the water flow temperature by:

$$\dot{q}_c = \frac{A_1 + B_1 \; (T_{w1} - 55) + C_1 T_a}{\left(1 - \dfrac{B_1}{\rho_w \, Cp_w \, \dot{V}_w}\right)}$$

The model incorporates an electrical supplementary heater located in the water flow pipe, downstream of the condenser, with its power output stepped in 2 kW stages up to a maximum of 6 kW. The control algorithm which governs the operation of the heat pump and supplementary heater is described in section 4.4. For simplicity the heat pump model does not include the effects of frost formation on the evaporator or defrosting, nor does it include an allowance for energy consumption while the heat pump is off, for example, for crankcase heating.

4.2 Heat Distribution System Model

The water filled steel panel radiator system is the most prevalent type of heat distribution system in the UK central heating market. For the purposes of modelling the entire radiator surface area of the system is lumped together and considered to respond to a single water temperature, the arithmetic mean of the flow and return temperatures from the heat pump. The instantaneous heat output rate depends on the radiator surface area and on the difference between the mean water temperature and the spatial mean room air temperature. The relationship used is:

$$\dot{q}_r = \dot{q}_{r,o} \left\{ \frac{(T_{wm} - T_{ar})}{(T_{wm,o} - T_{ar,o})} \right\}^n$$

$\dot{q}_{r,o}$ is the total radiative and convective heat output rate at a water to air temperature difference of $(T_{wm,o} - T_{ar,o})$. Thus the capacity of the heat distribution system is defined by specifying the heat output rate at a particular temperature difference, and the output rate at other water and air temperatures is found from the above equation. The exponent n was obtained from performance tables supplied by a leading radiator manufacturer to be 1.24 (38).

The thermal capacity of the heat distribution system comprises three major components; the mass of the steel radiators, the radiator water content, and the pipework water content. Clearly the radiator mass and water content are directly related to the total radiator area and hence to the heat dissipation rate. The following relationships were identified from manufacturer's catalogue data (38):

mass of steel $M_s = 15.7 \times \dot{q}_{r,60}$ [kg]

water content $M_{w,r} = 0.3 \times \dot{q}_{r,60}$ [kg]

where $\dot{q}_{r,60}$ is the total heat output rate in kW at a temperature difference of 60 deg C. The mass of steel applies to radiators with additional secondary surface area; unfinned panel radiators have slightly less mass per unit output. The water content of the pipework is entered as a constant, $M_{w,p}$. Thus the total thermal capacity of the heat distribution system is:

$$C_d = M_s \, Cp_s + (M_{w,r} + M_{w,p}) \, Cp_w$$

4.3 The Thermal Behaviour of the Building

The thermal behaviour of the building is described by an overall heat loss characteristic and a total thermal capacity. The internal room air temperatures are represented by the spatial mean temperature, T_{ar}. The heat loss rate is assumed to be proportional to the difference between the internal and ambient temperatures. The heat loss rate is specified to be $\dot{q}_{l,o}$ at an internal temperature of $T_{ar,o}$ and an ambient of $T_{a,o}$, usually -1°C. Then at any other temperature difference the heat loss rate is given by:

$$\dot{q}_l = \dot{q}_{l,o} \left\{ \frac{T_{ar} - T_a}{T_{ar,o} - T_{a,o}} \right\}$$

The heating requirement of the building is slightly less than the building heat loss rate, because of incidental heat gain. It is assumed that at some higher ambient temperature, $T_{a,1}$, taken to be 15°C, that the heat gain balances the heat loss rate so that the heating requirement is zero. Thus the heat gain is given by:

$$\dot{q}_g = \dot{q}_{l,o} \left\{ \frac{T_{ar,o} - T_{a1}}{T_{ar,o} - T_{a,o}} \right\}$$

It is further assumed that the heat gain is invariant with respect to room and ambient temperatures, thus the heating requirement is given by:

$$\dot{q}_d = \dot{q}_l - \dot{q}_g$$

$$\text{i.e.} \quad \dot{q}_d = \dot{q}_{l,o} \left\{ \frac{(T_{ar} - T_{ar,o}) - (T_a - T_{a1})}{(T_{ar,o} - T_{a,o})} \right\}$$

with the following typical values for the temperature constants:

$$T_{a,o} = -1°C$$
$$T_{a,1} = 15°C$$

$T_{ar,o} = 19°C$

The determination of representative values of the building thermal capacity, C_b, presented a difficulty. While there are well established, approximate calculation methods which may be applied to a particular design of building (39, 40), there is no generally accepted data base containing typical values for the generic designs and construction methods which make up the UK housing stock (41). Rather than perform detailed calculations on particular designs of housing, in which case the results would be specific to the designs chosen and dependent on the assumed material quantities, it was decided to use the measured thermal capacity from the heat pump field trial. Since room air temperature and the rate of heat output from the heat pump were frequently measured, the thermal capacity of the house could be directly measured. The building is a three bedroom, semi-detached, two storey house, with solid nine inch walls and brick internal walls built in 1936. Thus it is reasonable to expect this particular building to have a fairly high heat capacity, compared with modern designs.

4.4 Control of the Heat Pump

The model incorporates a simplified version of the complete prototype heat pump control system which is described in Chapter 6. The essential elements of the control system are:

a) Room temperature control

The heat pump is cycled on/off in response to the status of a room air thermostat. The model defines the operation of the thermostat by its set point, i.e. the temperature at which the thermostat opens to switch off the heat pump, and by its switching differential. Hence the temperature at which the thermostat closes to restart the heat pump is given by the set point minus the differential. It is assumed that the room thermostat adopts the mean building air temperature.

b) Water temperature control

The operation of the heat pump is also subject to the supervision of a flow water thermostat which shuts down the heat pump when the condenser water outlet temperature exceeds the set point.

c) Control of the supplementary heater

The sequencing of the electric supplementary heater is governed by two time delays. The 2 kW stage is energized if the heat pump alone does not satisfy the room thermostat after Dt1 minutes of operation. Subsequent 2 kW increases in supplementary heating are permitted if the room thermostat remains closed for a further Dt2 minutes. When the room thermostat opens the supplementary heater is switched off, and the timer Dt1 once again begins counting when the room thermostat re-closes. The supplementary heater is also subject to supervision from a maximum temperature thermostat which shuts down the heater when the water flow temperature exceeds the set point. In addition, in order to ensure that priority is given to the heat pump, the supplementary heater is made to step down one 2 kW stage when the heat pump is cycled off due to high water flow temperature.

4.5 Ambient Temperature Distribution Data

The variation in ambient air temperature is taken from long run average data supplied by the Meteorological Office (42). The data is based on hourly observations taken at Watnall, near Nottingham, for the period 1971 to 1982. For convenience, they have been expressed as the average number of hours per year for which the temperature falls within one degree C intervals in the range -8°C to 15°C, as shown in table 3.12.

4.6 The Interaction Between the Three Elements and Numerical Solution Procedure

The interaction between the heat pump, heat distribution system and the building is described by the following two energy balance equations for the internal air and radiator water respectively:

$$C_b \; \frac{dT_{ar}}{dt} = \dot{q}_r - \dot{q}_d$$

$$C_d \; \frac{dT_{w1}}{dt} = \dot{q}_c + \dot{q}_b - \dot{q}_r$$

The equations are numerically solved for T_{ar} and T_{w1} using finite differences in the FORTRAN computer program entitled DSI, which is short for "Dynamic Simulation of the Interaction Between the Heat Pump, Heat Distribution System and the Building". This program adopts the following procedure:

a) The ambient air temperature is varied from +15°C to -8°C in steps of 1 deg C. Each temperature interval is represented by the mid-point temperature, e.g. the interval 4 to 4.9°C is denoted as 4.5°C.

b) 24 hours of operation of the heat pump are simulated for each ambient temperature interval. The time step must be short enough so that the principal temperatures are reasonably constant during the time step, but of course a very small time step requires considerable computational effort. The time step chosen was 2 minutes.

c) At the start of the 24 hour simulation both the room air and water return temperatures are set to the value of the room thermostat set point. This is to ensure each simulation has the same starting point.

d) The program then marches forward in time through the simulation period in steps of 2 minutes. For each step the energy transfers associated with the heat pump, radiator network and the building are calculated. The temperature changes resulting from these energy transfers are found from the finite difference versions of the above differential equations:

$$C_b \frac{(T_{ar,t+\Delta t} - T_{ar,t})}{\Delta t} = \dot{q}_r - \dot{q}_d$$

$$C_d \frac{(T_{wl,t+\Delta t} - T_{wl,t})}{\Delta t} = \dot{q}_c + \dot{q}_b - \dot{q}_r$$

e) The status of each functional element in the control system is examined at the end of each time step. The program contains a series of statements which emulate the control functions described above.

f) At the end of the 24 hour simulation period the total energy transfers are calculated along with the details of the average cycle time and average temperatures. The mean radiator water temperature is time averaged across two time periods; the first is the total time, the second is the time for which the heat pump is actually producing heat.

g) When all the ambient temperature intervals have been processed in this way, the seasonal performance of the heating system is assessed by calculating the weighted total of the heat delivered, electricity consumption and the contribution from the supplementary heater. The weighting is the number of hours per typical year that the particular ambient temperature prevails.

The program contains facilities to review and alter the input data used in the analysis, for example building heat demand characteristics, radiator output capacity, thermostat settings and so on. The user may also select the degree of detail in the presentation of the results, from the minimum case of a summary of the performance across the heating season to the full transient results for each ambient temperature.

5. RESULTS OF THE SIMULATION EXERCISE

5.1 Base Case Data and Results

The findings from the simulation study are presented in the form of the detailed results from the base case run and then as a sensitivity analysis to illustrate the significance of the major variables. The base case data refers to a three bedroom, semi-detatched house with a radiator output capacity equal to the building heat loss rate at -1°C and a water flow temperature of 55°C. The data is summarized as:

a) House details

Heat loss rate at an ambient of -1°C and an internal mean temperature of 19°C:	6 kW
Heat demand is zero at an ambient of	15°C
Building thermal capacity (as measured in the field trial)	16.2 MJ/K (4.5 kWh/deg C)

b) Heat distribution system details

Heat output rate at a water flow temperature of 55°C and a flow rate of 1 m^3/h, with an internal air temperature of 19°C	6 kW
Heat output rate at a water flow temperature of 80°C with the same flow rate and internal temperature	11.4 kW
System thermal capacity, comprising radiator steel panels, radiator and pipework water content	331.2 kJ/K

c) Control settings

Room air thermostat : set point 19°C, differential 0.5 deg C

Condenser outlet thermostat : set point 55°C, differential 5.0 deg C

Supplementary heater thermostat : set point 85°C, differential 5.0 deg C

Time delay Dt1 120 minutes

Time delay Dt2 60 minutes.

For convenience it is assumed that the heating system is operated continuously throughout the heating season.

The results of the base case study, presented as table 3.13, contain a number of interesting features. The time averaged room air temperature is held within the 0.5 deg C switching differential of the room thermostat until the ambient temperature falls below -1°C. Below -1°C the radiators cannot dissipate sufficient heat to satisfy the building heat demand even at the maximum output temperature of the heat pump. The result is a reduction in the achieved internal air temperature. This reduction occurs in spite of the efforts of the supplementary heater, which, because of the adopted control strategy, is made to reduce heat output when the heat pump is switched off due to high water temperature. Thus the major effect of the supplementary heater is to decrease the response time of the heating system. The contribution of the supplementary heater is shown in figure 3.4 a) along with the distribution of the building heat demand across the ambient temperature range. The ambient temperature interval with the greatest heat demand is 4 to 5°C.

Figure 3.4 b) illustrates the difference between the mean water temperature while the heat pump is running and the overall time averaged water temperature. Clearly the heat pump is forced to operate at a higher water temperature, and therefore at a lower COP_h, than is required by the heat distribution system, because of the on/off method of capacity control.

5.2 Radiator Heat Dissipation Rate

The heat dissipation rate of the radiator network was varied between 3 kW and 12 kW at a flow temperature of 55°C. The heat capacity of the system was permitted to vary in accordance with the heat output rates, as described in section 4.2. Two sets of runs were carried out. The first applied the control strategy described in section 4.4, in which the operation of the heat pump is given priority over the supplementary heater by the action of stepping down the power level of the heater by 2 kW when the heat pump is cycled off due to high water temperature. The results of this simulation run are shown in table 3.14 a). Notice that the small radiator sizes lead to a considerable reduction in the achieved room temperature. This is because the heat pump cannot operate at the high water temperatures necessary for the small radiator areas, and the control strategy prevents the supplementary heater from making a greater contribution.

The second set of runs relaxes this constraint on the supplementary heater, i.e. it may continue operation as normal when high water temperature causes the heat pump to be switched off. The results are shown as table 3.14 b). The achieved room temperature is higher under this regime, and thus more likely to be acceptable to the end user.

The effect of radiator size on seasonal COP_h for both control strategies is illustrated in figure 3.5. The main abscissa is the heat output rate for the entire installed radiator network when operated at a water flow temperature of 55°C. The second scale shows the resultant heat output rate which would occur from the same installed area operating at the more usual flow temperature of 80°C. The COP_h characteristic has two distinct regions; a steeply rising portion below about 6 kW of installed radiator output where small increases in the radiator area lead to significant improvements in COP_h, and a less steep region above 6 kW of dissipation capacity.

The steep section of the characteristic is due to the excessive use of the supplementary heater, in circumstances in which the heat pump may well have adequate heat output capacity but is unable to operate because of the high water temperatures. As may be expected, this effect is more pronounced for the control strategy which does not inhibit the contribution of the supplementary heater. The general shape of the COP_h curve has been noted by others (43). The implications for the satisfactory operation of domestic heat pumps are clear; the radiator area must be sufficient to dissipate the building heat load at -1°C at a water flow temperature of 55°C. In terms of a radiator system originally designed to operate with a flow temperature of 80°C, for example for an oil fired boiler installation, the heat output capacity of the radiator system may need to be doubled to fulfill this requirement. However, the required increase in radiator output would be less than this if action were taken to reduce the heat demand of the building by thermal insulation measures, as this would have the effect of oversizing the original radiator capacity.

5.3 Building Heat Demand

The simulation program was executed with the building heat demand at -1°C changing from 3 kW to 12 kW. The total radiator output rate was set to meet the demand at a water flow temperature of 55°C. The thermal capacity of the building structure was held constant at the base case value. The control system followed the strategy set out in section 4.4.

The results are presented in table 3.15. The second column specifies the ambient temperature at which the supplementary heater is first brought into service. This is not the balance point temperature, but rather an indication of the ambient temperature at which the heat pump is unable to satisfy the building heating requirement within a reasonable period of time. Clearly the overall COP_h of the device falls with increasing building heat loss, although interestingly a 50 per cent increase in the building heat load from the nominal capacity of the heat pump of 6 kW produces a fall in COP_h of only 12.4 per cent. The reduction in the achieved room temperatures accompanying this change is more significant, suggesting that this particular size of heat pump is unsuitable for houses with heat demands greater than about 8 kW.

5.4 The Thermal Capacity of the Heat Distribution System

The water content of the heat distribution system was varied independently of the radiator heat dissipation rate. The range covered was 10 kg to 500 kg, the base case value being 57.2 kg. The results are shown in table 3.16. A very small water content is detrimental to the performance of the system because of the resultant high mean water temperatures. Indeed the transient results demonstrated that the heat pump was frequently cycled off by the outlet water thermostat for small water content systems. This occurs when the time taken to heat the constituents of the heat distribution system is shorter than the time required to satisfy the room thermostat.

The results also indicate a penalty associated with very large water contents; the device COP_h begins to fall at the water content of 400 kg. With such large water masses the response time of the heating system is very long, hence the control system brings the supplementary heater into service at higher ambient tempertures than would otherwise be the case, and this reduces the device COP_h.

The optimum water content, for the base case data, lies between 70 to 300 kg. The water content of the double panel radiator system for the base case is 57.2 kg which is outside the optimum range, but table 3.15 suggests that this will have only a minor impact on the seasonal COP_h of the device.

5.5 Building Thermal Capacity

The thermal capacity of the building was varied independently of the building heat demand, from 0.5 to 10 kWh/deg C (1.8 to 36 MJ/K). The base case value is 4.5 kWh/deg C (16.2 MJ/K). The results of the simulation runs are summarized in table 3.17. High building thermal capacity tends to reduce the seasonal COP_h because of the greater response time of the building to the heating stimulus. This leads to generally higher heat sink temperatures and therefore a lower cycle COP_h, and to greater use of the supplementary heater which, of course, significantly reduces the device COP_h. Hence the continuing trend towards highly insulated, low thermal mass housing should favour the heat pump.

5.6 Room Air Thermostat Switching Differential

The base case value for this parameter is 0.5 deg C, which can be readily achieved by low cost thermostats in present use and some manufacturers claim 0.15 to 0.25 deg C (44). The simulation program was executed with several values of the room thermostat differential, from 0.1 to 2 deg C. The results are summarized in table 3.18. A large switching differential has a similar effect on heat pump performance as a large thermal capacity of the building; the run time of the heat pump is higher, leading to high heat sink temperatures and greater usage of the supplementary heater. Hence close control over the room temperature tends to improve the peformance of the heat pump.

5.7 Summary of the Influences on Heat Pump Performance

In conclusion, the simulation exercise has identified a number of parameters which affect the achieved performance of the heat pump when operating as part of a building heating system. The sensitivity analysis of table 3.19 summarizes the results of the numerical explorations. The change in the seasonal COP_h of the heat pump as a result of a 10 per cent increase in each independent variable is presented for the five parameters studied. The table shows that increases in the radiator area have the greatest effect on COP_h, producing a 4.8 per cent increase for a 10 per cent increase in radiator heat output. Reducing the building heating requirement by 10 per cent is likely to improve the COP_h by 3.3 per cent. The effects of small changes in the thermal capacity of the heat distribution system and the building or in the room thermostat differential are small in comparison with those associated with changes in the radiator area and the building heating requirement.

The importance of correct sizing of the radiators in heat pump installations has been reported by others (43, 45). The influences of variables not studied in this present simulation have been identified by others. For example, by means of a simulation exercise which included the transient response of the heat pump and detailed models of the solar and other incidental gains to the building, Henderson demonstrated that for highly insulated dwellings the solar gain can reduce the load factor and the seasonal COP_h of the heat pump (45).

A favourable installation for the present heat pump would have the following features:

a) A radiator network with an output rate at a flow temperature of 55°C at least equal to and preferably greater than the heat demand of the building at -1°C.

b) A building heat demand at -1°C of no greater than 8 kW.

c) A low building thermal capacity, of 2.5 kWh/deg C (9 MJ/K) or less.

d) A close control room air thermostat.

An important conclusion of this study is that the achieved performance of a particular design of heat pump will vary between different houses, because of the inherent diversity in the above factors between installations.

6. CONCLUDING REMARKS

This chapter has reported the details of and the results from two computer models developed as part of the present research. The vapour compression model, VCCM is a useful design tool, as it can quantify the effects of changes in the performance of the major components, and predict the behaviour of the system across a range of operating conditions. The simulation model, DSI, predicts the performance of the heat pump when operated as part of a building heating system, and has been used to identify those factors which contribute towards a successful domestic heat pump installation.

		CONDENSER		
		CK8-20	KWG 3X	
l	pass length	9.800	7.045	m
d_1	inner tube bore	19.7	18.5	mm
d_2	root outside diameter	22.2	21.0*	mm
d_3	outer tube bore	38.1	34.7	mm
d_e	effective diameter	2.4	2.27 *	mm
A_i	internal (water side) area	0.607	0.409	m^2
$A_o = A_p + \eta_f A_s$ (put $\eta_f = 1.0$)		2.013	1.356*	m^2
R_f	relative roughness	1.12	2.29	
C_1		0.689	0.689	
$^1/h_f$	water side fouling coefficient (as received condition)	0	0	m^2K/W

* Estimated

TABLE 3.1: GEOMETRICAL AND FLUID FLOW DATA FOR TWO COAXIAL TUBE CONDENSERS

Test Run	Model Input			Model Output			Experimental Results			Difference between model output and experimental results		
	T_1 (°C)	P_1 (bar, abs)	T_c (°C)	$\dot{m}_r$ (kg/s)	$\dot{W}_c$ (kW)	T_2 (°C)	$\dot{m}_r$ (kg/s)	W_c (external) (kW)	T_2 (°C)	$\dot{m}_r$ (% of exp.result)	$\dot{W}_c$ (% of exp.result)	T_2 (deg C)
1	-4.8	3.37	33.7	0.0247	1.81	102.0	0.0245	1.86	100.2	+0.8%	-2.7%	+1.8
2	-7.4	3.58	56.2	0.0216	2.30	147.0	0.0216	2.36	137.0	0%	-2.5%	+10.0
15	4.8	4.84	50.5	0.0347	2.64	119.0	0.0345	2.59	110.4	+0.6%	+1.9%	+8.6
20	9.2	5.36	45.6	0.0405	2.54	104.2	0.0404	2.48	98.8	+0.2%	+2.4%	+5.4
25	14.6	6.20	48.7	0.0473	2.76	104.1	0.0471	2.73	98.8	+0.4%	+1.1%	+5.3

Source of experimental data: Chapter 5

Test runs selected to cover the working range of a heat pump compressor.

TABLE 3.2: VALIDATION OF COMPRESSOR MODEL - COMPARISON WITH EXPERIMENTAL RESULTS

Test	Model Input				Model Output				Experimental Results				Difference in Output from Model		
Run	T_1 (°C)	P_1 (bar,abs)	T_{w1} (°C)	$\dot{V}_w$ (m^3/h)	ΔP_w (kN/m^2)	T_{w2} (°C)	T_c (°C)	$\dot{q}_c$ (kW)	ΔP_w (kN/m^2)	T_{w2} (°C)	T_c (°C)	$\dot{q}_c$ (external) (kW)	T_{w2} (degC)	T_c (degC)	$\dot{q}_c$ (% of exp)
a) CK8-20 condenser															
Y16	-11.5	3.02	48.8	0.60	2.3	54.7	53.8	4.06	2.1	54.9	54.0	4.20	-0.2	-0.2	-3.2%
Y12	-3.0	4.09	46.4	0.60	2.3	55.1	54.6	6.00	2.1	55.1	54.4	5.99	0	+0.2	+0.2%
Y10	-2.8	4.09	49.7	1.00	5.7	54.9	55.0	5.97	5.7	54.8	54.7	5.90	+0.1	+0.3	+1.2%
Y2	15.0	6.56	46.3	1.00	5.7	54.9	56.3	9.92	5.7	55.0	55.2	9.98	-0.1	+1.3	-0.6%
b) KWG 3X condenser															
WB4	-10.9	3.08	49.8	0.62	4.9	55.5	55.6	4.06	4.8	55.3	54.9	3.90	+0.2	+0.7	+4.1%
WB2	-11.1	3.00	51.5	1.03	12.0	54.8	55.2	3.93	12.5	54.7	54.8	3.82	+0.1	+0.4	+2.9%
WB6	-3.4	3.85	49.5	1.01	11.6	54.3	55.4	5.52	11.5	54.2	54.5	5.42	+0.1	+0.9	+1.9%
WB14	14.5	6.35	46.3	1.02	11.8	54.4	57.8	9.50	11.5	54.0	55.5	8.97	+0.4	+2.3	+5.9%

Source of experimental data: Chapter 5

TABLE 3.3: VALIDATION OF CONDENSER MODEL - COMPARISON WITH EXPERIMENTAL RESULTS FOR TWO CONDENSERS

Refrigerant	Cycle							Compressor			Heat and Work Transfer Rates		
	T_e (°C)	T_1 (°C)	T_2 (°C)	T_c (°C)	P_1 (bar,abs)	P_2 (bar,abs)	$\frac{P_2}{P_1}$	η_v	η_{is}	$\dot{m}_r$ (kg/s)	$\dot{W}_c$ (W)	$\dot{q}_c$ (W)	$\frac{\dot{q}_c}{\dot{W}_c}$
R12 CCl_2F_2	-0.4	1.8	97.6	53.6	2.86	13.24	4.63	0.64	0.47	0.0273	1610	4203	2.61
R22 $CHClF_2$	-1.9	1.1	131.4	55.2	4.53	21.84	4.82	0.63	0.47	0.0307	2680	6718	2.51
R502 Azeotrope[1]	-1.8	0.8	96.9	55.9	5.18	23.89	4.61	0.64	0.47	0.0472	2810	6384	2.27

1 R502 consists of 48.8% R22 and 51.2% R115 by weight (R115 is $CClF_2$-CF_3)

TABLE 3.4: PERFORMANCE COMPARISON BETWEEN R12, R22 and R502 FOR AIR TEMPERATURE OF 4.0°C AND WATER RETURN TEMPERATURE OF 49°C. PREDICTED RESULTS FROM PROGRAM VCCM

η_v	η_{is}	T_e (°C)	T_2 (°C)	T_c (°C)	$\dot{m}_r$ (kg/s)	$\dot{q}_c$ (W)	$\dot{W}_c$ (W)	$\frac{\dot{q}_c}{\dot{W}_c}$
0.36	0.48	-0.6	117.8	52.8	0.0187	3963	1480	2.678
0.40	0.47	-0.8	120.4	53.2	0.0209	4450	1678	2.652
0.45	0.47	-1.0	122.8	53.6	0.0230	4930	1878	2.625
0.49	0.47	-1.3	125.0	54.0	0.0250	5401	2081	2.595
0.54	0.47	-1.5	127.4	54.4	0.0269	5843	2280	2.563
0.58	0.47	-1.7	129.4	54.8	0.0289	6295	2484	2.534
0.63	0.47	-1.9	131.4	55.2	0.0307	6718	2680	2.507
0.67	0.47	-2.1	133.4	55.6	0.0326	7178	2897	2.478
0.72	0.46	-2.3	135.3	55.9	0.0344	7610	3105	2.451

TABLE 3.5: THE INFLUENCE OF COMPRESSOR VOLUMETRIC EFFICIENCY ON HEAT PUMP PERFORMANCE. PREDICTED RESULTS FOR R22

η_{is}	T_2 (°C)	$\frac{\dot{q}_{c,d}}{\dot{q}_{c,d} + \dot{q}_{c,c}}$ (%)	$\dot{q}_{c,d} + \dot{q}_{c,c}$ (W)	$\dot{W}_c$ (W)	$\frac{\dot{q}_{c,d} + \dot{q}_{c,c}}{\dot{W}_c}$
0.33	173.7	42.3	7868	3815	2.062
0.38	155.0	38.4	7360	3314	2.221
0.43	140.6	35.1	6970	2929	2.380
0.47	131.4	32.7	6718	2680	2.507
0.48	129.3	32.1	6661	2624	2.538
0.53	120.1	29.5	6412	2376	2.699
0.58	112.5	27.2	6205	2171	2.858
0.63	106.2	25.1	6031	1998	3.019
0.68	100.8	23.2	5883	1851	3.178
0.73	96.2	21.5	5756	1724	3.339
0.78	92.2	20.0	5645	1613	3.500
0.83	88.4	18.5	5549	1515	3.663
0.88	85.7	17.3	5463	1429	3.823
0.93	83.0	16.1	5387	1351	3.987
0.98	80.6	15.0	5319	1282	4.149

TABLE 3.6: THE INFLUENCE OF COMPRESSOR ISENTROPIC EFFICIENCY ON HEAT PUMP PERFORMANCE. PREDICTED RESULTS FOR R22

Condenser length l (m)	T_e (°C)	T_2 (°C)	T_c (°C)	$\frac{P_2}{P_1}$	η_v	η_{is}	$\dot{m}_r$ (kg/s)	$\dot{q}_c$ (W)	$\dot{W}_c$ (W)	$\frac{\dot{q}_c}{\dot{W}_c}$	ΔP_w (kN/m^2)
2	-0.5	169.9	73.5	6.72	0.52	0.44	0.0267	5815	3054	1.904	1.17
3	-1.1	153.9	66.1	5.91	0.57	0.45	0.0287	6256	2948	2.122	1.75
4	-1.4	145.3	62.1	5.49	0.59	0.46	0.0295	6456	2862	2.256	2.33
5	-1.6	140.2	59.6	5.24	0.60	0.46	0.0300	6569	2803	2.343	2.92
6	-1.7	137.2	58.0	5.09	0.61	0.46	0.0302	6609	2758	2.396	3.50
7	-1.8	134.9	56.9	4.98	0.62	0.47	0.0304	6653	2728	2.439	4.08
8	-1.8	133.3	56.1	4.91	0.62	0.47	0.0305	6683	2707	2.469	4.67
9	-1.8	132.1	55.5	4.85	0.62	0.47	0.0306	6705	2690	2.493	5.25
9.8	-1.9	131.4	55.2	4.82	0.63	0.47	0.0307	6718	2680	2.507	5.72
11	-2.0	130.4	54.7	4.78	0.63	0.47	0.0307	6734	2667	2.525	6.42
12	-2.0	129.9	54.5	4.75	0.63	0.47	0.0308	6744	2659	2.536	6.99
15	-2.0	128.7	53.9	4.70	0.63	0.47	0.0309	6764	2644	2.558	8.75
20	-2.1	127.8	53.5	4.66	0.64	0.47	0.0309	6780	2630	2.578	11.67

TABLE 3.7: THE INFLUENCE OF CONDENSER SIZE ON HEAT PUMP PERFORMANCE

Condenser length (m)	Increase in capital cost of the heat pump (£)	Predicted[1] COP_h	Cost per[2] useful kWh of heat delivered (p/kWh)	Cost of[3] electricity used (£/yr)	Annual saving resulting from larger condenser (£/yr)	Simple payback period (years)
4	0	2.256	1.886	339.48		
6	20	2.396	1.775	319.50	19.98	1.0
8	40	2.469	1.723	310.14	29.34	1.4
9.8	58	2.507	1.697	305.46	34.02	1.7
12	80	2.536	1.677	301.86	37.62	2.1
15	110	2.558	1.663	299.34	40.14	2.7

NOTES:

1 At T_{a1} = 4°C, T_{w1} = 49°C

2 The Domestic Economy 7 tariff is assumed, ie an off peak rate applies for a seven hour period between midnight and 08:00 GMT. The Electricity Council indicated the UK average charges as: day 5.64p/kWh, night 2.04p/kWh as of 1/4/85. A mix of 38.5% night, 61.5% day is assumed

3 Assuming a total annual heating requirement of 18000 kWh

TABLE 3.8: PAYBACK ANALYSIS FOR INCREASES IN THE SIZE OF THE CONDENSER

$\dot{V}_w$ (m^3/h)	h_i (W/m^2K)	T_e (°C)	T_c (°C)	$\frac{P_2}{P_1}$	η_v	$\dot{m}_r$ (kg/s)	$\dot{q}_c$ (W)	$\frac{\dot{q}_c}{\dot{W}_c}$	ΔP_w (kW/m^2)	T_{w2} (°C)	$\frac{T_{w1} + T_{w2}}{2}$ (°C)
0.2	979	-0.9	68.7	6.19	0.55	0.0281	6119	1.945	0.34	75.6	62.3
0.4	1697	-1.4	61.2	5.40	0.59	0.0297	6495	2.170	1.15	63.1	56.1
0.6	2383	-1.7	58.0	5.09	0.61	0.0302	6609	2.273	2.34	58.6	53.8
0.8	2970	-1.8	56.3	4.92	0.62	0.0305	6676	2.333	3.87	56.3	52.7
0.9	3264	-1.9	55.7	4.86	0.62	0.0306	6699	2.355	4.75	55.5	52.3
1.0	3558	-1.9	55.2	4.82	0.63	0.0307	6718	2.373	5.72	54.8	51.9
1.1	3851	-2.0	54.7	4.78	0.63	0.0307	6734	2.390	6.75	54.3	51.7
1.2	4145	-2.0	54.4	4.74	0.63	0.0308	6747	2.403	7.86	53.9	51.5
1.4	4667	-2.0	53.8	4.70	0.63	0.0309	6767	2.424	10.29	53.2	51.1
1.6	5190	-2.1	53.4	4.65	0.64	0.0309	6782	2.441	13.00	52.7	50.9

TABLE 3.9: THE EFFECT OF WATER FLOW RATE ON HEAT PUMP PERFORMANCE. CONDENSER INLET TEMPERATURE HELD AT 49°C

$\dot{V}_w$ (m^3/h)	T_{w1} (°C)	T_{w2} (°C)	$\frac{T_{w1} + T_{w2}}{2}$ (°C)	T_e (°C)	T_c (°C)	$\dot{q}_c$ (W)	$\frac{\dot{q}_c}{\dot{W}_c}$
0.2	38.6	66.8	52.7	-1.4	61.5	6477	2.161
0.4	44.8	59.2	52.0	-1.7	57.8	6642	2.285
0.6	47.1	56.8	52.0	-1.8	56.4	6674	2.331
0.8	48.2	55.5	51.9	-1.9	55.6	6703	2.359
0.9	48.6	55.1	51.9	-1.9	55.3	6713	2.368
1.0	49.0	54.8	51.9	-1.9	55.2	6718	2.373
1.1	49.2	54.5	51.9	-1.9	54.9	6727	2.383
1.2	49.4	54.3	51.9	-2.0	54.8	6733	2.389
1.4	49.8	54.0	51.9	-2.0	54.6	6740	2.397
1.6	50.0	53.7	51.9	-2.0	54.3	6748	2.405

TABLE 3.10: THE EFFECT OF WATER FLOW RATE ON HEAT PUMP PERFORMANCE AT CONSTANT MEAN WATER TEMPERATURE

Parameter	Base case value	Base case value plus 10%	Effect on COP_h $\frac{\dot{q}_c}{\dot{W}_c+\dot{W}_f}$ Expressed as the percentage change compared with the base case result (%)	Effect on $\dot{q}_c$ (%)
Compressor volumetric efficiency	0.63	0.693	-1.3	+9.8
Compressor isentropic efficiency	0.47	0.517	+5.3	-3.6
Condenser tube length (m)	9.8	10.78	+0.6	+0.2
Evaporator tube length (m)	30	33	+0.2	+0.5

TABLE 3.11: SENSITIVITY OF HEAT PUMP OUTPUT AND COP_h TO 10% INCREASES IN THE VALUES OF FOUR IMPORTANT DESIGN VARIABLES

Ambient temperature interval (deg C)			Average number of hours per year for the period 1971-1982 at Watnall
-8	to	-7.1	4.5
-7	to	-6.1	5.6
-6	to	-5.1	9.8
-5	to	-4.1	18.3
-4	to	-3.1	30.7
-3	to	-2.1	54.6
-2	to	-1.1	98.4
-1	to	-0.1	176.2
0	to	0.9	277.1
1	to	1.9	320.1
2	to	2.9	378.4
3	to	3.9	437.6
4	to	4.9	499.4
5	to	5.9	513.7
6	to	6.9	550.3
7	to	7.9	557.0
8	to	8.9	558.1
9	to	9.9	563.7
10	to	10.9	533.7
11	to	11.9	509.9
12	to	12.9	492.9
13	to	13.9	430.8
14	to	14.9	377.8

TABLE 3.12: AMBIENT TEMPERATURE FREQUENCY DATA USED IN THE SIMULATION

SOURCE: METEOROLOGICAL OFFICE

Ambient temp. T_a (°C)	Heat pump and supplementary heater energy transfers W_c+W_f (kWh)	Q_c (kWh)	Q_b (kWh)	$\frac{Q_c+Q_b}{W_c+W_f+W_b}$	Time ave. room temp. T_{ar} (°C)	Time ave. mean water temp. $\frac{T_{w1}+T_{w3}}{2}$ (°C)	Mean water temp. while heat pump is running $\frac{T_{w1}+T_{w3}}{2}$ (°C)	Average heat pump run time (mins)
14.5	0	0	0	-	18.7	18.7	18.7	0
13.5	61.0	192.8	0	3.16	18.9	22.0	39.7	12.0
12.5	145.8	454.2	0	3.12	18.9	25.0	40.1	12.7
11.5	207.2	631.2	0	3.05	18.9	26.0	40.9	14.0
10.5	283.1	844.0	0	2.98	18.9	29.0	41.7	15.3
9.5	378.7	1104.5	0	2.92	18.8	31.2	42.4	16.0
8.5	476.0	1356.5	0	2.85	18.8	33.6	43.1	14.1
7.5	539.8	1497.5	0	2.77	18.8	35.2	44.1	16.1
6.5	617.5	1673.7	0	2.71	18.8	36.9	44.7	18.2
5.5	667.3	1764.6	0	2.64	18.7	38.8	45.4	19.5
4.5	763.3	1950.4	0	2.56	18.7	41.3	46.6	20.7
3.5	707.8	1756.8	59.6	2.37	18.7	42.8	47.4	24.4
2.5	645.1	1559.8	144.0	2.16	18.7	44.4	48.0	24.7
1.5	590.4	1384.0	226.7	1.97	18.8	46.7	48.8	20.9
0.5	507.5	1151.5	301.7	1.80	18.6	48.2	49.6	15.7
-0.5	343.9	755.5	242.8	1.70	18.6	49.7	50.3	15.7
-1.5	200.2	425.1	159.1	1.63	18.3	50.7	51.0	17.2
-2.5	115.8	239.9	90.8	1.60	17.9	50.8	51.1	23.9
-3.5	68.2	137.2	52.7	1.57	17.5	51.0	51.3	39.2
-4.5	41.1	80.5	35.0	1.52	17.0	51.1	51.4	66.4
Totals	7359.7	18959.7	1312.4	2.34				

TOTAL HEAT DELIVERED IS 20272.1 kWh, OF WHICH 6.5% IS SUPPLEMENTARY

TABLE 3.13: DETAILED BREAKDOWN OF RESULTS FROM SIMULATION USING BASE CASE DATA

Radiator output at 55°C flow temperature (kW)	Approximate radiator output at 80°C flow temperature (kW)	Proportion of total heat load provided by supplementary heater $\frac{Q_b}{Q_c+Q_b}$ (%)	Cycle COP_h $\frac{Q_c}{W_c+W_f}$	Device COP_h $\frac{Q_c+Q_b}{W_c+W_f+W_b}$	Mean room temperature at -1°C ambient
a) Priority given to heat pump: supplementary heater steps down 2kW when thermostat opens					
3	5.8	39.8	2.46	1.56	14.7
4	7.7	22.6	2.48	1.86	16.1
5	9.6	12.6	2.52	2.11	17.4
6	11.5	6.5	2.58	2.34	18.4
7	13.5	4.0	2.69	2.52	18.4
8	15.4	3.8	2.83	2.64	18.4
9	17.3	3.5	2.94	2.75	18.5
10	19.2	3.2	3.04	2.85	18.5
11	21.2	2.9	3.13	2.94	18.5
12	23.1	2.8	3.20	3.02	18.5
b) No particular priority given to heat pump					
3		60.4	2.60	1.32	17.6
4		44.6	2.61	1.52	17.9
5		27.0	2.61	1.82	18.2
6		13.1	2.62	2.16	18.5
7		6.0	2.71	2.46	18.4
8		4.1	2.83	2.63	18.4
9		3.5	2.94	2.75	18.5
10		3.2	3.04	2.85	18.5
11		2.9	3.13	2.94	18.5
12		2.8	3.20	3.02	18.5

TABLE 3.14: THE EFFECT OF RADIATOR AREA ON THE SEASONAL PERFORMANCE OF THE HEAT PUMP

Building heating requirement at -1°C	Ambient temp. at which supplementary heater is first used	Proportion of the total load provided by the supplementary heater $\frac{Q_b}{Q_c+Q_b}$	Cycle COP_h $\frac{Q_c}{W_c+W_f}$	Device COP_h $\frac{Q_c+Q_b}{W_c+W_f+W_b}$	Mean room temperature at -1°C ambient
(kW)	(°C)	(%)			(°C)
3	-2	0.1	2.75	2.74	18.7
4	0	0.8	2.68	2.65	18.7
5	3	3.0	2.62	2.50	18.5
6	4	6.5	2.58	2.34	18.4
7	6	10.0	2.55	2.21	17.4
8	7	12.7	2.53	2.12	17.0
9	8	14.9	2.52	2.05	15.7
10	8	16.4	2.51	2.01	15.0
11	9	17.7	2.50	1.98	14.4
12	9	18.4	2.50	1.96	13.8

TABLE 3.15: THE EFFECT OF BUILDING HEAT DEMAND ON THE SEASONAL PERFORMANCE THE HEAT PUMP

System water content	Thermal capacity of heat distribution system	Proportion of total heat load provided by supplementary heater $\frac{Q_b}{Q_c+Q_b}$	Cycle COP_h $\frac{Q_c}{W_c+W_f}$	Device COP_h $\frac{Q_c+Q_b}{W_c+W_f+W_b}$
(kg)	(kJ/K)	(%)		
10	132.4	6.4	2.54	2.31
30	216.0	6.0	2.55	2.33
50	299.6	6.3	2.57	2.34
70	383.3	6.4	2.60	2.36
100	508.7	6.9	2.62	2.36
300	1344.9	9.3	2.75	2.36
400	1763.0	10.9	2.77	2.33
500	2181.1	12.6	2.81	2.29

TABLE 3.16: THE EFFECT OF THE THERMAL CAPACITY OF THE HEAT DISTRIBUTION SYSTEM ON THE SEASONAL PERFORMANCE OF THE HEAT PUMP

Building thermal capacity		Proportion of total heat load provided by supplementary heater $\frac{Q_b}{Q_c+Q_b}$	Cycle COP_h $\frac{Q_c}{W_c+W_f}$	Device COP_h $\frac{Q_c+Q_b}{W_c+W_f+W_b}$
(kWh/deg C)	(MJ/K)	(%)		
0.5	1.8	2.3	2.74	2.64
1.5	5.4	3.5	2.67	2.52
2.5	9.0	4.5	2.63	2.45
3.5	12.6	5.4	2.60	2.40
4.5	16.2	6.5	2.58	2.34
5.5	19.8	7.2	2.57	2.31
6.5	23.4	8.0	2.55	2.27
7.5	27.0	8.8	2.54	2.24
10.0	36.0	10.8	2.52	2.16

TABLE 3.17: THE EFFECT OF THE THERMAL CAPACITY OF THE BUILDING ON THE SEASONAL PERFORMANCE OF THE HEAT PUMP

Switching differential of the room air thermostat (deg C)	Proportion of total heat load provided by supplementary heater $\frac{Q_b}{Q_c+Q_b}$ (%)	Cycle COP_h $\frac{Q_c}{W_c+W_f}$	Device COP_h $\frac{Q_c+Q_b}{W_c+W_f+W_b}$
0.1	3.8	2.67	2.15
0.2	4.4	2.64	2.46
0.3	5.1	2.61	2.41
0.4	5.5	2.60	2.39
0.5	6.5	2.58	2.34
0.6	6.9	2.57	2.32
0.8	8.0	2.55	2.27
1.0	9.3	2.53	2.22
1.2	10.4	2.53	2.18
1.4	12.0	2.50	2.12
1.6	12.5	2.50	2.10
2.0	14.8	2.48	2.03

TABLE 3.18: THE EFFECT OF THE SWITCHING DIFFERENTIAL OF THE ROOM AIR THERMOSTAT ON THE SEASONAL PERFORMANCE OF THE HEAT PUMP

Parameter	Base Case Value	Base Case Value Plus 10%	Effect on seasonal COP_h of the device, $\frac{Q_c+Q_b}{W_c+W_f+W_b}$ expressed as a percentage of the base case result
Radiator heat output rate at 55°C flow temperature (kW)	6	6.6	+4.8
Building heat demand at -1°C (kW)	6	6.6	-3.3
Thermal capacity of the heat distribution system (kJ/K)	329.8	362.7	+0.6
Thermal capacity of the building (MJ/K)	16.2	17.82	-0.6
Switching differential of room thermostat (deg C)	0.5	0.55	-0.4

TABLE 3.19: SENSITIVITY ANALYSIS ON THE MAJOR PARAMETERS INFLUENCING THE SEASONAL COP_h OF THE HEAT PUMP

CHAPTER 4 AIR SOURCE EVAPORATOR

1. INTRODUCTION

The function of the evaporator is to extract heat from the surrounding air. In order to minimize the irreversibilities the transfer of heat from the source to the working fluid must be achieved at a small temperature difference. This need to keep the temperature difference small, and the generally low coefficients of heat transfer for air under forced convection, require a significant enhancement of the air side surface area. The material and manufacturing costs involved in the construction of an air source evaporator result in this heat exchanger becoming the single most expensive component in the heat pump circuit.

The broad design objectives and requirements for the evaporator may be summarized as:

a) To provide the required heat transfer capacity at an acceptably low temperature difference.

b) Low manufacturing cost.

c) The ability to tolerate frost fouling effects.

d) Low noise levels associated with the movement of air through the heat exchanger.

The design of evaporator most widely employed on air source heat pumps is the tube-in-plate type cross-flow heat exchanger, borrowed from air conditioning applications. Copper tubes are assembled into a matrix of continuous plate fins, commonly made of aluminium. Multiple refrigerant side passes are achieved by soldering U-bends on the tubes where they protrude from the end plates. This arrangement produces a highly compact heat exchanger, with a small face area relative to the total heat transfer area.

The fan requirements are low volume flow with high pressure rise, and the evaporator may be operated in forced draught or induced draught mode. In air conditioning applications the fin spacing is typically 2 mm or less, but a wider spacing is necessary for heat pump duty because of frost blockage.

The design considerations to minimize the susceptibility to frost related difficulties of the tube-in-plate heat exchanger have been investigated by others. Stoecker showed that it is the increase in the air side pressure drop caused by the frost layer which has the greatest effect on the COP_h of the system (46). He measured both the change in overall heat transfer coefficient and the air side pressure drop as a function of the weight of the frost on the coil, and concluded that the increased pressure drop, in reducing the volumetric air flow rate through the evaporator, has a much more significant effect on the performance of the system than the increased thermal resistance of the ice layer. Indeed he found that, for light frost deposits, the overall heat transfer coefficient actually increased since the decrease in thermal conductivity was more than offset by the larger surface area and greater air velocity resulting from the reduction in flow area. These conclusions have subsequently been supported by others (47, 48). Stoecker went on to investigate the relationship between fin spacing and system performance. He found that if a wide finned coil and a close finned coil are selected to give the same evaporating temperature and duty at frost-free conditions, then the wide finned coil will produce a higher evaporating temperture at all conditions of frost growth and will therefore lead to the higher COP_h.

As the tube-in-plate evaporator is widely applied to air conditioning systems the manufacturing costs are low for quantities which merit the application of automated production machinery. Low volume orders, however, are likely to result in a high unit cost due to the labour in assembly. Furthermore, the specific adaptation necessary for heat pump duty, the increase in fin spacing, is also likely to increase the manufacturing cost.

This chapter presents the analysis and experimental results of an alternative design of evaporator, constructed from a wire wound heat transfer tubing and formed into a two row helical coil. Although the material cost is high, the simple geometry results in a low assembly cost. On the basis of quotations obtained from manufacturers of tube-in-plate evaporators, the proposed design is competitive at least for small batch production (49).

1.1 Heat Transfer Tubing

The tubing consists of a plain bore copper tube with a copper wire loop type fin soldered onto the outside surface of the tube, as shown in figure 4.1. The tubing is manufactured by Clayton Dewandre Ltd of Lincoln, and the product range offers choices in the tube diameter, loop height and loop density. The tube diameter was selected on the basis of the heat transfer considerations of the refrigerant side, principally the desire to induce a sufficiently high refrigerant velocity at all operating conditions to achieve annular flow of the refrigerant liquid, see section 2.3. The selection of fin height and density was based on the required enhancement of the air side surface area. This could be achieved either by combining a small fin outside diameter with a high fin density or vice versa; the actual selection was arbitrarily based on one of the manufacturer's more common configurations, consistent with the required surface enhancement. The dimensions and other pertinent data on the selected heat transfer tubing are summarized in table 4.1.

1.2 Evaporator Configuration

The configuration of the evaporator is illustrated in figure 4.2. The evaporator is formed by initially coiling a continuous length of the heat transfer tubing around a circular former of diameter equal to the required internal diameter of the coil. The second row is then simply coiled around the first row producing the staggered tube arrangement shown in the diagram. The tubes are held in position by soldering thin brass straps vertically down the inner and outer faces of the heat exchanger.

The result is a two row, helical coil heat exchanger. For experimental purposes two evaporators of this design were manufactured. The same heat transfer tubing, tube and row spacing, and coil outside diameter were used in each case. The total nominal tube lengths of the two test evaporators were 20 m and 30 m. The dimensions and data of the test evaporators are summarized in table 4.2. A slight difference in design between test evaporators is the relative lengths of the two rows. A greater proportion of the total length was deployed on the outer row of the 30 m evaporator than was the case in the 20 m version, because the test results of the smaller coil indicated that the heat transfer capacities of the two rows were not well matched. Clearly the temperature of the air impinging on the second (outer) row is lower than that arriving on the first row, thus a greater surface area is needed on the second row. While the design of the 20 m coil did allow an additional 12.1 per cent surface area on the outer row, this was found to be insufficient and so the 30 m coil was equipped with an outer row of 40.4 per cent greater length than the inner row. The extra tube length was accommodated in the design by coiling the lowest turn of the outer row in towards the centre of the coil in a spiral geometry, as illustrated by the shaded area of figure 4.2. This results in a flat base to the heat exchanger which benefits the positioning and fixing of the evaporator into the heat pump.

The air side configuration comprises a cross-flow arrangement with forced draught from a high volumetric flow propeller fan. The fan directs the air into the centre of the coil, with the result that the air is constrained to flow radially through the tube bundle of the evaporator. The refrigerant side has two circuits, i.e. each tube row provides a parallel refrigerant circuit, and the evaporator is operated in direct expansion mode with the refrigerant entering at the top of the coil and leaving at the bottom. The refrigerant flow is split into two equal mass flow rates by a venturi type distributor, the outlet of which is connected to evaporator inlets by small bore distributor tubing. A simple manifold arrangement rejoins the circuits at the evaporator outlet.

2. HEAT EXCHANGER ANALYSIS

2.1 Air Side Heat Transfer and Pressure Drop

The Briggs and Young correlation is commonly applied to staggered arrays of high finned tubes (50):

$$Nu = 0.134\ Re^{0.681}\ Pr^{1/3} \left(\frac{2w_2}{d_3-d_2} \right)^{0.2} \left(\frac{w_2}{d_4} \right)^{0.1134}$$

Here the air side Reynolds number is based on the maximum air velocity through the tube bank, that is the velocity which occurs at the minimum flow area. Depending on the geometry of the tube bundle, the minimum area can either be transverse to the airflow or on the diagonals between the tubes, see figure 4.3. Assuming the flow through the transverse area splits evenly between the two diagonal paths, the minimum flow area will occur on the diagonals when

$$2s_3 < s_1$$

otherwise the minimum area will be transverse to the flow entering the tube bank. For all the combinations of tube spacing of interest in the present design it was found that the minimum area occurs transverse to the flow. The mean face area of the coil is given by:

$$A_{f,m} = \frac{l\ s_1}{N_c}$$

The transverse area is given by:

$$A_{t,m} = \frac{l}{N_c} (s_1 - d_2 - B'_{a,f})$$

where $B'_{a,f}$ is the blockage to airflow in the transverse direction, per unit length of tube per row, caused by the loop fins, and is obtained from:

$$B'_{a,f} = \frac{2}{w_2}\ d_4 \left(\frac{d_3}{2} - \frac{d_2}{2} \right) \times 2$$

$$= \frac{2}{w_2}\ d_4\ (d_3 - d_2)$$

Hence the ratio of the evaporator mean face area to the minimum flow area is:

$$\frac{A_{f,m}}{A_{t,m}} = \frac{s_1}{s_1 - d_2 - B'_{a,f}}$$

The above equation is used to convert a known face velocity u_f into a maximum coil velocity u_t as follows:

$$u_t = u_f \frac{A_{f,m}}{A_{t,m}}$$

and the maximum mass flux rate can be found from:

$$\dot{m}''_a = \frac{\dot{m}_a}{A_{t,m}} = \rho_a \; u_t = \rho_a \; u_f \frac{A_{f,m}}{A_{t,m}}$$

When attention is focussed on a particular tube and fin design the Briggs and Young equation may be simplified by incorporating the two geometrical terms

$$\left(\frac{2 \; w_2}{d_3 - d_2} \right)^{0.2} , \quad \left(\frac{w_2}{d_4} \right)^{0.1134}$$

into the numerical constant, thus:

$$Nu = C \; Re^{0.681} \; Pr^{1/3}$$

The tube manufacturer supplied charts showing the measured air side heat transfer coefficient as a function of the face velocity for three tube spacings. The functional relationships in these charts were found to be in good agreement with the above correlation. The value of the constant C was found from the supplied data to be 0.2717. Hence the reduced Briggs and Young equation was used to determine the air side heat transfer coefficient for the tubing design of the experimental evaporators.

In order to determine the fan power requirement it is necessary to calculate the total air side pressure drop. The overall pressure drop has two components; the friction and blockage caused by the tube bundle and a ducting loss. Also supplied by the manufacturer were charts indicating the measured air side pressure drop as a function of the mean face velocity for three tube spacings and these data were fitted to the general equation:

$$\Delta P_{a1} = f\, N_c \frac{\dot{m}''_a{}^2}{2\, \rho_a}$$

where the friction factor f is a function of the tube and bundle geometry and the inter-tube Reynolds number (51);

$$f = \frac{A}{Re^{0.3}}$$

where A is a numerical constant.

Thus

$$\Delta P_{a1} = N_c\, B \frac{(\dot{m}''_a)^{1.7}}{2\, \rho_a}$$

where B is constant for small changes in air viscosity, found to be 1.989 from the data provided.

The frictional losses associated with flow through the duct consist of losses from the inlet and discharge grilles, a flow contraction into the fan bellmouth, an expansion into the evaporator intake as well as two changes of direction. The pressure loss coefficients for each of these components were calculated from the techniques and empirical relations presented by ESDU (52). For convenience the individual coefficients were expressed in terms of the total volumetric air flow rate and totalled to provide an estimate of the total duct losses:

$$\Delta P_{a2} = K_d\, \dot{V}_a^2$$

where K_d was found by calculation to be 37.5 kg/m^7

An additional consideration of the air side heat transfer is the calculation of the surface effectiveness of the densely finned air side area. The surface effectiveness is defined as:

$$\eta_a = 1 - \frac{A_s'}{A_o'} (1 - \eta_f)$$

where A_s' is the surface area of the fins per unit length of tube and η_f is the fin efficiency.

From the geometry of the selected tube

$$A_s' = 0.4959 \ m^2/m$$
$$A_o' = 0.5358 \ m^2/m$$

The fin efficiency is calculated by assuming a plain cylindrical fin (53):

$$\eta_f = \frac{\tanh \ (\beta m)}{\beta m}$$

where the fin height $m = (d_3 - d_2)/2$

and $\beta = \left(\frac{h_o \ p}{k_f \ A_x} \right)^{0.5}$

where the fin perimeter $p = \pi d_4$

and the fin cross-sectional area $A_x = \frac{\pi d_4^2}{4}$

and h_o is the air side heat transfer coefficient, k_f is the thermal conductivity of the fin material.

For an air face velocity of 1.5 m/s the fin efficiency is calculated to be 0.9257, which results in a surface effectiveness of 0.9312.

2.2 Refrigerant Side Pressure Drop

The frictional pressure drop of the refrigerant flow through the evaporator needs to be estimated in order to evaluate the degree of exit superheat. Martinelli and Nelson developed a predictive method for two phase flow pressure drop evaluation in which the total pressure drop is considered to be the sum of two components (54). The first component, ΔP_{e1}, is the frictional loss, which may be determined by first calculating the frictional pressure drop which would occur if the pipe were full of the liquid phase only, and multiplying this by a two phase correction factor which depends on the fluid, its quality and the mean pressure. The second component, ΔP_{e2}, is the pressure drop caused by the momentum change of the two phase mixture as the liquid vapourises, and is found from:

$$\Delta P_{e2} = (\dot{m}_r'')^2 (v_1 - v_4)$$

Martinelli and Nelson developed their method for two phase water mixtures. Altman et al applied the method to R22 evaporating in horizontal tubes, with good agreement between the predicted and measured pressure drop (55). They present the two phase correction factor for R22 in graphical form against the mean pressure with the exit vapour quality as a parameter. This may be expressed algebraically for an outlet quality of unity and for the suction pressures relevant to this present study as:

$$R_p = \frac{\Delta P_{e1}}{\Delta P_l} = e^{5.693 - 1.223 \ln (P_e)}$$

The frictional pressure drop of single phase liquid flow through smooth pipes may be estimated from:

$$\Delta P_l = \frac{4 f \rho_l u^2 l_{max}}{d_1}$$

where l_{max} = length of the longest circuit through the evaporator

$$f = 0.079\,(Re_1)^{-0.25}, \text{ i.e. Blasius's formula for smooth tubes}$$

$$Re_1 = \frac{\dot{m}_r''\, d_1}{\mu_1}$$

By substituting for u and incorporating the expression for the friction factor, the equation for the single phase pressure drop becomes:

$$\Delta P_1 = \frac{2.528\, Re_1^{\,-0.25}\, \dot{m}_r^{\,2}\, l_{max}}{\pi^2\, \rho_1\, d_1^{\,5}}$$

Hence the total pressure drop is calculated from:

$$\Delta P_e = \Delta P_{e1} + \Delta P_{e2}$$

where $\Delta P_{e1} = R_p\, \Delta P_1$

The evaporator outlet pressure is related to the mean pressure by:

$$P_1 = P_e - \frac{\Delta P_e}{2}$$

2.3 Refrigerant Side Heat Transfer

There have been numerous attempts to correlate the internal heat transfer coefficient for forced convection evaporation of refrigerants inside a tube. An informative review of the significant earlier correlations is given by Anderson et al (56). The problem is complicated by the existence of several flow regimes, see figure 4.4, taken from ASHRAE (30). Figure 4.4 b) is representative of the conditions pertaining to a heat pump evaporator controlled by a thermostatic expansion valve, for which the inlet quality is in the range 0.2 and 0.4 and the exit superheat is 2 to 5 deg C. For this application three heat transfer mechanisms are likely to be important:

a) Two phase forced convective heat transfer which commences at low to medium qualities when the vapour flow rate becomes significant, and continues into the dry out region.

b) Post dry out heat transfer, characterized by dry wall heat transfer to a mist flow.

c) Single phase forced convection heat tranfer to the vapour flow.

Two phase forced convection heat transfer is characterized by annular liquid flow in contact with the tube wall, and vapour with some entrained liquid droplets flowing through the centre of the tube. Evaporation takes place from the liquid/vapour interface. High heat transfer coefficients, between 1000 and 10000 W/m^2K are produced. In a horizontal tube evaporator a high refrigerant mass flux is necessary to ensure the complete wetting of the circumference of the tube, otherwise stratification effects become important and the upper surfaces of the tube may become dry. Since dry out occurs prior to complete evaporation, the remaining liquid is entrained in droplet form within the bulk vapour flow. The heat transfer coefficient is considerably lower in the post dry out region due to the absence of the liquid film. Towards the exit of the evaporator the remaining liquid vaporizes and true superheating commences. Reference 57 provides further discussion of the fundamental mechanisms associated with boiling heat transfer, and Hughes et al report two phase flow patterns for R12 in a horizontal tube, temperature difference controlled evaporator (58).

Ideally a calculation method to determine the heat transfer rate to the refrigerant would consider all three mechanisms. However, a substantial difficulty arises in the determination of the point of dry out for a horizontal tube heat exchanger. Investigations into dry out of vertical tube evaporators have highlighted a number of different dry out mechanisms and have met with some success in the prediction of dry out in high heat flux applications. A review of the important correlations is given by Turner (59). However, it is unwise to apply the methods developed for vertical pipes to horizontal ones, due to the assymmetry resulting from the effect of gravity. The method of Shah can be used to predict a single mean heat transfer coefficient for the annular and mist flow regions, and the coefficient for the pure vapour flow can be determined separately (60). Shah's method is based on the determination of the two phase heat transfer enhancement factor Ψ defined as:

$$\psi = \frac{h_{i,1}}{h_1}$$

where $h_{i,1}$ is the two phase coefficient and h_1 is the liquid phase coefficient determined from the Dittus-Boelter equation:

$$Nu_1 = 0.023\ Re_1^{\ 0.8}\ Pr_1^{\ 0.4}$$

$$\text{where } Re_1 = \dot{m}_r''\ \frac{(1 - x)}{\mu_1}\ d_1$$

The enhancement factor is determined from three dimensionless groups:

$$\text{Convection number} \quad Co = \left(\frac{1 - x}{x}\right)^{0.8} \left(\frac{\rho_g}{\rho_1}\right)^{0.5}$$

$$\text{Boiling number} \quad Bo = \frac{\dot{q}''}{\dot{m}''\ (h_1' - h_4')}$$

$$\text{Froude number} \quad Fr_1 = \frac{m''^2}{\rho_1^{\ 2}\ g\ d_1}$$

The Froude number accounts for stratification effects in horizontal two phase flow. If $Fr_1 > 0.04$ this indicates that the entire circumference is wetted, in which case Fr_1 is not subsequently used in the calculation. The Boiling number accounts for the enhancement of heat transfer attributable to bubble nucleation in the nucleate boiling regime. As described above, due to the high vapour content at the inlet to the evaporator, nucleate boiling is not a significant mechanism in the heat pump evaporator. Shah presents a graphical method to determine given Co, Bo and Fr_1. However, for $Fr_1 > 0.04$ and in the absence of bubble nucleation, the method simplifies to:

$$\psi = 1.8\ Co^{-0.8} \quad (Fr_1 > 0.04,\ Co < 1.0)$$

These two conditions were found to be satisfied across the full operating range of the evaporator designs under consideration. Hence the above equation was used to determine the local two phase heat transfer coefficient for the evaporator. An estimate of the mean, inside coefficient for the entire two phase region is obtained by applying the method several times for different vapour qualities, and then taking the average. For example, if the inlet quality is 0.3 and the end of the two phase region is assumed to be defined by a quality of 1.0, calculate the local coefficient for x = 0.3, 0.4, 0.5 etc. and assume the effective value is determined by the arithmetic mean of the local coefficients. Shah reports that this method is also applicable to the post dry out region, which may extend from x = 0.9 to 1.0, provided the overall change in vapour quality in the evaporator is large. Thus in the absence of a recognised method for predicting the incidence of dry out in horizontal tubes, Shah's method as described above is applied for qualities up to 1.0.

The heat transfer coefficient in the superheat region is calculated from the Dittus-Beolter equation for single phase vapour flow:

$$Nu_v = 0.023 \; Re_v^{\,0.8} \, Pr_v^{\,0.4}$$

The presence of lubricating oil in the evaporator may well modify the refrigerant flow patterns and heat transfer coefficient, though such effects are as yet not clearly understood. Green and Furse reported increases in the average boiling coefficient for R12 with oil concentrations up to 8 per cent (61). It was suggested that the oil was behaving as a wetting agent and thus improving the liquid film to tube surface contact, as well as modifying the properties of the mixture. Stephan investigated R12 and R22 boiling on a horizontal plate and concluded that below 3 per cent oil concentration the heat transfer coefficient increased slightly, particularly at low evaporating temperatures (62). The coefficient decreased at higher oil concentrations. Hughes et al observed a decrease in the local heat transfer coefficient in the fully developed annular flow regime of a horizontal tube evaporator for a 6 per cent naphthenic oil/R12 mixture (58). They also measured an increase in refrigerant side pressure drop with increasing oil concentration. For the small oil concentrations typical of a reciprocating compressor/R22 combination it is unlikely that the presence of oil will cause a major change to the average heat transfer coefficient, hence the present analysis adopts the simplification of considering the working fluid to be a single component refrigerant.

2.4 Evaporator Model and Solution Procedure

The evaporator model is a component of the vapour compression cycle model, VCCM, described in Chapter 3. The function of the evaporator model is to calculate a revised estimate of the mean evaporating temperature, given an initial estimate of this and the heat exchanger geometry and fluid flow details. The revised estimate is returned to the main program and the iteration is continued until both the condensing and evaporating temperatures converge.

The prediction method developed in this study is described below. The evaporator is considered to be two heat exchangers in series, a true evaporator connected to a vapour superheater. The method calculates the air side heat transfer coefficient and the mean refrigerant side coefficients for the two heat exchangers. The area required for superheating is then determined, and from the area remaining for evaporation, a revised estimate of the evaporating temperature is obtained. The actual exit superheat is then compared with the fixed value determined by the thermostatic expansion valve, and unless close agreement (within 0.1 deg C) is achieved, the evaporator calculations are repeated with the revised evaporating temperature. This process continues until the appropriate exit superheat is predicted by the model. The latest T_e value is then returned to the main vapour compression cycle program.

The evaporator model and the solution procedure are described in greater detail by the following steps. A flow chart for the evaporator model is included in illustration of the complete vapour compression cycle model, figure 3.1.

Known : Evaporator geometrical details; d_1, d_2, d_3, d_4, w_1, w_2, N_1, s_1, s_2, l, l_{max}, N_c, A_i', A_o'/A_i', η_a
Air side conditions, $\dot{V}_a$, T_{a1}
Superheat requirement, ΔT_{sh}

Given : Estimated refrigerant conditions; T_e, $\dot{m}_r$, h_4

To find : Revised estimate of T_e

Step 1 : Determine evaporator outlet conditions

$$P_1 = P_e - \frac{\Delta P_e}{2}$$

where
$$\Delta P_e = R_p \, \Delta P_1 + \Delta P_{e2}$$

$$R_p = e^{5.693 - 1.223 \ln (P_e)}$$

$$\Delta P_1 = \frac{2.528 \, Re_1^{-0.25} \, \dot{m}_r^2 \, l_{max}}{\pi^2 \, \rho_1 \, d_1^{\,5}}$$

$$\Delta P_{e2} = (\dot{m}_r'')^2 \, (v_1 - v_4)$$

$T_1 = \Delta T_{sh}$ + (saturated temperature corresponding to P_1)

Step 2 : Calculate the heat transfer rates in the evaporator and superheater regions from the refrigerant conditions.

evaporator $\dot{q}_{e,1} = \dot{m}_r \, (h_1' - h_4)$
superheater $\dot{q}_{e,2} = \dot{m}_r \, (h_1 - h_1')$

Step 3 : Determine the air side heat transfer coefficient, pressure drop and mean exit temperature.

$$h_o = 0.2717 \, \frac{k_a}{d_2} \, Re^{0.681} \, Pr^{1/3}$$

$$\Delta P_a = \frac{0.9945 \, N_c \, (\dot{m}_a'')^{1.7}}{\rho_a} + 37.5 \, \dot{V}_a^2$$

$$T_{a2} = T_{a1} - \frac{(\dot{q}_{e,1} + \dot{q}_{e,2})}{\rho_a \, \dot{V}_a \, Cp_a}$$

Step 4 : Estimate the area required for superheating.

$$A_{o,2} = \frac{\dot{q}_{e,2}}{U_{o,2}\,\Delta T_{LM,2}}$$

where $$U_{o,2} = \left(\frac{1}{\eta_a h_o} + \frac{1}{h_{i,2}} \times \frac{A_o}{A_i} \right)^{-1}$$

$$h_{i,2} = 0.023 \frac{k_v}{d_1} Re_v^{0.8} \; Pr_v^{0.4}$$

$$\Delta T_{LM,2} = \frac{(T_{a1} - T_1) - (T_{a2} - T_e)}{\ln \left(\frac{T_{a1} - T_1}{T_{a2} - T_e} \right)}$$

i.e. approximating the cross flow superheater to a counterflow heat exchanger.

Step 5 : Calculate the area available for evaporation.

$$A_{o,1} = A_o - A_{o,2}$$

Step 6 : Estimate the mean two phase heat transfer coefficient and the overall coefficient for the true evaporator. Calculate the mean coefficient from ten sample values of the local coefficient.

$$h_{i,1} = \frac{1}{10} \sum_{n=1}^{10} \quad (h_{i,1} \text{ is a function of } x_n)$$

where x varies systematically from x_4 to 1.0 and the functional relationship for $h_{i,1}$ is given by:

$$h_{i,1} = 0.0414 \frac{k_l}{d_1} \; Co^{-0.8} \; Re_l^{0.8} \; Pr_l^{0.4}$$

where $$Co = \left(\frac{1 - x}{x} \right)^{0.8} \left(\frac{\rho_v}{\rho_l} \right)^{0.5}$$

$$Re_l = \frac{(1 - x)\, m_r''\, d_l}{\mu_l}$$

The overall coefficient is found from:

$$U_{o,1} = \left(\frac{1}{\eta_a\, h_o} + \frac{1}{h_{i,1}} \times \frac{A_o}{A_i} \right)^{-1}$$

Step 7 : Refine the estimate of T_e using the Number of Transfer Units/Effectiveness method:

$$T_e = T_{a1} - \frac{(T_{a1} - T_{a2})}{E}$$

where $E = 1 - e^{-NTU}$

$$NTU = \frac{U_{o,1}\, A_{o,1}}{\rho_a\, \dot{V}_a\, Cp_a}$$

Step 8 : Determine the actual exit superheat, based on the revised evaporating temperature:

$$P_1 = (\text{saturated pressure corresponding to } T_1) - \frac{\Delta P_e}{2}$$

$$\Delta T_{sh} = T_1 - (\text{saturated temperature corresponding to } P_1)$$

If the actual superheat differs from the prescribed value by more than 0.1 deg C then return to step 1. Once an acceptable T_{sh} is achieved then return the latest estimate of T_e to the main program. When successive calls to the evaporator model result in changes of the estimate of T_e by less than 0.01 deg C then the evaporator model has converged. The fan motor power requirement is found from:

$$\dot{W}_f = \frac{\dot{V}_a\, \Delta P_a}{\eta_{fm}}$$

where η_{fm} is the combined efficiency of the fan and fan motor.

An estimate of the surface temperature of the evaporator coil is found from:

$$T_s = T_e + \frac{\dot{q}_{e,1}}{h_{i,1}\ A_{i,1}}$$

2.5 Validation

The predictive accuracy of the evaporator model was assessed by comparison with experimental results. For this validation exercise the vapour compression model was operated in a reduced form; the condenser model was not included, requiring the saturated condensing temperature to be supplied as an input to the program. Therefore the predictions refer to the calculated equilibrium conditions between the evaporator and the compressor.
The predictions were compared against measured values for the following circumstances:

a) The 30 m evaporator and prototype fan (designated fan B) at fixed speed across a range of air temperatures.

b) The 30 m evaporator at constant air temperature with fan A operated at a range of speeds.

c) The 20 m evaporator in similar circumstances to b).

Full details of the experimental fans are given later in this chapter.

The results are summarized in table 4.3. At constant fan speed the model tends to slightly overestimate the overall heat transfer coefficient, leading to higher mean evaporating temperatures than were measured. This may be primarily due to the assumption of wet wall heat transfer up to unity quality, which would tend to overestimate the inside heat transfer coefficient compared to the more realistic premise that dry out occurs prior to complete evaporation. The resulting error,

however, is acceptably small. The model appears to underestimate the heat transfer rate across the evaporator. This is because of the assumption of negligible liquid subcooling prior to expansion. In practice some heat is lost from the refrigerant liquid as it is stored in the receiver and flows through the pipework, causing some subcooling. The result is a decrease in the specific enthalpy of the refrigerant entering the evaporator, and thus an increase in the enthalpy change and heat transfer rate across the evaporator. Measurements of the refrigerant temperature prior to expansion indicate 2 to 3.7 deg C of subcooling. The accuracy of prediction of refrigerant mass flow rate is high for tests a). The exact agreement between measured and predicted fan power arises because the fan motor efficiency constant was calculated for the prototype fan at its rated speed of 900 rev/min.
In comparing predicted with measured results for the variable fan speed tests it is apparent that the prediction accuracy diminishes at low air flow rates. This is probably due to the air side heat transfer coefficient departing from the Briggs and Young correlation at low velocities. Acceptable accuracy in heat transfer rate, mass flow rate and evaporating temperature is exhibited for air flow rates of 0.9 m^3/s and above for both evaporators. The poor agreement on fan motor power is caused by two factors. Firstly, the fan applied to the test has a different motor and propeller design from the prototype fan which is modelled. Secondly, the model assumes constant motor efficiency across the speed range, whereas it is likely that the actual efficiency will reduce at low speeds and increase at high speeds.

To summarize the results of the validation exercise; heat transfer rate is predicted to $\pm$ 4 per cent and evaporating temperature to better than $\pm$ 2 deg C in all the situations explored except for air flow rates less than 0.9 m^3/s.

2.6 Heat Pump Performance as a Function of Evaporator Size

The evaporator model was used to determine the relationship between evaporator size and the performance of the heat pump and to indicate the range of sizes which produce an acceptable payback period. The input data to the model are:

a) Operating Conditions

ambient air temperature	4°C
water return temperature	49°C
water flow rate	1.0 m^3/s
evaporator outlet superheat	4.0 degC

b) Heat Pump Components

compressor	Maneurop MT 32 JF5
condenser	YIA CK8-20
evaporator	2 row, 2 circuit helical coil

The total tube length of the evaporator was varied from 10 m to 50 m in steps of 10 m, and for each length the air volumetric flow rate was varied. The results of COP_h against mean air velocity for each evaporator size are shown in figure 4.5. Clearly there is an optimum air velocity, at which the COP_h is a maximum, for each size of evaporator,and as the coil size increases the optimum velocity falls. The maximum in COP_h arises because of the rapid rise in significance of the fan power compared with the compressor power as the velocity increases. Below the optimum velocity an increase in velocity results in a greater increase in heat pump output than total power consumption. Above the optimum velocity, the increase in fan power caused by the increase in velocity more than offsets the improvement in the cycle performance and a drop in COP_h results.

Figure 4.6 shows the maximum COP_h results plotted against evaporator tube length, where the mean face velocity is the optimum value for each evaporator size. The figure also shows the heat pump heat output rate available for the specified conditions. Clearly the rate of increase in COP_h falls with increasing size, suggesting that successive increases in evaporator length are progressively poorer "value for money". To demonstrate this the following exercise calculates the payback period for increases in the length of the evaporator above the 10 m size. Assuming that the increase in capital cost of the heat pump to the customer is simply the cost of the extra evaporator tubing, and taking the COP_h results obtained for the above conditions to be roughly indicative of the annual performance, the payback periods for the sizes

20 to 50 m are listed in table 4.4. This analysis ignores the effect of supplementary heating and so does not credit the improved heat output rates from the larger evaporators. The results indicate excessively long payback times for size increases to 40 m and greater.

2.7 Evaporator Performance Characteristics

To investigate the performance characteristics of the evaporator against operating conditions the model was operated for the 30 m evaporator with an air flow rate of 1.25 m^3/s for air temperatures in the range -5°C to 15°C and return water temperatures 30°C to 60°C. The remaining input data are the same as those listed in section 2.6. The results are presented in detail in table 4.5.

Varying the water return temperature and thus altering the condensing conditions affects the performance of the evaporator in two ways. Firstly, changing the condensing temperature changes the pressure ratio across the compressor and thus alters the mass throughput of the refrigerant vapour. Changing the refrigerant mass flow rate affects the evaporator two phase and superheating heat transfer coefficients, as shown in the table. Secondly, the higher the condensing pressure, the higher the mixture quality at the inlet to the evaporator, which leads to a lowering in evaporator performance because a greater proportion of the total heat transfer rate must occur in the superheating region. These two effects combine to produce a noticeable reduction in evaporator performance at high water return temperatures. This is illustrated in figure 4.7, in which the evaporator heat transfer rate per unit temperature difference, and the proportion of the total area devoted to superheating, are plotted against the air and water temperatures. At an air temperature of 0°C, $\dot{q}_e/(T_{a1} - T_e)$ varies from 714 W/deg C at a return temperature of 30°C to 650 W/deg C at 60°C, and the area fraction for superheating varies from 9.8% to 17.2% for the same change in water temperature.

Variation in air temperature affects the evaporator performance by changing the pressure ratio across the compressor and the density of the suction vapour, the combined effects of which are significant changes to the refrigerant mass flow rate. Thus as the air temperature rises, the refrigerant mass flow rate and thus the evaporator internal heat transfer coefficients increase, leading to an improvement in the heat transfer rate per unit temperature difference.

The estimated evaporator surface temperature is also plotted in figure 4.7. This temperature is important in relation to the rate of frost accumulation on the evaporator coil. The figure indicates that above an air temperature of 6°C the surface temperature is greater than 0°C, hence frost will not be formed. At an air temperature of 5°C, frost could be formed depending on the condenser water return temperature. Low water temperatures tend to promote the formation of frost on the evaporator surface.

3. EXPERIMENTAL APPARATUS

An experimental test facility was developed to evaluate the performance of the evaporator design. The facility comprises three components; the Evaporator Test rig, the Environmental Chamber and the data acquisition system.

3.1 Evaporator Test Rig (ETR)

This is a fully instrumented heat pump and heat sink system, designed to accommodate a range of evaporator and fan sizes. The refrigerant and heat sink circuits are shown schematically in figure 4.8. The major components incorporated in the refrigerant circuit, the compressor, receiver, filter, thermostatic expansion value (TEV), suction accumulator, and service and solenoid valves are identical to those employed on the prototype heat pump and are described in Chapter 6. An exception is the water cooled condenser; the ETR condenser is a coaxial tube heat exchanger manufacturered by Wieland, model KWG 3X. This condenser is the subject of performance testing in the next chapter. Refrigerant pipework sizing also reflects the design of the prototype heat pump. The outside surfaces of the compressor, condenser, receiver, filter and interconnecting pipework were thermally insulated. The circuit contains three sight glasses located at the condenser outlet, the receiver outlet and in the suction line.

The ETR heat pump system is contained within a three tier free standing framework, as shown in figure 4.9. The lower compartment houses the compressor, condenser and most of the other circuit components. The central compartment contains the specimen evaporator. The height of the central compartment is adjustable to suit a range of evaporator sizes. The upper compartment houses the propeller fan. The sides of the central and upper compartments are covered by an aluminium grille. The air intake is through the side grilles of the fan compartment, the air is then directed axially into the evaporator housing and is forced radially across the tube bundle and through the discharge grilles. The fan motor electrical supply contains a motor speed controller so that the fan shaft speed and hence the air volumetric flow rate can be adjusted. The photographs in figure 4.9 were taken before the thermal insulation was added, and they show the 20 m experimental evaporator in the rig.

The heat sink system is designed to supply the condenser with water at a stable temperature and flow rate. It consists of three water circuits. The primary circuit is a pumped flow from the header tank, through the condenser and flow meter and return to tank. The flow rate is adjustable by means of a manually operated regulating valve. A branch and valve network is installed downstream of the flow meter to permit the gravimetric calibration of the flow rate transducer. The secondary circuit removes heat from the header tank by rejecting it to a water to water shell and tube heat exchanger. The cooling capacity of the secondary circuit is regulated by a temperature control system based on the temperature of the water entering the condenser, $T_{control}$, as detected by a resistance thermometer. The thermometer is connected to an electronic three term controller which produces an output signal related to the difference between the measured and desired temperatures. This signal governs the position of an electrically operated three way diverter valve installed in the secondary circuit which controls the proportion of the flow which bypasses the heat exchanger. Thus the cooling capacity is automatically varied from zero to the maximum available, depending on $T_{control}$. The third circuit is the site cooling water supply which removes heat from the shell side of the heat exchanger, and ultimately rejects it to the atmosphere in the site cooling towers. In order to rapidly preheat the heat sink circuit at the start of an experiment, the header tank is equipped with four 3 kW immersion heaters.

3.2 Environmental Chamber

The Environmental Chamber is designed to provide controlled air conditions for evaporator testing. It consists of a large, insulated, walk-in chamber and air conditioning equipment to the following specification:

Dry bulb temperature	-15°C to +50°C
Temperature stability	±0.5°C
Relative humidity	10 to 95%
RH stability	±2%
Maximum heat load	±10 kW
Working volume	3 x 3 x 2.5 m high

As the Environmental Chamber represented a major capital investment for the Company, the specification was deliberately widened beyond the immediate requirements for evaporator tests, in order to provide a general purpose environmental test facility.

The specification satisfies the proposed recommendations for facilities to be used in the rating of air to water heat pumps from the British Air Conditioning Approvals Board (63). The air conditioning system is shown in schematic form in figure 4.10. It consists of a direct expansion air cooler and circulation fan positioned inside the chamber, an electrical re-heater battery, an electrically powered steam generating humidifier and air flow distribution grille. Most of the process air is recirculated through these components. However a small volume of fresh air is taken from outside, chilled and chemically dried and then mixed with the main process air. Air temperature and relative humidity are sensed at the point of entry to the controlled volume. The capacities of the main cooler, re-heater, humdifier and drier can be modulated, and are controlled by two independent electronic three term controllers acting on the measured air temperature and relative humidity. Two views of the chamber are shown in figure 4.11.

3.3 Instrumentation and Data Acquisition

The ETR refrigerant circuit contained pressure, temperature and rate of flow transducers, as shown in figure 4.8. Refrigerant temperature and pressure were measured at four circuit positions in order to define the thermodynamic cycle. The measurement positions were the suction line immediately downstream of the evaporator manifold (P_1, T_1), the condenser inlet (P_2, T_2), immediately upstream of the expansion valve (P_3, T_3) and the inlet to the distributor (P_4, T_4). The instrumentation coverage was increased for tests on the 30 m evaporator with the installation of a fifth state point measurement (P_5, T_{5B}) at the inlet to the outer row of the evaporator, and by additional temperature measurement at the inlet and output of each evaporator circuit (T_{5A}, T_{6A}, T_{6B}). In all cases the thermometers selected for these precision measurements were four wire, stainless steel sheathed, platinum resistance thermometers (PRT) to BS 1908 : 1984 class A (64). The measuring section of each thermometer stem was completely immersed in the refrigerant flow, and the thermometers were compression sealed into the circuit. The pressure measurement used strain gauge type transducers which when excited by a stable 10 V dc source developed a voltage output proportional to the applied pressure.

The local barometric pressure was measured during each experiment and used to determine the absolute pressures of the refrigerant. Additional temperature measurements (T_8 - T_{12}) were taken to provide a check on the accuracy of principal thermometers. These were made with surface mounted class B PRT's clamped to the pipework at the positions shown in figure 4.8 and externally insulated.

A pelton wheel type flow meter was installed in the liquid line to meter the refrigerant volumetric flow rate. However, difficulty was experienced with this transducer; throughout the experiments on the 20 m evaporator the indicated reading was significantly lower than was expected, and during the tests on the 30 m evaporator the output gradually fell to zero. The pressure of time prevented an investigation into the transducer malfunction, and an alternative method of determining the refrigerant flow rate was adopted, based on energy balance considerations across the condenser, as discussed in section

4.3.

The temperature of the air flowing across the evaporator was measured at the intake and discharge position (T_{a1} and T_{a2}) by means of PRT averaging bulbs which determine the mean air temperature across the four sides of the test rig. The siting of the air discharge thermometer was a difficulty as it was not possible to prevent some direct contact between the bulb and the surface of the evaporator. There is evidence in the test results to suggest that this measurement is depressed below the true air discharge temperature, because of this contact. Thus, for the presentation of results, the discharge temperature was calculated from the measured intake temperature, air volume flow rate and evaporator heat transfer rate, as explained in section 4.3.2.

The compressor motor power consumption was measured by a wattmeter transducer installed in the motor electrical supply. The fan motor power consumption was similarly measured by a portable digital wattammeter instrument which could be connected into the motor supply as required. The rotational speed of the fan was measured by a non-contact optical tachometer. Air velocity and air volumetric flow rate were determined using a calibrated hot wire anemometer. The air volumetric flow rate was obtained from a series of velocity measurements at the centre of 50 mm x 50 mm cells marked on the air intake grille. The flow rate is calculated from integrating the cell velocity results across the air intake area.

The outputs from the environmental chamber instrumentation system, the air temperature (T_a) and relative humidity (Rh) at the point of entry to the controlled environment, were recorded.

The heat sink water flow rate is measured by a pelton wheel type flow meter. The water temperature is sensed at the condenser inlet (T_{w1}) and outlet (T_{w2}) by a matched pair of insertion PRT's to tolerance class A. A calibration exercise was carried out to determine the likely thermometry error in the temperature difference (T_{w2} - T_{w1}). The thermometers were immersed in a well stirred water bath and the difference in resistance between the two was measured at five different water bath temperatures covering the operating range. The results were formed into a correction table which was used to adjust the measured temperature differences for known offset errors.

This instrumentation was monitored by a microcomputer based data acquisition system. Two Hewlett Packard HP3421A data acquisition units, each configured to accept 10 resistance thermometers and 10 voltage measurements, were connected via the HP-IB parallel interface to a HP86B micro computer. The system also included a dual disk drive unit and a line printer. This data logging package contains the same hardware and employs a similar operating method as that used in the prototype field trial monitoring exercise and the detailed description is given in Appendix A.

With the exceptions of the fan power, fan speed, air velocity and barometric pressure measurements, all the instrumentation outlined above were connected to the computer based logging system. The computer program to operate the data loggers was written specifically for this application and contains functions to convert the transducer signals to engineering quantities by applying the calibration equations described later, to calculate the heat transfer rates and COP_h from the measured data and to display, print and record the data. Typically all the tranducers were scanned every 2 minutes, although this interval could be adjusted as required. For instance, when steady-state conditions were achieved, the system performed 10 complete scans at 15s intervals and printed out the average values of the 10 readings.

The final component in the instrumentation and data acquisition system was a two pen chart recorder which recorded the instantaneous output of two pressure transducers (pressures P_1 and P_2). This provided a convenient method of indentifying steady-state conditions and of following rapid, transient events.

4. EXPERIMENTAL METHODS AND DATA PROCESSING

4.1 Experimental Methods

The desired air temperature and relative humidity conditions were entered into the environmental chamber controllers and the air conditioning system was started. The set point for the water entry temperature to the condenser was entered into the heat sink temperature controller and the primary and secondary pumps were started. If required the heat sink immersion heaters were energized to heat the water rapidly. The required fan speed was set on the motor speed controller and after a delay of about 30 minutes the ETR fan and compressor were started and the data acquisition system was activated. The heat sink water flow rate was checked and adjusted as necessary. When the heat sink temperature approached the set point the immersion heaters were switched off.

Typically an elapsed time of 2 to 4 hours was required for equilibrium conditions to be established for the air condition, heat sink temperature and the heat pump system. During the settling period the transducers were regularly logged at 2 minute intervals. Once steady state conditions were established the acquisition system logged the transducers ten times in rapid succession and the average values were recorded on disk and printed out. The last set of instantaneous readings were also printed so that any movement in the data during the averaging process could be detected. Any supplementary measurements; fan speed, air velocity etc were manually recorded at this time.

The procedure for an experiment of a transient nature such as a frosting trial differed from the above. The start up of the ETR fan and compressor were delayed until the set points of air temperature, relative humidity and heat sink water temperature had been achieved and these conditions were steady. The acquisition system was set to print the instantaneous results at 60s intervals throughout the experiment and the data averaging function was not used.

4.2 Summary of Evaporator Test Programme

A summary of the evaporator experimental programme is presented as table 4.6. Three major items of hardware were varied in the programme:

a) Evaporator size. Two sizes of evaporator were tested, designated by the nominal tube lengths of 20 m and 30 m. The design details are given in section 1.2.

b) Fan design. Three designs of fan were investigated, all of which were manufactured by Woods of Colchester Ltd:

A 450 mm diameter, 4 bladed propeller powered by a D71 speed controlled motor with a maximum speed of 1350 rev/min. This fan is from the GP range of propeller fans made by Woods (65). The fan characteristic presented in the catalogue is shown in figure 4.12 a).

B 450 mm diameter, 5 bladed propeller powered by a specially wound 900 rev/min motor. This fan is a prototype version of the Group 1 series design which will supercede the GP range. The preliminary performance characteristic is shown in figure 4.12 b) (66).

C 500 mm diameter, 4 bladed propeller powered by a F1644 speed controlled motor with a maximum speed of 900 rev/min. Similarly to A, this fan is taken from the GP range. The catalogue performance characteristic is shown in figure 4.12 c).

All three fans were fitted with a bellmouth mounting. Table 4.7 summarises the fan designation system.

c) Distributor tubing. The small bore tubing linking the distributor to the individual evaporator circuits is designed to introduce a frictional pressure loss in each circuit which is large compared with any likely difference in pressure drop between the two evaporator circuits, in order to promote an equal split of the refrigerant flow. The bore diameter and length of this tubing were adjusted in order to produce a satisfactory compromise between distribution equality and flow resistance to pure vapour flow when the heat pump is in hot gas bypass defrost mode. Details of the distributor tubes are given in table 4.8. The choice of distributor tubing was found to have negligible effect on evaporator performance, but for reference purposes the details of the tubing used in each experiment are summarised in table 4.6.

The test conditions are also listed in table 4.6. For the basic comparison of the performance of the two evaporator sizes, tests A and B, and for test C the relative humidity was kept low in order to suppress condensation and frosting effects. The investigation into frost fouling effects was confined to the 30 m evaporator and involved a systematic examination of the influence of air temperature (test D), of condenser water temperature (test E), of fan characteristic (test F) and of relative humidity (test G).

4.3 Data Processing

4.3.1 Conversion of the Transducer Signals

The digital processing capability of the computer was employed to convert the measured transducer signals to engineering quantities by means of the calibration equations and constants listed in table 4.9. The right hand side of the equations is the transducer output signal, which is either a dc voltage or, for all temperature measurements, an electrical resistance value. The constants in the equations are determined by direct calibration (e.g. pressure and rate of flow), by reference to the supplied transducer characteristics (e.g. compressor power) or by reference to an appropriate British Standard (e.g. temperature measurement).

4.3.2 Heat Transfer Calculations

The heat transferred to the heat sink by the condenser is calculated from

$$\dot{q}_c = \rho_w \dot{V}_w \, Cp_w \, (T_{w2} - T_{w1} - \delta\epsilon)$$

where ρ_w and Cp_w are the density and specific heat capacity of water at the mean temperature $(T_{w1} + T_{w2})/2$, $\delta\epsilon$ is the offset correction between T_{w1} and T_{w2} at the mean temperature.

The refrigerant mass flow rate is determined by an energy balance across the condenser, assuming all the heat liberated by the working fluid in the heat exchanger is transferred to the heat sink:

$$\dot{m}_r = \frac{\dot{q}_c}{(h_2 - h_3)}$$

The evaporator heat transfer rate is deduced from the above mass flow rate and the measured refrigerant condition across the evaporator:

$$\dot{q}_e = \dot{m}_r \; (h_1 - h_4)$$

The assumption of isenthalpic expansion is used to determine h_4:

$$h_4 = h_3$$

The bulk discharge temperature of the air is determined from the above heat extraction rate, the measured air volume flow rate and measured inlet air temperature;

$$T_{a2} = T_{a1} - \frac{\dot{q}_e}{\rho_a \; \dot{V}_a \; Cp_a}$$

The mean evaporating and condensing temperatures are deduced from the mean absolute pressures in each heat exchanger;

$$T_e = TSAT \left\{ \frac{P_1 + P_5}{2} \right\} \text{ and } T_c = TSAT \left\{ \frac{P_2 + P_3}{2} \right\}$$

Where TSAT implies the saturated temperature corresponding to the known pressure. However for the 20 m experiments there was only one pressure measurement for the evaporator, P_1. The procedure to account for this is as follows. From the 30 m test results the evaporator pressure difference $(P_5 - P_1)$ was plotted as a function of the refrigerant mass flow rate. This graph was used to estimate the equivalent pressure drop for the 20 m experiments for the same mass flow rate by scaling the 30 m result by the ratio of the length of two evaporators. The saturated temperature corresponding to the estimated mean pressure could then be determined.

The log mean temperature difference for the evaporator is determined for the two phase region only:

$$\Delta T_{LM,1} = \frac{(T_{a1} - T_e) - (T_{a2} - T_e)}{\ln \left(\frac{T_{a1} - T_e}{T_{a2} - T_e} \right)}$$

The mean overall heat transfer coefficient, referred to the air side area, is estimated from:

$$U_o = \frac{\dot{q}_e}{A_o \, \Delta T_{LM,1}}$$

4.4 Analysis of Experimental Errors

4.4.1 Primary Measurements

The maximum likely uncertainties in the measurement of pressure, temperature, rate of flow and electrical power consumption are set out in table 4.10. The total error is considered to contain three elements; the hysteresis and non-linearity effects of the transducer, the uncertainty in the calibration constants, and the error inherent in data acquisition system i.e. the uncertainty in the voltage or resistance measurement of the HP3421A instrument. The transducer error is taken from the manufacturer's specification, or in the case of the resistance thermometers, from the appropriate standard. The calibration error reflects the precision to which the calibration constants are known. The data acquisition errors shown are the worst case values based on the one year specification of the HP3421A (67). The exception to this is the determination of the differential temperature (T_{w2} - T_{w1}). Since the two resistance measurements are taken close together in time, only the random count component of the error is relevant, and this is shown in the table. In most cases the data acquisition component of the total error is considerably smaller than the transducer and calibration errors.

4.4.2 Heat Transfer Measurements

The uncertainties in the primary measurements are combined to produce the overall errors in the calculation of heat transfer, temperature difference, etc. as shown in table 4.11. Notice the uncertainty on U_o, the evaporator heat transfer coefficient, is high because it is deduced from two measurements which both also carry high uncertainties.

5. RESULTS

5.1 The Effect of Evaporator Size and Evaporator Air Discharge Velocity on System Performance - Experiment A

The results of the test runs on the two evaporators are shown in table 4.12. Figure 4.13 illustrates the overall effect of evaporator size and discharge velocity on the heat transfer rates and COP_h of the heat pump. The air velocity recorded in the table and plotted in the figure is the mean velocity over the discharge face (i.e. outer tube row) of the evaporator. Figure 4.13 a) shows the heat pump heat output rate ($\dot{q}_c$), and evaporator heat extraction, rate ($\dot{q}_e$). The improvement in heat transfer across the evaporator with increasing air velocity is clearly shown in this figure. The effect of evaporator size may be summarized as an increase in heat pump heat output of 1 kW at a face air velocity of 1 m/s, produced by an increase in tube length from 20 to 30 m.

Figure 4.13 b) shows the COP_h of the heat pump expressed in two ways for each evaporator. The upper curves represent the COP_h of the refrigerant cycle, which excludes the power of the fan motor. This ratio is therefore the measured heat output rate to the measured compressor motor power, and it increases with increasing air velocity due to the improved evaporator heat transfer characteristics. The lower curves include the effect of fan power in the ratio, and a quite different result is obtained. Although the fan power was measured during the experiments, the measured power levels were not used to determine the COP_h because it became apparent that the method of controlling the fan speed, the electronic speed controller, was causing a disproportionate decrease in motor efficiency at low speeds. Instead a more realistic indication of the relationship between fan speed and motor power was obtained from the manufacturer's catalogue data which indicate the power levels for several, fixed speed motors (65). The quoted power levels are included in figure 4.12 a).

When fan power is included the curves contain points of maximum COP_h, demonstrating that at air velocities greater than the optimum value the increase in fan power necessary to further increase air velocity offsets the improvement in COP_h resulting from enhanced evaporator heat transfer. The 30 m coil shows a higher maximum COP_h, and has a lower optimum face velocity than the 20 m evaporator, a result which is in line with the theoretical predictions presented in section 2.6.

Evaporator heat exchange characteristics are examined in figure 4.14. The evaporator temperature difference is plotted in two forms for each evaporator, against face air velocity in figure 4.14 a). The upper curve for each evaporator is the simple air-on to refrigerant saturated temperature difference, $T_{a1} - T_e$, and the lower curve is the log mean temperature difference, as defined in section 4.3.2. This figure illustrates both the reduction in temperature difference attributable to the higher air velocities, which shows a rapid decline at low velocities and a levelling out at high velocities, and the decrease in temperature differences attributable to the greater surface area of the 30 m evaporator. The experimentally measured overall heat transfer coefficients are plotted in figure 4.14 b). As expected the coefficient increases with discharge velocity, and the rate of change diminishes at the higher velocities. A smooth curve is drawn through the points for the 30 m evaporator. The position of the 20 m data points suggests that the overall coefficient of the smaller evaporator is lower at low air velocities, although attention must be drawn to the high degree of uncertainty in U_o, as indicated on one of the data points. The predicted overall heat transfer coefficient is very similar for the two evaporators. Clearly the achieved evaporator performance is slightly less than expected.

Further results from this experiment are presented in figure 4.15. Here the fan motor shaft speed is the abscissa, and the relationship between motor speed and face velocity is shown in 4.15 a). The volumetric air flow rates are shown in figure 4.15 b). The results for the 30 m evaporator are based on measurement of the air velocity across a two dimensional grid marked out on the intake sections, whereas the line shown for the 20 m coil is obtained from the direct measurement of the discharge velocity. At a given fan speed the 30 m evaporator has a slightly higher air volumetric flow rate. This is because the frictional pressure drop of the tube bundle is lower for the 30 m evaporator at a given fan speed because of the lower air velocity through the coil. Figure 4.15 c) delineates the fan motor power consumption. Figure 4.15 d) indicates the dependence of fan sound pressure levels on fan speed. The data is supplied by the fan manufacturer and applies to free field dB(A) measurements at 3 m from the source (65). The circles plotted on this graph are the results of measurements taken with a portable, single octave analyser at 3 m from the ETR under non-ideal conditions. Background noise levels have distorted these readings but the results serve to reinforce the trend in the manufacturer's data.

To summarize the results from this experiment, the effect of increasing the surface area of the evaporator is to:

1. Significantly increase the heat pump heat output rate.

2. Slightly increase the heat pump COP_h.

3. Permit the fan to be operated at lower speed, which reduces the noise level.

5.2 The Effect of Evaporator Size on Heat Pump Performance for a Range of Ambient and Heat Sink Temperatures at Constant Fan Speed - Experiments B and C

In experiment B both evaporators were subject to a series of tests at ambient air temperatures in the range -6°C to 15°C. The 450 mm diameter fan was rotated at its maximum speed of 1360 rev/min in both cases. Heat sink conditions were held constant throughout the test series.

The results of test series B are presented in table 4.13. In addition, data from test series A at maximum fan speed, runs A-20-4 and A-30-4 are relevant and have been included in the plots. Figure 4.16 shows the principal heat pump heat and work transfer rates, $\dot{q}_c$, $\dot{q}_e$ and $\dot{W}_c$. The strong dependence of $\dot{q}_c$ on T_{a1} is clearly demonstrated. As before, the major effect of the greater evaporator surface area is an increase in heat output capacity. Taking the fan power as 0.44 kW in each case, the COP_h results for each evaporator are; at -1°C the 20 m coil gives 1.87, the 30 m coil gives 1.93, and at +7°C the respective results are 2.19 and 2.26. Notice that the fan speed set for this experiment is higher than the optima values for the evaporators and this leads to reduced COP_h due to the fan power penalty.

The heat transfer characteristics of the test evaporators are examined in figure 4.17. The log mean temperature difference across the evaporators is shown in figure 4.17 a). As the air temperature increases, the temperature difference also increases to bring about the rise in evaporator heat transfer rate. Since the change in temperature difference follows the increase in evaporator heat transfer rate, it follows that the evaporator heat transfer coefficient must be fairly constant throughout the temperature range, and this is shown to be so in figure 4.17 b). The coefficient is greater for the 20 m coil because the face air velocity is higher; at 1360 rev/min the 20 m coil is subjected to a mean face velocity of 3.3 m/s, while the 30 m coil has 2.6 m/s at the same fan speed. The overall coefficient does exhibit a slight rise with increasing air temperatures, and this is explained by the improvement in the internal coefficient caused by the increase in refrigerant mass flow rate, which is shown in figure 4.17 c).

To summarize the results of experiment B; the principal effect of the larger evaporator is an increase in heat pump heat output rate by 300-400 W across the temperature range. The COP_h is increased by 3 per cent, although both evaporators were operated at higher air velocities than the optima values.

Experiment C was similar to B except its purpose was to demonstrate the effect of condensing conditions on evaporator performance. Only the 30 m evaporator was tested, and the following conditions applied:

a) The superheat setting of the expansion valve was adjusted to increase the evaporator exit superheat. This is because the low superheat levels recorded for test B resulted in instability of the TEV operation, and periodic oscillation of the suction pressure was observed. The increase in superheat setting was sufficient to suppress the TEV instability.

b) The fan was replaced by a five bladed 450 mm diameter propeller fitted with a high efficiency 900 rev/min motor (fan B). This fan produces a slightly lower volumetric flow rate than the four bladed propeller; 1.25 m^3/s compared with 1.32 m^3/s, however the motor power consumption falls from 0.180 kW to 0.150 kW.

c) System steady state performance was measured at two condenser outlet water temperatures, 40°C and 55°C.

The results of this experiment, test C, are summarized in table 4.14 and figure 4.18. The increase in cycle COP_h as a result of lower water temperature is indicated by the increase in $\dot{q}_c$ and decrease in $\dot{W}_c$ at any air temperature. The increase in the heat transfer rate across the evaporator caused by the reduction in water temperature is about 0.7 kW, or 20 per cent at 0°C. The measured log mean temperature difference increases by this proportion, indicating that the overall heat transfer coefficient does not change significantly with water temperature, and this is also demonstrated in the figure. The measured evaporator heat transfer rate at a water outlet temperature of 55°C is very close to the values predicted by the analytical model, see table 4.5.

The performance curves presented in figure 4.18 are significant in that the evaporator and fan combination is the same as that used on the prototype heat pump. The prototype differs from the ETR configuration only in respect of the condenser selection.

5.3 Evaporator and System Performance Under Frosting Conditions

5.3.1 The Effect of Air Temperature - Experiment D

The 30 m evaporator, fitted with fan B at 900 rev/min was tested under the following conditions:

a) Heat sink water flow rate of 1 m^3/h and condenser water outlet temperature of 55°C.

b) Chamber air relative humidity was held at 90%

Three transient test runs were performed at chamber air temperatures of

-5°C, 0°C and +5°C. The results are presented in tables 4.15, 4.16, 4.17 respectively and plotted in figure 4.19. This figure illustrates the deterioration of the main performance parameters COP_h, $\dot{q}_c$, $\dot{V}_a$ and U_o in time, expressed as a proportion of the initial, non-frosted values. The figure also shows the change in temperature difference across the evaporator (T_{a1} - T_e) as the frost layer develops.

The figure indicates that the frost fouling effect is most pronounced at an air temperature of 0°C. The measured overall heat transfer coefficient falls to 50% of the initial value after 220 minutes of operation, compared with 360 minutes at -5°C and 460 minutes at 5°C. The pattern of frost development was the same for each air temperature. Initially a thin uniform coating of grey frost formed on both the primary and secondary surfaces of the heat exchanger. In time the frost layer thickened and took on a white, crystalline appearance. Eventually the gap between adjacent tube turns would be bridged, causing a substantial blockage to the flow of air through the tube bundle. However the frost did not bridge the gap between adjacent fins along the tubing, even under the most severe conditions. As shown in the figure, the frost layer does substantially reduce the air flow rate through the evaporator, which causes a decrease in the overall heat transfer coefficient. (The coefficient is also reduced by the consequential decrease in the refrigerant flow rate, but this is of secondary importance compared with the air side effects). The fall in U_o in turn increases the air to refrigerant temperature difference across the evaporator, and the reduction in suction pressure produces a decrease in heat pump heat output and COP_h.

5.3.2 The Effect of Water Temperature - Experiment E

The test conditions of the previous experiment were reproduced with the difference that the water outlet temperature from the condenser was reduced to 40°C. A frosting test was carried out for an air temperature of +5°C. The results are contained in table 4.18 and illustrated in figure 4.20 in which the system performance deterioration under frosting conditions for a water temperature of 40°C is contrasted with that for 55°C from Experiment D, for the same air temperature of 5°C.

The figure indicates that the reduction in water temperature causes a marked increase in the rate of performance deterioration due to frost fouling. The overall heat transfer coefficient falls to 50% of its initial value after 160 mins operation at 40°C, compared with 460 mins at 55°C. This is be explained by the decrease in the surface temperature of the evaporator caused by the increased heat extraction duty at the lower condensing temperature, as discussed in section 2.7.

5.3.3 The Effect of Fan Characteristic - Experiment F

Fans A and C were tested under the following conditions to determine the influence of fan characteristic on the performance of the evaporator under frosting conditions:

a) Heat sink water flow rate of 1.0 m^3/h and condenser water outlet temperature of 55°C.

b) Chamber air temperature of 5°C, 90% relative humidity.

The fan shaft speeds were selected to provide the same volumetric air flow rate from each fan under non frosting conditions; thus fan A was operated at 900 rev/min, fan B at 700 rev/min. The results of the two test runs are contained in tables 4.19 and 4.20, and illustrated in figure 4.21. Clearly the 450 mm diameter fan, A, fairs better under frosting conditions. This may be explained by the higher pressure rise characteristic of the smaller, faster fan, see figure 4.12. For a flow rate of 1.0 m^3/s fan A produces a static pressure rise of 30 Pa at 900 rev/min, compared with 18 Pa from fan C at 700 rev/min. This can be interpreted as an indication of the fan's ability to maintain flow under conditions of increasing flow resistance. This result demonstrates the importance of the pressure rise characteristic of the fan to the present application.

5.3.4 The Effect of Air Relative Humidity - Experiment G

In order to investigate the influence of relative humidity on frosting performance the 30 m evaporator was fitted with fan B which was rotated at 900 rev/min. The following conditions were maintained:

a) Heat sink water flow rate of 1.0 m^3/h and condenser water outlet temperature of 55°C.

b) Chamber air temperature of 0°C.

The chamber air relative humidity was varied in steps of 10% from 50% to 90%. No frosting effects were observed at relative humidities less than 70%. Tables 4.21, 4.22 and 4.16 record the results from the tests at 70%, 80% 90% relative humidity, and the comparative performances are shown in figure 4.22. The significance of relative humidity is clear from this figure. At 90% relative humidity the volumetric air flow rate falls to half its initial level after 170 mins compared with 370 mins at 80%. At 70% relative humidity 12 hours continuous operation were required before frost fouling effects were noticeable.

6. DISCUSSION

6.1 Selection of Evaporator and Fan for the Prototype Heat Pump

The payback analysis of section 2.6 concluded that an evaporator size greater than 30 m is unlikely to be economic. The detailed experiments performed on the 20 m and 30 m test evaporators have broadly confirmed the predicted performance differences between the two sizes. The analysis predicts that the 20 m evaporator is the more economic, but on balance the 30 m coil was selected for the prototype heat pump for the following considerations:

a) The 30 m coil results in a heat pump of noticeably greater heat output, and this benefit was not taken into consideration in the simple payback analysis.

b) The lower optimum air velocity is a benefit in minimizing the noise emission from the heat pump.

c) The 30 m coil has a greater capacity reserve and so should be able to operate for longer in frosting conditions.

The 450 mm diameter, five bladed prototype fan at 900 rev/min was chosen in preference to the 500 mm diameter fan at 700 rev/min because of the ability of the higher speed fan to continue supplying air to the evaporator under frosting conditions. This is due to the higher pressure rise developed by the higher speed fan.

6.2 Frost Accumulation Rate

The experimental results show the effects of frost growth on evaporator and heat pump performance and suggest that the rate of frost growth is influenced by a number of factors, e.g. air temperature and relative humidity, refrigerant condensing conditions and fan characteristic. This information may be marshalled in such a form to enable the prediction of the rate of frost growth, and hence the interval between defrost cycles, for the prototype heat pump. The frost growth model is consequently specific to the prototype air side configuration, i.e. 30 m evaporator and fan B rotating at 900 rev/min.

The frost accumulation rate, for the purpose of modelling, is represented by the rate of increase of the air to refrigerant temperature difference across the evaporator. This is calculated from the experimental results as the average rate of increase from the initial, frost-free temperature difference to an arbitrary maximum level of 12 deg C, the level at which a defrost cycle is likely to be initiated. The average results are presented in table 4.23. The growth model assumes a simple relationship (for a given evaporator and fan) between the average rate of increase in temperature difference and the difference between the dew point temperature of the incoming air and the mean surface temperature of the coil, thus:

$$\frac{d}{dt}(T_{a1} - T_e) = K_1 + K_2 (T_d - T_s) \text{ for } T_s \leqslant 0°C \text{ and } T_d > T_s$$

$$\frac{d}{dt}(T_{a1} - T_e) = 0 \text{ for } T_s > 0°C \text{ or } T_d < T_s$$

where K_1 = constant, with units deg C/h

K_2 = constant, with units h^{-1}

T_d = dew point temperature of the incoming air

T_s = mean outside surface temperature of the evaporator

$$T_s = \frac{(T_{a1} - T_{a2})}{2} - \frac{\dot{q}_e}{h_o \ \eta_a \ A_o}$$

The conditions attached to the model ensure that there is no frost growth if the coil surface temperature exceeds 0°C or exceeds the dew point temperature. The experimental results are plotted in figure 4.23 in such a form to enable the constants to be determined:

K_1 = 0.65 deg C/h, K_2 = 0.5 h^{-1}

The test point below the curve is for run D at T_{a1} = 3.7°C and T_{w2} = 55°C. Table 4.23 shows the mean surface temperature of the coil to be slightly above the freezing point. The most likely explanation for the observed non-zero frost growth rate is a gradual reduction in coil surface temperature during the test run, caused by the air side pressure drop increasing with the deposition of liquid condensate, until the surface temperature fell sufficiently below zero and ice began to form.

The experimental results also demonstrate the usefulness of the air to refrigerant temperature difference, $T_{a1} - T_e$, as an indicator of frost thickness. Since this temperature diffence rises significantly with increasing frost fouling it provides a convenient signal which may be used to automatically initiate a defrosting operation. This scheme was adopted on the prototype heat pump, and is discussed further in Chapter 6.

7. CONCLUSIONS

1. A high-finned, wire wound heat transfer tubing was selected as the basis of the heat pump evaporator. The tubing is formed into a two row, helical coil configuration. The evaporator is categorized as a cross-flow, two-circuit, direct expansion heat exchanger.

2. Analytical methods have been developed to model the air and refrigerant side heat transfer and pressure drop. The accuracy of the analytical predictions has been confirmed by experiment for air

flow rates greater than 0.9 m^3/s.

3. A comprehensive experimental facility has been developed to test the performance of the air source evaporators. This consists of fully instrumented heat pump and heat sink systems connected to a multi-channel, computer controlled data logger, and a walk-in environmental chamber to control air temperature and relative humidity.

4. Two experimental evaporators of the same configuration have been manufactured and tested using this facility. The nominal tube lengths and air side surface areas of the test evaporators were 20 m, 10.27 m^2 area and 30 m, 16.24 m^2 area.

5. The experimental results have confirmed that the principal effect of greater evaporator surface area is an increase in the heat output rate of the heat pump. The COP_h is also increased, but by a much lesser degree. However, a further benefit of the greater surface area is a reduction in the optimum air velocity through the coil. This helps to minimize the noise emission from the heat pump. The 30 m evaporator was selected for the prototype heat pump.

6. Tests were conducted on the 30 m evaporator to assess its performance under frosting conditions. Frost growth rates were highest at the air condition of 0°C/90 per cent relative humidity, coupled with low condenser water temperatures. These tests demonstrated the usefulness of the air to refrigerant temperature difference across the evaporator as an indication of the frost thickness, and therefore as a means of initiating a defrosting operation.

   The frost endurance tests also established the importance of fan selection with regard to pressure rise characteristic. A high pressure rise was found to be beneficial in maintaining a satisfactory air flow rate through a partially blocked evaporator. Accordingly, the 450 mm diameter, 5 blade, 900 rev/min fan was selected for the prototype heat pump.

d_1	tube bore diameter	10.9	mm
d_2	tube outside diameter	12.7	mm
d_3	fin outside diameter	38.5	mm
d_4	wire diameter	0.71	mm
t_w	tube wall thickness	0.9	mm
w_1	inside width of loop	1.3	mm
w_2	loop spacing	5.8	mm
N_1	number of loops per revolution	48.38	
A_i'	total internal area per meter run	0.0342	m^2/m
A_o'	total external area per meter run	0.5358	m^2/m
η_f	fin efficiency	0.9257	
η_a	surface effectiveness	0.9312	

Manufacturers tubing reference: 22/½/20/1.515/30.8/52.6/0.05 No 150

TABLE 4.1: SUMMARY OF THE DIMENSIONS AND DATA OF THE EVAPORATOR HEAT TRANSFER TUBING

		Evaporator 1	2	
	Nominal size	20	30	m
	Inner row tube length	9.247	12.606	m
	Outer row tube length	10.370	17.703	m
l	total tube length	19.617	30.309	m
A_i	total internal area	0.671	1.037	m^2
A_o	total external area	10.270	16.240	m^2
h	coil maximum height	0.286	0.390	m
d_5	coil outside diameter	0.639	0.639	m
$A_{f,o}$	coil outside face area	0.574	0.783	m^2
s_1	tube spacing	41.3	41.3	mm
s_2	row spacing	32.5	32.5	mm
s_3	diagonal spacing	52.6	52.6	mm
$A_{f,m}$	coil mean face area	0.405	0.626	m^2
N_c	No. of ref. side circuits	2	2	
$B'_{g,f}$	blockage caused by fins	6.317×10^{-3}	6.317×10^{-3}	m^2/m
$\frac{A_{f,m}}{A_{t,m}}$	ratio of mean face area to transverse area	1.853	1.853	m^2/m^2

TABLE 4.2: SUMMARY OF THE DIMENSIONS AND DATA OF THE TWO TEST EVAPORATORS

Run	Model Input				Model Output				Experimental Results				Error in output from model as a percentage of experimental result			
	T_{a1}	$\dot{V}_a$	ΔT_{sh}	T_c	$\dot{m}_r$	$\dot{q}_e$	T_e	$\dot{W}_f$	$\dot{m}_r$	$\dot{q}_e$	T_e	$\dot{W}_f$	$\dot{m}_r$	$\dot{q}_e$	T_e	$\dot{W}_f$
	(°C)	(m^3/s)	(deg C)	(°C)	(kg/s)	(kW)	(°C)	(kW)	(kg/s)	(kW)	(°C)	(kW)	(%)	(%)	(deg C)	(%)
a) 30m evaporator variation in T_{a1} at constant $\dot{V}_a$																
C-30-5	-6.31	1.25	5.5	55.5	0.0192	2.555	-10.96	0.150	0.0189	2.621	-11.95	0.150	+1.6	-2.5	+0.99	0
C-30-6	-1.34	1.25	6.0	55.7	0.0236	3.190	-7.01	0.150	0.0237	3.314	-7.59	0.150	-0.4	-3.7	+0.58	0
C-30-7	3.18	1.25	6.3	55.6	0.0285	3.904	-3.21	0.150	0.0285	4.004	-3.77	0.150	0	-2.5	+0.56	0
C-30-8	12.26	1.25	7.4	56.0	0.0388	5.435	4.01	0.150	0.0388	5.532	3.77	0.150	0	-2.1	+0.24	0
b) 30m evaporator, variation in $\dot{V}_a$ at T_{a1} nominally 6°C																
A-30-1	6.32	0.47	3.0	55.1	0.0288	3.894	-3.46	0.009	0.0259	3.553	-7.12	0.031	+11.2	+9.6	+3.66	-71
A-30-2	5.86	0.94	3.7	55.4	0.0318	4.338	-1.14	0.066	0.0306	4.238	-3.05	0.087	+3.9	+2.4	+1.91	-24
A-30-3	5.67	1.41	4.4	55.4	0.0328	4.503	-0.32	0.214	0.0326	4.541	-1.49	0.185	+0.6	-0.8	+1.17	+16
A-30-4	5.83	2.05	4.7	55.5	0.0341	4.693	0.55	0.639	0.0343	4.795	-0.19	0.444	-0.6	-2.1	+0.74	+44
c) 20m evaporator, variation in $\dot{V}_a$ at T_{a1} nominally 6°C																
A-20-1	4.96	0.46	3.7	54.3	0.0261	3.550	-5.84	0.011	0.0219	2.985	-10.59	0.036	+19.2	+18.9	+4.75	-69
A-20-2	5.16	0.75	3.7	54.1	0.0290	3.977	-3.76	0.045	0.0260	3.577	-7.19	0.088	+10.3	+11.2	+3.43	-49
A-20-3	5.80	1.32	4.1	55.0	0.0316	4.332	-1.60	0.229	0.0316	4.352	-2.59	0.204	0	-0.5	+0.99	+12
A-20-4	5.92	1.90	4.1	55.1	0.0330	4.538	-0.68	0.651	0.0327	4.508	-1.68	0.444	+0.9	+0.7	+1.0	+47

TABLE 4.3: VALIDATION OF EVAPORATOR MODEL - COMPARISON WITH EXPERIMENTAL RESULTS

Evaporator size (m)	Increase in capital cost of the heat pump (£)	Predicted COP_h at 4°C air temp. 49°C water return temp.	Cost per useful kWh (p/kWh)	Cost of electricity used (£/year)	Annual saving resulting from larger evaporator (£/year)	Simple pay back period for increase in evaporator cost over 10m size (years)
10m	0	2.267	1.875	337.5	-	-
20	83	2.383	1.783	320.9	16.6	5.0
30	167	2.424	1.753	315.5	22.0	7.6
40	250	2.442	1.740	313.2	24.0	10.4
50	333	2.451	1.734	312.1	25.4	13.1

NOTE: Electriity prices and annual heating requirement as in table 3.8.

TABLE 4.4: PAYBACK ANALYSIS FOR INCREASES IN THE SIZE OF THE EVAPORATOR

T_{w1}	T_{a1}	q_e	$T_{a1}-T_e$	$\frac{q_e}{T_{a1}-T_e}$	$\frac{A_{o,2}}{A_o}$	m_r	$h_{i,1}$	$h_{i,2}$	x_4	T_s
(°C)	(°C)	(W)	(deg C)	(W/deg C)		(kg/s)	(W/m^2K)	(W/m^2K)		(°C)
30	-5	3867	5.62	688.1	0.112	0.0242	2163	212	0.26	-8.7
30	0	4621	6.47	714.3	0.098	0.0289	2345	249	0.25	-4.4
30	5	5443	7.39	736.5	0.087	0.0340	2518	289	0.24	-0.1
30	10	6330	8.38	755.4	0.079	0.0396	2684	332	0.22	+4.1
30	15	7283	9.44	771.5	0.072	0.0457	2844	378	0.21	+8.2
40	-5	3379	5.06	667.8	0.133	0.0229	2093	204	0.32	-8.3
40	0	4096	5.85	700.2	0.113	0.0277	2295	241	0.30	-3.9
40	5	4873	6.71	726.2	0.099	0.0330	2488	282	0.29	+0.4
40	10	5715	7.64	748.0	0.089	0.0387	2672	327	0.28	+4.6
40	15	6618	8.63	766.9	0.081	0.0449	2849	375	0.27	+8.8
50	-5	2890	4.45	649.4	0.162	0.0214	2001	193	0.37	-7.9
50	0	3545	5.22	679.1	0.136	0.0262	2216	232	0.36	-3.5
50	5	4280	6.01	712.1	0.116	0.0316	2431	274	0.35	+0.9
50	10	5072	6.88	737.2	0.102	0.0375	2635	320	0.34	+5.2
50	15	5924	7.81	758.5	0.092	0.0440	2830	270	0.33	+9.4
60	-5	2357	3.92	601.3	0.211	0.0194	1859	179	0.44	-7.4
60	0	2985	4.59	650.3	0.172	0.0244	2111	220	0.42	-3.0
60	5	3658	5.31	688.9	0.141	0.0299	2344	263	0.42	+1.4
60	10	4400	6.10	721.3	0.121	0.0361	2572	311	0.41	+5.7
60	15	5196	6.96	746.6	0.107	0.0427	2788	363	0.40	+10.0

TABLE 4.5: PREDICTED PERFORMANCE CHARACTERISTICS FOR 30m EVAPORATOR WITH 1.25 m^3/s AIR FLOW RATE

Test	Evap.	Fan	Fan speed (rev/min)	Distributor	T_a (°C)	Rh (%)	T_{w2} (°C)	V_w (m^3/s)
A	20 + 30	A	varies	E	7	60	55	1.0
B	20 + 30	A	1350	E	varies	60	55	1.0
C	30	B	900	D	varies	50	varies	1.0
D	30	B	900	D	varies	90	55	1.0
E	30	B	900	D	5	90	varies	1.0
F	30	A + C	900 + 700	D	5	90	55	1.0
G	30	B	900	D	0	varies	55	1.0

TABLE 4.6: SUMMARY OF THE EVAPORATOR TEST PROGRAMME

Designation	Fan diameter (mm)	No. of blades	Max. speed (rev/min)
A	450	4	1350
B	450	5	900
C	500	4	900

TABLE 4.7: FAN DESIGNATION

Designation	Tube bore (mm)	Tube length (mm)
A	3.34 ($^3/_{16}$ inch O.D.)	1000
B	3.34	600
C	3.34	200
D	4.93 ($^1/_4$ inch O.D.)	1000
E	3.34	560
F	3.34	1320

TABLE 4.8: DISTRIBUTOR TUBING DESIGNATION

Measurement	Equation	Calibration constants
Pressure	$V_p = k_p [A_p P + B_p]$	k_p is the ratio of the applied excitation voltage to the nominal voltage of 10V. A_p + B_p are calibration constants determined by measuring the signal voltage at two known pressures within the range of the transducer.
Rate of flow	$V_v = M_v \dot{V}_w + C_v$	M_v and C_v found by gravimetric calibration.
Compressor power	$V_w = M_w \dot{W}_c + C_w$	M_w and C_w found from supplied transducer characteristics and checked by calibration.
Temperature	$R_T = R_o\ (1 + A_T T + B_T T^2)$	From BS 1904: 1984 $A_T = 3.90802 \times 10^{-3}$ °C^{-1} $B_T = -5.802 \times 10^{-7}$ °C^{-2} R_o = resistance of element at 0°C
Relative humidity	$V_{Rh} = M_R Rh + C_R$	M_R and C_R found by comparing the signal voltage with the indicated relative humidity from the Environmental Chamber control system

TABLE 4.9: EVAPORATOR TEST RIG INSTRUMENTAION SYSTEM. THE RELATIONSHIPS BETWEEN THE MEASURED QUANTITIES AND THE TRANSDUCER SIGNALS

Measurement	Transducer* range	Transducer* output	Maximum transducer error	Maximum calibration error	Maximum data acquisition error	Maximum total error
Pressure P_1, P_4, P_5	0-7 bar, g (3 bar, g)	0-0.2V (0.086V)	$\pm$ 0.3%	$\pm$ 0.3%	$\pm$ 0.02%	$\pm$ 0.62%
Pressure P_2, P_3	0-35 bar, g (21 bar,g)	0-0.2V (0.12V)	$\pm$ 0.2%	$\pm$ 0.3%	$\pm$ 0.02%	$\pm$ 0.52%
Temperature (all insertion type thermometers)	-15°C to 130°C (0°C)	94.2 to 149.4 (100Ω)	$\pm$ (0.15+0.002 $\|T\|$) ie $\pm$0.15 deg C at 0°C		$\pm$ 0.023	$\pm$ 0.21 deg C at 0°C
Differential Temperature (T_{w2}-T_{w1}-$\delta\epsilon$)	5-12 deg C (6 deg C)			$\pm$ 0.05 deg C	$\pm$ 0.01	$\pm$ 0.08 deg C
Water flow rate V_w	0.06-3.9 m^3/h ($1m^3/h$)	0-2V (0.51V)		$\pm$ 1%	$\pm$ 0.02%	$\pm$ 1.02%
Compressor Electrical Power $\dot{W}_c$	0-6kW (2.5kW)	0-10V (4.2V)	$\pm$ 1.2% at 2.5kW		$\pm$ 0.02%	$\pm$ 1.22%

* figure in brackets indicates typical measured value

TABLE 4.10: MEASUREMENT ACCURACY OF THE PRIMARY VARIABLES

Measurement	Method	Maximum Error	Notes
Condenser heat transfer rate, $\dot{q}_c$	$\dot{q}_c = \rho_w \dot{V}_w Cp_w (T_{w2} - T_{w1} - \delta\epsilon)$	$\pm$ 2.3%	$\delta\epsilon$ is the known thermometer offset error at temperature $(T_{w1} + T_{w2})/2$
Refrigerant mass flow rate, $\dot{m}_r$	$\dot{m}_r = \frac{\dot{q}_c}{(h_2 - h_3)}$	+ 4.3% - 2.6%	Error in $\dot{q}_c$ = + 2.3%, error due to loss from condenser and pipework = +1.7%, error in specific enthalpy change owing to uncertainties in temperature + pressure measurement = $\pm$ 0.3%.
Evaporator heat transfer rate, $\dot{q}_e$	$\dot{q}_e = \dot{m}_r (h_1 - h_3)$	+ 4.7% - 3.0%	Error in $(h_1 - h_3)$ is $\pm$ 0.4%
Evaporating temperature, T_e	$T_e = \text{TSAT} \frac{(P_1 + P_5)}{2}$	$\pm$ 0.14 deg C	
Air temperature drop, $(T_{a1} - T_{a2})$	$T_{a1} - T_{a2} = \frac{\dot{q}_e}{\rho_a \dot{V}_a Cp_a}$	+ 0.3 deg C - 0.2 deg C	Based on a 3 deg C temperature fall. Error in $\dot{V}_a$ is estimated to be $\pm$ 5%
Air to refrigerant temperature difference	$\Delta T_e = (T_{a1} - T_e)$	$\pm$ 0.35 deg C	This is equivalent to 6% of the typical temperature difference of 6 deg C.
Evaporator overall heat transfer coefficient	$U_o = \frac{\dot{q}_e}{A_o \Delta T_{LM,1}}$	+ 10.4% - 9.0%	Taking the accuracy of ΔT_e to be representative of $\Delta T_{LM,1}$

TABLE 4.11: UNCERTAINTIES IN THE HEAT TRANSFER CALCULATIONS

Run	u_f	T_a	Rh	T_{a1}	T_{w2}	$\dot{V}_w$	T_e	T_c	ΔT_{sh}	ΔT_{sc}	$\dot{q}_c$	$\dot{W}_c$	$\dot{m}_r$	$\dot{q}_e$	T_{a2}	$\Delta T_{LM,1}$	U_o
	(m/s)	(°C)	(%)	(- °C -)		(m^3/h)	(-°C -)		(- deg C -)		(- kW -)		(kg/s)	(kW)	(°C)	(deg C)	(W/m^2K)
a) 20m evaporator																	
A-20-1	0.8	6.8	31	4.96	55.0	1.00	-10.59	54.3	3.7	2.0	4.91	2.38	0.0219	2.985	-0.16	12.82	22.7
A-20-2	1.3	6.8	29	5.16	54.5	1.00	-7.19	54.1	3.7	1.6	5.69	2.52	0.0260	3.577	1.35	10.33	33.7
A-20-3	2.3	6.8	48	5.80	55.1	1.00	-2.59	55.0	4.1	1.2	6.67	2.70	0.0316	4.352	3.18	7.00	60.6
A-20-4	3.3	6.9	36	5.92	55.1	1.00	-1.68	55.1	4.1	1.2	6.87	2.74	0.0327	4.508	4.03	6.61	66.4
b) 30m evaporator																	
A-30-1	0.6	6.9	41	6.32	55.0	0.99	-7.12	55.1	3.0	2.8	5.59	2.54	0.0259	3.553	0.23	10.09	21.7
A-30-2	1.2	6.9	47	5.86	55.1	0.99	-3.05	55.4	3.7	2.3	6.50	2.72	0.0306	4.238	2.27	6.96	37.4
A-30-3	1.8	6.8	64	5.67	55.0	0.99	-1.49	55.4	4.4	2.2	6.85	2.77	0.0326	4.541	3.09	5.77	48.4
A-30-4	2.6	6.9	36	5.83	54.9	0.99	-0.19	55.5	4.7	2.2	7.15	2.81	0.0343	4.795	3.94	5.02	58.9

TABLE 4.12: RESULTS OF EXPERIMENT TO DETERMINE THE EFFECT OF AIR VELOCITY ON EVAPORATOR AND HEAT PUMP PERFORMANCE FOR TWO SIZES OF EVAPORATOR. EXPERIMENT A

Run	T_a (°C)	Rh (%)	T_{a1} (°C)	T_{w2} (°C)	$\dot{V}_w$ (m^3/h)	T_e (°C)	T_c (°C)	ΔT_{sh} (-deg	ΔT_{sc} C-)	$\dot{q}_c$ (- kW	$\dot{W}_c$ -)	$\dot{m}_r$ (kg/s)	$\dot{q}_e$ (kW)	T_{a2} (°C)	$\Delta T_{LM,1}$ (deg C)	U_o (W/m^2K)
a) 20m evaporator																
B-20-1	-5.4	30	-5.73	55.1	0.99	-11.33	55.4	2.6	2.7	4.67	2.33	0.0208	2.837	-6.87	5.01	55.2
B-20-2	-0.3	30	-0.94	55.0	0.99	-7.41	54.6	3.1	1.9	5.56	2.51	0.0256	3.509	-2.37	5.73	59.7
B-20-3	9.2	42	7.60	55.8	1.00	-0.35	55.9	4.1	0.9	7.17	2.83	0.0345	4.740	5.61	6.91	66.8
B-20-4	15.3	42	13.70	55.3	1.00	4.29	55.9	4.4	0.3	8.36	2.99	0.0413	5.722	11.22	8.11	68.7
b) 30m evaporator																
B-30-1	-5.1	67	-5.38	55.0	1.00	-10.00	54.9	2.3	3.9	4.96	2.41	0.0224	3.073	-6.54	4.01	47.2
B-30-2	4.9	85	4.27	55.2	1.00	-1.77	55.5	4.2	2.4	6.87	2.76	0.0327	4.544	2.49	5.10	54.9
B-30-3	14.8	58	13.62	55.0	0.99	5.87	55.9	7.1	1.4	8.75	3.04	0.0433	6.156	11.11	6.41	59.1

TABLE 4.13: RESULTS OF EXPERIMENT TO DETERMINE THE EFFECT OF AIR TEMPERATURE ON EVAPORATOR AND HEAT PUMP PERFORMANCE. EXPERIMENT B, FAN A AT 1350 rev/min

Run	T_a (°C)	Rh (%)	T_{a1} (°C)	T_{w2} (°C)	$\dot{V}_w$ (m^3/h)	T_e (°C)	T_c (°C)	ΔT_{sh} (-deg	ΔT_{sc} C-)	x_4	$\dot{q}_c$ (- kW	$\dot{W}_c$ -)	COP_h	$\dot{m}_r$ (kg/s)	$\dot{q}_e$ (kW)	T_{a2} (°C)	$\Delta T_{LM,1}$ (deg C)	U_o (W/m^2K)
C-30-1	-4.88	50	-6.21	40.0	1.04	-12.94	41.2	7.07	3.66	0.28	4.812	2.021	2.047	0.0208	3.284	-8.18	5.688	35.55
C-30-2	-0.53	40	-2.39	40.0	1.04	-9.68	41.4	7.31	2.99	0.28	5.527	2.133	2.421	0.0244	3.870	-4.75	6.033	39.50
C-30-3	4.41	41	2.60	39.9	1.05	-5.90	41.5	7.89	2.31	0.26	6.424	2.213	2.719	0.0219	4.651	-0.29	6.955	41.18
C-30-4	14.82	40	11.73	40.2	1.04	1.80	42.2	8.77	1.69	0.23	8.364	2.436	3.234	0.0393	6.357	7.64	7.705	50.80
C-30-5	-5.17	50	-6.31	55.1	1.03	-11.95	55.5	5.50	4.62	0.38	4.315	2.262	1.789	0.0189	2.621	-7.88	4.812	33.54
C-30-6	0.16	41	-1.34	55.2	1.03	-7.59	55.7	5.97	3.86	0.36	5.220	2.449	2.008	0.0237	3.314	-3.36	5.174	39.44
C-30-7	4.91	38	3.18	55.0	1.03	-3.77	55.6	6.32	3.01	0.34	6.096	2.602	2.215	0.0285	4.004	0.69	5.613	43.92
C-30-8	14.84	40	12.26	55.1	1.04	3.77	56.0	7.43	1.94	0.33	7.973	2.924	2.594	0.0388	5.532	8.70	6.550	52.01

TABLE 4.14: RESULTS OF EXPERIMENT TO DETERMINE THE EFFECT OF AIR AND WATER TEMPERATURES ON THE PERFORMANCE OF THE 30m EVAPORATOR. EXPERIMENT C, FAN B AT 900 rev/min

Time	T_a	Rh	$\dot{V}_a$	T_{a1}	T_{w2}	$\dot{V}_w$	T_e	T_c	ΔT_{sh}	ΔT_{sc}	$\dot{q}_c$	$\dot{W}_c$	$\dot{m}_r$	$\dot{q}_e$	T_{a2}	$\Delta T_{LM,1}$	$T_{a1}-T_e$	U_o
(mins)	(°C)	(%)	(m^3/s)	(°C)	(°C)	(m^3/h)	(°C)	(°C)	(-deg	C-)	(- kW	-)	(kg/s)	(kW)	(°C)	(- deg	C -)	(W/m^2K)
120	-5.2	88	1.41	-6.23	54.9	1.00	-12.03	55.5	4.4	1.9	4.298	2.343	0.0192	2.587	-7.61	5.08	5.80	31.38
180	-5.2	88	1.15	-5.78	55.2	1.00	-11.79	55.9	4.1	2.1	4.283	2.340	0.0194	2.559	-7.48	5.11	6.01	31.32
240	-5.2	89	0.99	-5.63	55.2	1.00	-12.04	55.8	4.0	2.1	4.269	2.313	0.0193	2.582	-7.60	5.36	6.41	29.65
310	-5.2	90	0.41	-5.33	54.9	1.00	-13.88	55.5	3.8	2.1	3.905	2.245	0.0173	2.358	-9.72	6.09	8.55	23.83
370	-5.2	89	0.50	-5.26	54.3	1.00	-16.23	54.9	3.3	1.9	3.523	2.139	0.0159	2.123	-8.46	9.28	10.97	14.09
420	-5.2	88	0.42	-5.15	53.9	1.00	-18.60	54.4	2.2	2.9	3.138	2.020	0.0137	1.832	-8.47	11.71	13.45	9.63

TABLE 4.15: EVAPORATOR PERFORMANCE UNDER FROSTING CONDITIONS. RESULTS OF EXPERIMENT D AT -5°C

Time	T_a	Rh	$\dot{V}_a$	T_{a1}	T_{w2}	$\dot{V}_w$	T_e	T_c	ΔT_{sh}	ΔT_{sc}	$\dot{q}_c$	$\dot{W}_c$	$\dot{m}_r$	$\dot{q}_e$	T_{a2}	$\Delta T_{LM,1}$	$T_{a1}-T_e$	U_o
(mins)	(°C)	(%)	(m^3/s)	(°C)	(°C)	(m^3/h)	(°C)	(°C)	(-deg	C-)	(- kW	-)	(kg/s)	(kW)	(°C)	(- deg	C -)	(W/m^2K)
34	0.0	89	1.20	-0.70	55.4	1.00	-6.40	56.3	5.1	0.9	5.328	2.599	0.0250	3.378	-2.86	4.54	5.70	45.87
110	-0.2	90	1.00	-0.73	55.3	1.00	-7.01	56.2	4.9	1.0	5.234	2.538	0.0245	3.307	-3.28	4.90	6.28	41.58
170	-0.2	90	0.67	-0.58	55.1	1.01	-8.42	56.0	5.1	1.3	5.042	2.494	0.0234	3.162	-4.21	5.84	7.84	33.36
230	-0.2	90	-	-0.54	54.4	1.00	-11.45	55.3	3.9	1.8	4.448	2.339	0.0205	2.766	-5.38	8.26	10.91	20.63
240	-	-	0.44	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
290	-0.2	90	0.33	-0.28	53.6	1.00	-16.38	54.3	3.5	2.2	3.601	2.108	0.0164	2.206	-5.43	13.36	16.10	10.17

TABLE 4.16: EVAPORATOR PERFORMANCE UNDER FROSTING CONDITIONS. RESULTS OF EXPERIMENT D AT 0°C

Time	T_a	Rh	$\dot{V}_a$	T_{a1}	T_{w2}	$\dot{V}_w$	T_e	T_c	ΔT_{sh}	ΔT_{sc}	$\dot{q}_c$	$\dot{W}_c$	$\dot{m}_r$	$\dot{q}_e$	T_{a2}	$\Delta T_{LM,1}$	$T_{a1}-T_e$	U_o
(mins)	(°C)	(%)	(m^3/s)	(°C)	(°C)	(m^3/h)	(°C)	(°C)	(-deg	C-)	(- kW	-)	(kg/s)	(kW)	(°C)	(- deg	C -)	(W/m^2K)
60	4.9	90	1.25	3.70	55.0	0.99	-2.83	55.1	6.2	2.6	6.272	2.701	0.0296	4.177	1.08	5.11	6.53	50.32
120	4.9	90	1.25	3.53	55.0	1.00	-2.95	55.2	6.2	2.7	6.239	2.690	0.0294	4.144	0.94	5.07	6.48	39.38
180	4.6	89	1.25	3.49	55.0	1.00	-3.79	55.1	6.3	2.6	6.115	2.665	0.0288	4.049	0.95	5.92	7.28	42.10
300	4.8	90	1.07	3.83	55.0	1.00	-3.74	55.2	6.4	2.8	5.989	2.646	0.0282	3.962	0.92	6.00	7.57	40.69
360	4.8	90	1.02	4.32	55.0	1.00	-4.36	55.2	6.1	2.8	5.926	2.640	0.0278	3.898	1.32	7.08	8.68	33.92
420	4.4	89	0.88	4.38	54.9	1.00	-5.66	55.0	5.7	2.9	5.703	2.578	0.0266	3.719	1.07	8.28	10.04	27.67
480	4.9	90	0.70	4.52	54.6	1.00	-6.52	54.8	5.7	2.9	5.476	2.562	0.0252	3.530	0.54	8.90	11.04	24.42

TABLE 4.17: EVAPORATOR PERFORMANCE UNDER FROSTING CONDITIONS. RESULTS OF EXPERIMENT D AT + 5°C

Time	T_a	Rh	$\dot{V}_a$	T_{a1}	T_{w2}	$\dot{V}_w$	T_e	T_c	ΔT_{sh}	ΔT_{sc}	$\dot{q}_c$	$\dot{W}_c$	$\dot{m}_r$	$\dot{q}_e$	T_{a2}	$\Delta T_{LM,1}$	$T_{a1}-T_e$	U_o
(mins)	(°C)	(%)	(m^3/s)	(°C)	(°C)	(m^3/h)	(°C)	(°C)	(-deg	C-)	(- kW	-)	(kg/s)	(kW)	(°C)	(- deg	C -)	(W/m^2K)
60	4.7	90	1.24	3.56	41.1	1.01	-3.94	42.1	7.8	1.5	6.790	2.340	0.0311	4.929	0.47	5.82	7.50	52.14
120	4.8	90	1.02	4.26	39.9	1.00	-4.78	41.0	7.9	2.1	6.570	2.277	0.0299	4.798	0.57	7.04	9.04	41.99
180	4.9	90	0.60	4.44	38.5	1.00	-11.86	39.5	7.0	2.4	5.160	2.072	0.0229	3.644	-0.30	13.79	16.30	16.27

TABLE 4.18: EVAPORATOR PERFORMANCE UNDER FROSTING CONDITIONS. RESULTS OF EXPERIMENT E AT +5°C AIR TEMPERATURE AND 40°C WATER FLOW TEMPERATURE

Time	T_a	Rh	$\dot{V}_a$	T_{a1}	T_{w2}	$\dot{V}_w$	T_e	T_c	ΔT_{sh}	ΔT_{sc}	$\dot{q}_c$	$\dot{W}_c$	$\dot{m}_r$	$\dot{q}_e$	T_{a2}	$\Delta T_{LM,1}$	$T_{a1}-T_e$	U_o
(mins)	(°C)	(%)	(m^3/s)	(°C)	(°C)	(m^3/h)	(°C)	(°C)	(-deg	C-)	(- kW	-)	(kg/s)	(kW)	(°C)	(- deg	C -)	(W/m^2K)
60	4.9	90	1.26	3.68	55.3	0.99	-3.18	56.0	3.6	2.6	6.396	2.707	0.0303	4.182	1.08	5.46	6.80	47.21
120	4.9	90	1.31	3.72	55.2	0.99	-2.69	56.0	4.2	2.7	6.336	2.713	0.0301	4.175	1.22	5.06	6.41	50.86
180	4.9	90	1.26	4.07	55.3	0.99	-2.62	56.2	3.4	2.8	6.399	2.711	0.0305	4.207	1.46	5.28	6.69	49.10
240	5.0	90	1.21	4.35	55.1	0.99	-3.38	55.8	3.8	2.5	6.292	2.686	0.0298	4.119	1.69	6.30	7.73	40.20
300	4.9	90	0.98	4.64	55.1	0.99	-4.44	55.8	2.4	2.6	6.154	2.627	0.0291	3.987	1.46	7.37	9.08	33.29
360	5.0	90	0.89	4.73	54.8	0.99	-6.24	55.4	1.8	2.8	5.800	2.554	0.0273	3.727	1.45	9.23	10.97	24.85
420	4.9	90	0.70	4.77	53.9	0.99	-10.53	54.4	1.6	3.1	4.869	2.376	0.0224	3.061	2.08	13.73	15.12	13.73

TABLE 4.19: EVAPORATOR PERFORMANCE UNDER FROSTING CONDITIONS. RESULTS OF EXPERIMENT F. FAN A AT 900 rev/min

Time	T_a	Rh	$\dot{V}_a$	T_{a1}	T_{w2}	$\dot{V}_w$	T_e	T_c	ΔT_{sh}	ΔT_{sc}	$\dot{q}_c$	$\dot{W}_c$	$\dot{m}_r$	$\dot{q}_e$	T_{a2}	$\Delta T_{LM,1}$	$T_{a1}-T_e$	U_o
(mins)	(°C)	(%)	(m^3/s)	(°C)	(°C)	(m^3/h)	(°C)	(°C)	(-deg	C-)	(- kW	-)	(kg/s)	(kW)	(°C)	(- deg	C -)	(W/m^2K)
20	5.0	93	1.32	4.01	55.0	1.00	-2.90	56.0	6.8	2.7	6.386	2.727	0.0310	4.348	1.43	5.52	6.91	48.49
60	4.8	89	1.27	3.94	55.1	1.00	-3.24	56.0	6.5	2.9	6.392	2.697	0.0298	4.185	1.36	5.80	7.18	44.47
120	5.0	88	1.04	3.96	55.1	1.00	-3.20	55.9	6.7	3.0	6.358	2.693	0.0296	4.156	0.83	5.45	7.16	46.98
180	5.0	90	0.99	4.54	55.1	1.00	-3.71	55.9	6.5	3.0	6.281	2.691	0.0292	4.095	1.30	6.50	8.25	38.81
240	5.0	91	-	4.23	54.4	1.00	-5.54	55.2	6.4	3.2	5.767	2.578	0.0263	3.702	-	-	9.77	-
300	4.9	90	0.54	4.23	54.3	1.00	-11.52	54.7	6.1	4.2	4.739	2.339	0.0208	2.919	0.0	13.53	15.75	11.41
360	4.9	95	0.36	4.06	53.3	1.00	-17.56	53.5	5.4	4.7	3.613	2.069	0.0155	2.169	-0.65	19.17	21.62	6.97

TABLE 4.20: EVAPORATOR PERFORMANCE UNDER FROSTING CONDITIONS. RESULTS OF EXPERIMENT F. FAN C AT 700 rev/min

Time	T_a	Rh	$\dot{V}_a$	T_{a1}	T_{w2}	$\dot{V}_w$	T_e	T_c	ΔT_{sh}	ΔT_{sc}	$\dot{q}_c$	$\dot{W}_c$	$\dot{m}_r$	$\dot{q}_e$	T_{a2}	$\Delta T_{LM,1}$	$T_{a1}-T_e$	U_o
(mins)	(°C)	(%)	(m^3/s)	(°C)	(°C)	(m^3/h)	(°C)	(°C)	(-deg	C-)	(- kW	-)	(kg/s)	(kW)	(°C)	(- deg	C -)	(W/m^2K)
30	-0.1	70	1.25	-1.43	55.1	1.00	-7.53	55.6	6.1	4.2	5.233	2.476	0.0238	3.329	-3.48	5.01	6.10	40.95
180	-0.4	69	1.25	-1.51	55.0	0.99	-7.89	55.6	6.1	4.3	5.130	2.444	0.0234	3.268	-3.52	5.31	6.38	37.88
420	-0.1	70	1.25	-1.36	55.0	0.99	-7.76	55.7	6.2	4.4	5.151	2.445	0.0235	3.293	-3.39	5.32	6.40	38.11
720	-0.1	70	1.11	-0.54	55.1	0.99	-8.04	55.8	6.2	4.4	5.061	2.442	0.0230	3.208	-2.76	6.33	7.50	31.23

TABLE 4.21: EVAPORATOR PERFORMANCE UNDER FROSTING CONDITIONS. RESULTS OF EXPERIMENT G AT 70% Rh

Time (mins)	T_a (°C)	Rh (%)	$\dot{V}_a$ (m^3/s)	T_{a1} (°C)	T_{w2} (°C)	$\dot{V}_w$ (m^3/h)	T_e (°C)	T_c (°C)	ΔT_{sh} (-deg C-)	ΔT_{sc} (-deg C-)	$\dot{q}_c$ (- kW -)	$\dot{W}_c$ (- kW -)	$\dot{m}_r$ (kg/s)	$\dot{q}_e$ (kW)	T_{a2} (°C)	$\Delta T_{LM,1}$ (- deg C -)	$T_{a1}-T_e$ (- deg C -)	U_o (W/m^2K)
30	-0.2	80	1.25	-1.58	55.0	1.00	-8.02	55.6	6.2	4.3	5.064	2.453	0.0231	3.224	-3.56	5.39	6.44	36.83
60	-0.1	78	1.25	-1.34	55.0	1.00	-7.45	55.7	5.9	4.3	5.229	2.466	0.0241	3.372	-3.42	5.00	6.11	41.54
120	-0.2	80	1.20	-1.12	55.1	1.00	-7.37	55.7	6.0	4.3	5.240	2.465	0.0239	3.343	-3.26	5.12	6.25	40.23
180	-0.2	80	1.11	-0.96	55.1	1.00	-7.83	55.7	6.1	4.3	5.161	2.449	0.0236	3.295	-3.24	5.70	6.87	35.61
240	-0.1	80	1.07	-0.44	55.0	1.00	-8.05	55.7	6.2	4.4	5.069	2.425	0.0231	3.229	-2.76	6.38	7.61	31.16
300	-0.2	80	0.93	-0.40	54.8	1.00	-9.48	55.4	6.4	4.6	4.804	2.379	0.0216	3.024	-2.90	7.76	9.08	24.00
360	-0.2	80	0.65	-0.39	54.5	1.00	-11.04	55.1	6.0	4.7	4.491	2.282	0.0200	2.787	-3.69	8.90	10.65	19.29
420	-0.2	80	0.56	-0.35	54.1	0.99	-13.02	55.7	5.6	4.9	4.118	2.188	0.0182	2.538	-3.84	10.83	12.67	14.43
480	-0.2	80	0.42	-0.38	53.8	0.99	-14.98	54.3	5.0	5.2	3.787	2.105	0.0164	2.282	-4.56	12.39	14.60	11.34

TABLE 4.22: EVAPORATOR PERFORMANCE UNDER FROSTING CONDITIONS. RESULTS OF EXPERIMENT G AT 80 % Rh

Run	T_{a1} (°C)	Rh (%)	air dew point T_d (°C)	Evaporator mean surface temperature T_s (°C)	$\frac{d}{dt}(T_{a1}-T_e)$ (deg C/h)	T_d-T_s (deg C)
D	-6.2	90	-7.3	-8.3	1.07	1.0
D	-0.7	90	-1.8	-3.6	1.54	1.8
D	3.7	90	2.0	+0.1	0.70	1.9
E	3.6	90	2.0	-0.6	1.95	2.6
G	-1.6	90	-4.1	-4.5	0.90	0.4
G	-1.4	70	-5.7	-4.3	0.00	-1.4

TABLE 4.23: SUMMARY OF RESULTS OF FROSTING EXPERIMENTS PERFORMED ON THE 30m EVAPORATOR WITH FAN B AT 900 rev/min

CHAPTER 5 COMPONENT SELECTION

This chapter deals with the selection of the compressor, condenser and expansion valve for the heat pump. From an examination of the performance data on commercially available, hermetically-sealed, reciprocating compressors a model was selected for detailed testing. A purpose built experimental facility, the Component Test Rig (CTR), was assembled for this purpose. The design requirements of the condenser are discussed. The experimental evaluation of the condensers had two stages; a water side pressure drop assessment which removed from further consideration those models with excessive pressure drop, and a thermal performance evaluation. The superheat characteristics of a thermostatic expansion valve were also measured.

1 COMPONENT TESTING

1.1 Component Test Rig

The design objective for the CTR was to provide a means of evaluating the performance of a heat pump compressor and condenser under realistic operating conditions. To this end a complete heat pump refrigerant circuit was assembled, composed of circuit components and pipe diameters proposed in the preliminary design study. These are similar, and in some cases identical, to those eventually selected for the refrigerant circuit of the prototype heat pump. However, as a convenient simplification, the CTR has a liquid source evaporator in order to avoid the complications of dehumidification and freezing effects associated with an air source evaporator, and to facilitate the measurement of heat source fluid flow rate and temperature difference.

The CTR is shown in diagrammatic form in figure 5.1, and in the photographs of figure 5.2. The test rig consists of three circuits; the heat pump circuit, the heat source circuit and the heat sink system.

The minor components of the heat pump circuit are described in Chapter 6. The expansion valve was a cross charged, externally equalized, thermostatic type, manufactured by Sporlan, model GVE-2-CP100. As a liquid source evaporator was used the circuit did not include a defrosting system or a suction accumulator. All components and pipework in the heat pump circuit were thoroughly thermally insulated with 'Armaflex' material.

The heat source liquid was a 37 per cent by weight ethanediol solution in water which has a freezing point of about -20°C (68). This liquid was circulated in counterflow through the outer tube of the coaxial tube evaporator. Heat was provided by electric immersion heaters in the heat source liquid storage tank. A variable voltage control was used to adjust the heat output from these heaters.

The heat sink was a three circuit system identical in principle to the ETR heat sink arrangement, described in the previous chapter. The difference in the CTR case was that the temperature control system was a direct mechanical control rather than electronic. A temperature sensor in the primary circuit at the condenser inlet adjusted the position of the secondary circuit 3 way diverter valve by a vapour pressure linkage. The sensor had a rotary control enabling a range of water inlet temperatures to be set.

1.2 Instrumentation

The temperature and pressure of the refrigerant were measured at four positions in the circuit using stainless steel sheathed platinum resistance thermometers inserted directly into the refrigerant flow path and strain gauge pressure transducers. The measuring points were the inlet and outlet of the evaporator, and the inlet and outlet of the condenser, see figure 5.1. Additional flat PRT's were mounted on the outside surface of the refrigerant pipework to monitor details of the refrigerant temperature distribution, particularly at the flow meter, expansion valve temperature sensing bulb and compressor outlet positions.

Metering the flow rate of the refrigerant presented difficulties. Initially a turbine type flow meter was used, but persistent problems with the reliability of this transducer eventually led to the adoption of a different type of flow meter, a pelton wheel type, manufactured by Litre Meter Ltd. Even this unit suffered a failure during its commissioning as a result of a membrane deforming during circuit evacuation, but this fault was remedied by the manufacturer and it has subsequently operated satisfactorily. An additional difficulty was the possibility of two phase flow through this transducer. Since it measured the volumetric flow rate, the presence of even a small quantity of vapour in a predominantly liquid stream could have led to a considerable increase in the indicated flow rate. The precautions taken to overcome this problem included the positioning of the transducer in the location which had the greatest probability of single phase liquid flow, that is in the liquid line downstream of the receiver, installing a sight glass at the receiver outlet so that the presence of entrained vapour could be observed, and the measurement of the pipewall temperature immediately upstream of the meter. This temperature was compared with the saturated temperature corresponding to the measured pressure to determine the likely state of the metered fluid. If either the fluid was thought to be saturated, or if vapour bubbles were observed, the flow meter reading was disregarded.

The heat source liquid circuit contained flow rate and temperature instrumentation. The temperature difference between the evaporator inlet and outlet was measured by a matched pair of stainless steel sheathed PRT's immersed in the liquid flow. The flow rate was measured by a pelton wheel type flow meter.

Similar temperature and flow rate instrumentation were installed in the primary heat sink circuit.

The flow meters provided an electrical pulse output to digital displays. The thermometers were connected in three wire mode through a manually operated channel selector unit to a digital display. The pressure transducers were connected through a multi-channel signal conditioning and selector unit to a digital display. A chart recorder monitored evaporator and condenser refrigerant pressures to identify when steady-state conditions had been reached. The compressor electrical power consumption was measured both by a precision digital watt meter and a kWh meter to provide a high level of confidence in this important measurement.

1.3 Experimental Method

The procedure followed for each steady-state test run is outlined below:

a) Pre-heat the heat sink water to approximately the desired temperature using the immersion heaters.

b) Adjust the heat sink temperature control to provide the desired condenser inlet temperature.

c) Start the compressor and all cirulation pumps and adjust the flow regulation valves of the heat source and heat sink primary circuits to produce the desired flow rates.

d) Energize and adjust the voltage to the heat source immersion heaters to produce the desired heat source evaporator inlet temperature.

e) As the heat source, heat pump and heat sink circuits approach thermal equilibrium, make further adjustments to the heat source heater voltage and heat sink temperature control valve as necessary.

f) At steady-state conditions with the correct inlet conditions, manually log all pressure, temperature, flow rate and power readings.

Because of the thermal linkages between the source, heat pump and sink systems, and the high heat capacities of the source and sink circuits, considerable time was required to establish thermal equilibrium for the test rig as a whole. It was not unusual to take 8 hours or more to accomplish steps a) to f).

1.4 Data Processing

The raw data is processed to determine the heat and work transfer rates of the cycle and the compressor volumetric and isentropic efficiencies. The detailed measurements made on this test rig provide two independent methods of calculating the heat transfer rates across the two heat exchangers and the compressor power, $\dot{q}_e$, $\dot{q}_c$, $\dot{W}_c$. Therefore it is useful to introduce a notation to distinguish between the two methods. The first method is termed the internal measurements, denoted by the second subscript i, e.g. $\dot{W}_{c,i}$, which are obtained from the refrigerant mass flow rate and specific enthalpy changes across the major components. The second method is termed the external measurements, denoted by the second subscript e, which are obtained from measurements taken external to the refrigerant circuit. For example, the externally measured compressor power, $\dot{W}_{c,e}$, is the measured electrical power absorbed by the compressor motor.

The calculation details are set out below.

a) Externally measured condenser heat output rate, $\dot{q}_{c,e}$

This is determined from:

$$\dot{q}_{c,e} = \rho_w \dot{V}_w Cp_w (T_{w2} - T_{w1})$$

where the properties ρ_w and Cp_w apply to pure water at a temperature of $\frac{1}{2}(T_{w2} + T_{w1})$.

b) Externally measured evaporator heat extraction rate $q_{e,e}$

As above the equation:

$$\dot{q}_{e,e} = \rho_g \dot{V}_g Cp_g (T_{g1} - T_{g2})$$

is applied, where ρ_g and Cp_g are obtained from the published property data on ethanediol solutions (68).

c) Refrigerant cycle

The state points and properties of the refrigerant are calculated as shown below, with the following assumptions:

(i) Pure R22 is assumed at all times. No allowance is made for the modification of the properties of the refrigerant due to the presence of lubricating oil, water, air or other contaminents.

(ii) Isenthalpic expansion from P_3 to P_4 is assumed, enabling the state point at the entry to the evaporator to be determined.

The thermodynamic properties are determined from the measured pressures and temperatures with the aid of the refrigerant properties computer sub-routine as described in Chapter 3. The refrigerant mass flow rate is deduced directly from the measured volumetric flow rate and the liquid density corresponding to state point 3;

$$\dot{m}_r = \rho_3 \dot{V}_r$$

The compressor volumetric efficiency is calculated from the mass flow rate and the vapour density at the suction line, state point 1;

$$\eta_v = \frac{\dot{m}_r}{\rho_1} \times \frac{1}{\dot{V}_{sw}}$$

where $\dot{V}_{sw}$ is the rated displacement rate of the compressor.

Two methods are employed to calculate the isentropic efficiency of the compressor. The first is based solely on the measured pressures and temperatures of the fluid before and after compression:

$$\eta_{is,i} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

where h_1 and h_2 are the specific enthalpies of the fluid before and after compression and h_{2s} is the specific enthalpy assuming ientropic compression from state 1 to pressure P_2. This is termed the internally measured isentropic efficiency. The second method is based on the measured instantaneous power consumption of the compressor motor, $\dot{W}_c$;

$$\eta_{is,e} = \dot{m}_r \frac{(h_{2s} - h_1)}{\dot{W}_c}$$

This is termed the externally measured isentropic efficiency.

The refrigerant mass flow rate and state point measurements are used to determine the internally measured heat and work transfer rates;

compressor $\dot{W}_{c,i} = m_r (h_2 - h_1)$
condenser $\dot{q}_{c,i} = m_r (h_2 - h_3)$
evaporator $\dot{q}_{e,i} = m_r (h_1 - h_4)$

As the transducers and data processing methods are similar to those used in the evaporator experimental programme, the experimental errors are similar to those described in Chapter 4, section 4. The external measurement of the heat transfer rate across the evaporator, $\dot{q}_{e,e}$, is confined to this present set of experiments, and carries an uncertainty of $\pm$ 8 per cent.

2 COMPRESSOR SELECTION

2.1 Design Requirements and Available Types

The compressor is at the heart of the vapour compression cycle and its performance largely dictates the performance of the complete system. Clearly a high overall isentropic efficiency is a primary design requirement. The volumetric efficiency is also indicative of the inherent losses in the compressor flow paths and may be used as a basis of selection. The other, specific design requirements for a residential heat pump compressor are:

- low cost
- high reliability, particularly at high pressure ratios
- low noise
- single phase operation

There are four basic types of compressor applied to vapour compression systems; reciprocating, rotary, screw and centrifugal. As a rough guide the first two types are applied to small and medium capacity duties and the last two are suitable for larger capacities. The reciprocating compressor is the workhorse of the refrigeration industry. It is produced in large quantities which reduces cost, and is regarded as efficient and reliable. It is best suited for small displacement, high pressure ratio applications, of which this present study is an example.

The reciprocating compressor is available in three forms related to the drive arrangement. The open type consists simply of the compressor, with an extension of the crankshaft passing through rotary seals in the casing. The drive may be directly coupled or transmitted through belts or gearing. Even though designers have continually developed better seals, piercing the casing always represents a possible source of leakage. Open compressors are usually of high displacement or applied in situations where the flexibility of the drive arrangement is an advantage.

In an hermetic compressor the problem of leakage is avoided by housing the compressor and drive motor in the same case. The motor rotor is usually directly coupled to the crankshaft. As the refrigerant is in contact with the motor windings a special varnish is applied to prevent short circuiting. Motor cooling is usually achieved by heat transfer to the incoming refrigerant vapour, although some designs require the hermetic case to be air cooled. The hermetic compressor is widely applied to low capacity refrigeration and air conditioning systems. In the event of a failure of either the motor or the compressor, the complete sealed unit must be replaced.

The semi-hermetic compressor is a variant of the above which by virtue of the bolted casing allows access to the motor or compressor for servicing. Due to the more complex mechanical design of the casing this type of compressor is more expensive than an equivalent hermetic unit.

The hermetically sealed, reciprocating type compressor, with the specific advantages of low cost and high reliability, was regarded as the most suitable type for the present application.

2.2 Survey of Commercially Available Compressors

Manufacturers differ widely in the detail and relevance in the presentation of technical performance data on their compressors. For example, evaporator heat extraction rates and condenser heat output rates are often quoted for a cycle containing unrealistically high levels of liquid subcooling and suction superheat. To provide a common basis of comparison, the tabular and graphical information supplied by most of the major European compressor manufacturers were manipulated to determine the volumetric and isentropic efficiencies for each compressor. Isentropic efficiency was calculated from the specified electrical power input, i.e. the externally measured version, as defined in section 1.4. The compressors examined can operate on a single phase, 240 V supply and provide approximately the desired duty. The refrigerant used in the comparison is R22.

The results are summarized in table 5.1. The performance of the compressor chosen for experimental evaluation, the Maneurop MT32 JF5 is shown alongside the entries for each competitive compressor. All but one of the compressors considered are the hermetically sealed reciprocating type. The L'Unite Hermetique model AH8538E was a newly developed product and was not available in time for the experimental evaluation. The data entries for this model were derived from provisional performance data for the UK selling agent (70). The high isentropic efficiency of this compressor is noted. The Danfoss SC15/15D is a twin compressor arrangement, in order to increase the refrigerant throughput. However, the resulting capacity is still rather low for the heat pump system. The DWM Copeland CRGI-0250PFJ provides a performance similar the Maneurop MT32 JF5, and may provide a useful second source of supply if the latter compressor is adopted for the production version of the heat pump. The Prestcold LG300/0042 semi-hermetic design has a capacity greater than the present requirements, although it offers good volumetric and isentropic efficiencies.

The Maneurop MT32 JF5 compressor was selected for the experimental study because it was presented as a model specifically designed for heat pump duty with claimed reliability at high pressure ratio, coupled with a favourable price and reasonable, though not superior performance.

2.3 Design Details of the Selected Compressor

The Maneurop MT32 JF5 compressor is a single cylinder, single acting hermetically sealed reciprocating type, driven by a 2 pole, split phase induction motor. The manufacturer highlights the following design features (75):

- Insensitive to liquid slugging. The outer case acts as a suction accumulator in the event of liquid return to the compressor. The hot gas from the discharge port circulates through a simple heat exchanger in the sump of the compressor in order to evaporate any liquid present.

- The motor is wholly suction gas cooled, which obviates the need for external cooling of the case and enables the useful recovery of the heat generated by the motor.

- Operation at high pressure ratio is possible due to the bearing and valve plate design.

The design and application of this compressor is further discussed in Chapter 6.

2.4 Performance Test Results

The results of 26 steady-state test runs are presented in table 5.2. Throughout the tests the volumetric flow rates of the heat source and heat sink liquids were held constant at nominally 2.6 and 1.0 m^3/h respectively. The temperatures of the heat source and sink liquids were varied to encompass all likely operating conditions of a domestic air to water heat pump. The source liquid temperature at the inlet to the evaporator, T_{g1}, was controlled to the following nominal values; -5°C, 0°C, 5°C, 10°C and 15°C. At each source temperature the heat sink system was adjusted to provide condenser outlet temperatures, T_{w2}, in the range 35°C to 60°C. Table 5.2 groups the test runs in sets of equal T_{g1}. A minimum of four condenser outlet temperatures were applied to each source temperature, and additional test runs were carried out at source temperatures of particular interest to the domestic application, e.g. at 0°C and 5°C.

2.4.1 Volumetric Efficiency

The measured volumetric efficiency of the compressor is plotted against absolute pressure ratio (P_2/P_1) in figure 5.3. The highest volumetric efficiency observed was 0.79 at a pressure ratio of 2.18, run 23, and the efficiency declines with increasing pressure ratio to 0.55 at P_2/P_1 = 6.25, run 4. For a typical heat pump duty of T_e = -5°C and T_c = 50°C the volumetric efficiency is about 0.63. The straight line fitted to the data points has the equation:

$$\eta_v = 0.8975 - 0.05625 \left(\frac{P_2}{P_1} \right)$$

2.4.2 Isentropic Efficiency

The internally and externally measured compressor isentropic efficiencies are plotted against absolute pressure ratio in figure 5.4. For every test run the internally measured isentropic efficiency, based on the observed specific enthalpy change of the working fluid through the compressor, was higher than the externally measured value, which is calculated from the observed fluid conditions at the inlet to the compressor and the measured electrical power input to the compressor motor. The reason for the discrepancy lies in the difference between the internally measured work transfer rate to the fluid by the compressor and the actual compressor motor power input, and this is explored in detail in section 2.4.4. For the purposes of understanding the compressor performance and inferring the economic benefits of the heat pump system, it is the externally defined efficiency which is relevent. As seen from the figure this parameter varies from 0.44 to 0.50 across the range of application. $\eta_{is,e}$ shows little variation with (P_2/P_1) but does exhibit a trend towards higher efficiencies at high suction pressures. The straight line fitted through the $\eta_{is,e}$ points has the equation:

$$\eta_{is,e} = 0.540 - 0.015 \left(\frac{P_2}{P_1} \right)$$

2.4.3 Heat and Work Transfer Rates and COP_h

The externally measured heat and work transfer rates and COP_h are plotted against evaporating temperature with the condensing temperature the parameter in figure 5.5. Since the experimental data contains small variations about the nominal target conditions in both T_e and T_c, the following procedure was adopted in the preparation of figure 5.5. Since the variation in T_e is in general less than that in T_c, the measured heat and work transfer rates were initially plotted against T_c, with T_e the parameter. The arithmetic mean of the evaporating temperatures within each grouping of test runs was taken as the representative evaporating temperature. This plot was then used to read off the heat and work transfer rates for the three condensing temperatures of 40, 50 and 60°C, and these results were replotted against mean evaporating temperature in figure 5.5 Also shown in the figure is the actual deviation in evaporating temperature for each grouping of results; the deviation is defined simply as the difference between the maximum and minimum values in each grouping.

The performance data supplied by the compressor manufacturer is also shown in the figure. These data have been corrected for the effects of the high suction superheat and liquid subcooling included in the original data and refer to the test condition of 5 deg C suction superheat and zero subcooling. The results are in reasonable agreement with the heat output rate data and suggest that the manufacturer's data are conservative in this respect. However, the compressor motor power levels are higher than expected at all operating conditions, by typically 7 per cent. The overall effect is to slightly reduce the COP_h of the cycle from that predicted by the manufacturer's data.

2.4.4 Energy Balance Discrepancy in Compressor Results

The results presented in table 5.2 show a discrepancy between the externally measured electrical consumption of the compressor motor and the enthalpy rise of the working fluid. The discrepancy persisted in spite of an exhaustive transducer and instrumentation re-calibration exercise. Considering a control volume to be constructed around the compressor, intersecting the pipework at the measuring stations 1 and 2, the following equation reflects the energy balance:

$$\dot{W}_{c,e} - \dot{q}_L = \dot{m}_r \left\{ \left(h_2 + \frac{u_2^2}{2} + \rho Z_2\right) - \left(h_1 + \frac{u_1^2}{2} - \rho Z_1\right) \right\} + \dot{E}_L$$

where $\dot{q}_L$ is the heat transferred from the control volume through the compressor casing to the environment.

$\dot{E}_L$ encompasses all the other energy transfers to the environment; noise, electromagnetic radiation etc.

$\dot{q}_L$ is estimated to be about 150 W. $\dot{E}_L$ is negligible. The changes in potential and kinetic energies of working fluid are also negligible. Thus the energy balance simplifies to:

$$\dot{W}_{c,e} - \dot{q}_L = \dot{m}_r (h_2 - h_1)$$

Taking run number 9 as representative of typical heat pump operating conditions gives $\dot{W}_{c,e}$ = 2540 W and $\dot{m}_r (h_2 - h_1)$ = 2150 W. Hence once $\dot{q}_L$ has been considered a discrepancy of 240 W remains. The experimental uncertainties are estimated to be $\pm$ 50 W on $\dot{W}_{c,e}$ and $\pm$ 90 W on $\dot{m}_r (h_2 - h_1)$, thus even the worst case combination of the errors cannot explain the discrepancy.

A possible explanation for this lies in the effect of the lubricating oil on the thermodynamic properties of the working fluid. R22 is partially soluble in the naphthenic type oil supplied in the compressor (the oil is based on Shell Clavus 32 (76)). Thus the fluid returning to the compressor is a mixture of superheated refrigerant vapour and oil containing dissolved refrigerant liquid. As some of the refrigerant is in liquid phase the enthalpy of the mixture is less than that of a pure refrigerant vapour, hence the true enthalpy change across the compressor is greater than was assumed for pure R22.

A sample calculation illustrates this effect; the data presented are taken from test run 9. The calculation methodology is taken from Hughes et al, 1982 (77). At state point 1, the suction line, the specific enthalpy of the mixture can be written as:

$$h_{mix} = Z\, h_{liq} + (1 - Z)\, h_{rvap}$$

where Z = liquid fraction in total mixture

$$= \frac{\text{mass of liquid refrigerant + oil}}{\text{total mass of refrigerant + oil}}$$

h_{liq} = enthalpy of the liquid mixture

$= w\, h_{rliq} + (1 - w)\, h_{oil}$

w = refrigerant fraction in the liquid mixture

$$= \frac{\text{mass of refrigerant liquid in solution}}{\text{mass of refrigerant liquid + oil}}$$

h_{rliq} = specific enthalpy of the refrigerant liquid.

Strictly h_{mix} should also contain the heat of solution of the refrigerant in the oil, but this has been shown to be small compared with the other terms (78). The final mixture definition is:

X = oil fraction in the total mixture

$$= \frac{\text{mass of oil}}{\text{total mass of refrigerant + oil}}$$

$$\text{thus } Z = \frac{X}{1 - w}$$

The oil fraction in a heat pump circuit with a reciprocating compressor is typically less than 5 per cent (79). Thus, in the absence of direct measurements on the present system, put X = 0.04 for the sample calculation. ASHRAE publish charts on refrigerant/oil concentration as a function of temperature and pressure for various refrigerants and oil. Extrapolating the data shown in figure 11, p311 of the 1969 Equipment Volume, to the required temperature of -1.4°C, indicates the concentration of R22 in a naphthenic 150 Saybolt Universal viscosity oil is about 13.7 per cent by weight (80).

$$\text{Thus } Z = \frac{0.04}{1 - 0.137} = 0.046$$

The specific enthalpy of the oil is obtained from the following equation (77):

$$h_{oil} = 67.12 + 1.754\, T_1 + 0.0019\, T_1^2$$

$$= 64.67 \text{ kJ/kg at } T_1 = -1.4°\text{C}$$

In this equation the reference temperature for the specific enthalpy of oil is -40°C in order to be consistent with the refrigerant enthalpy.

The specific enthalpy of the refrigerant liquid is obtained by extrapolation from the subcooled liquid value to the higher temperature:

h_{rliq} = 42.60 kJ/kg

The h_{rvap} is obtained directly from charts or programs;

h_{rvap} = 250.86 kJ/kg

$$\text{Thus } h_{liq} = 0.137 \times 42.60 + (1-0.137) \times 64.67$$
$$= 61.65 \text{ kJ/kg}$$
$$\text{and } h_{mix} = 0.046 \times 61.65 + (1-0.046) \times 250.86$$
$$= 242.16 \text{ kJ/kg}$$

Hence the specific enthalpy of the mixture at point 1 is 8.70 kJ/kg less than that of the pure R22 vapour, due to the presence of liquid refrigerant dissolved in the oil. Assuming the properties at point 2, compressor discharge, are unaffected by the presence of oil, then h_2 = 329.41 kJ/kg. The change of specific enthalpy across the compressor is 78.55 kJ/kg for pure R22 and 87.25 kJ/kg when the effects of oil are included. This corresponds to an additional power requirement of 235 W, which matches the 240 W discrepancy. Thus this sample calculation demonstrates that the existance of refrigerant liquid in entrained oil droplets in the flow entering the compressor is a plausible explanation of the energy balance discrepancy in the compressor results.

3. CONDENSER SELECTION

3.1 Design Requirements and Equipment Types

The role of the condenser is to transfer the heat collected by the evaporator and the waste heat recovered from the compressor to the heating water. The condensing duty ranges from about 4 kW at low air temperature and high water temperature conditions, to 9.7 kW at high air and low water temperature conditions, see figure 5.5. A typical duty, at T_e = -5°C and T_c = 50°C is 6.1 kW. In order to maintain an attractive COP_h this heat must be transferred across a small refrigerant to water temperature difference, requiring a large heat exchange area. However, capital cost requirements naturally limit the size of a practical condenser. An important requirement for a domestic heat pump condenser is to offer low water side resistance. This is because:

a) In order to enhance water side heat transfer and to produce a high mean radiator temperature for a given flow temperature, a high water flow rate is required.

b) The additional lengths of water piping to the heat pump and flow through the supplementary heater increase the frictional pressure drop of the water flow.

c) The desire to employ a single, conventional domestic central heating circulating pump in the heat pump installation, in order to minimize the costs to the customer.

The required water flow rate is in the region of 0.7 to 1.0 m^3/h. Conventional central heating practice centres on setting the flow rate to attain a difference between flow and return temperatures of 10 deg C, hence for a 6 kW nominal load a flow rate of 0.52 m^3/h would be selected (81). Thus good practice for a heat pump installation results in a flow rate of 34 per cent to 92 per cent higher than conventional heating systems. At a flow rate of 0.85 m^3/h, a low cost domestic central heating circulator designed for high resistance duties develops about 50 kN/m^2 of dynamic pressure, see for example reference 82. The frictional pressure drop depends on the design of the pipework and radiator system and will vary from one installation to another, but it is wise to absorb no greater than about 10 kN/m^2 in pressure drop within the heat pump itself. There is some evidence to suggest the water side pressure drop on currently available heat pumps is excessive. Difficulty in achieving satisfactory flow rates in some field trial installations led the Electricity Council to recommend the inclusion of a second circulating pump in heat pump systems (83). The present approach is to obviate the second circulator by selecting a condenser with a low water side pressure drop.

The shell and tube type condenser is widely used throughout the refrigeration industry when water is the coolant. The refrigerant is typically directed to the shell side of the exchanger to mitigate the effects of condensate flooding the heat transfer surfaces and to facilitate the cleaning of the water passages which are prone to fouling. Two variants of the shell and tube heat exchanger have been developed for small systems and are considered further here. The coaxial tube condenser is a single tube, single pass arrangement in which an externally finned tube is sealed inside a larger diameter plain tube. Water is circulated through the inner tube and the refrigerant flows in counter flow in the annular space. The condensate collects in the bottom of the outer tube and flows towards the outlet. This method of construction is comparatively inexpensive, as off-the-shelf tubing is employed, rather than specially constructed pressure vessels, and removable end plates are not required. A number of manufacturers supply this type of condenser and three models from two suppliers were considered in the experimental evaluation. The shell and coil condenser consists of a closely wound helical tube enclosed within a steel shell. The coolant flows through the tube in a single pass while the refrigerant condenses on the outside of the tube and collects in the sump of the shell. This design offers compactness, both from the compact nature of the heat exchanger itself and because a separate refrigerant liquid receiver is not needed. An example of this type of condenser was included in the experimental evaluation.

3.2 Design Details of the Test Condensers

Table 5.3 summarizes the details of the water passages and overall dimensions of the test condensers. The Wieland KWG 3X condenser was used on the CTR throughout the compressor performance tests, and on the ETR for the evaporator experiments. According to its manufacturer, this model was introduced to complete a range of condenser sizes, and has the specific feature of lower water side pressure drop (84). It appears to be designed particularly for heat pump systems.

The two condensers supplied by IMI Yorkshire Imperial Alloys are taken from the range of 'Yorco-ax' coaxial tube heat exchangers (85). The inside tube of each heat exchanger is formed from the company's low finned 'Integron' heat transfer tubing (86). A modified version of the

CK8-20 condenser was installed in the prototype heat pump, hence additional detail on this model is presented in Chapter 6.

The shell and coil type condenser was supplied by the Standard Refrigeration Company of the USA, who manufacture a range of condensers of this type (87).

3.3 Water Side Pressure Drop Tests and Test Results

The test apparatus is shown in figure 5.6 The frictional pressure drop across each test condenser was measured using a mercury well-type manometer. The manometer scale is graduated to account for the area correction factor, thus the height noted was the displacement of the column from the zero flow condition. Fluid pressure was transmitted to the manometer by water filled flexible tubes. The water flow rate was measured using a calibrated pelton wheel type flow meter and varied in the range 0.2 to 1.4 m^3/h by means of a manual regulating valve. The water temperature was measured by a PRT and maintained in the range 47 to 52°C by electrical immersion heaters.

The results are contained in table 5.4 and plotted as figure 5.7. Two sets of results are presented for the Wieland condenser, 'as received' and after 966 hours of operation. The condenser was retested after this period of service in the CTR because a layer of scale had developed throughout the length of the water tubing. The thickness of the scale was 0.2 to 0.3 mm and it consisted of the carbonates of calcium, magnesium and iron. These ions are present in the site water supply and fouling of heat exchange equipment using the supply is a recognised problem on the site. However, the deposition rate may also have been increased by the cooling method initially employed on the CTR. This involved a direct connection between the condenser and a small dedicated cooling tower. Due to evaporation in the cooling tower frequent topping up with fresh water was required. Later this arrangement was revised and an intermediate water to water heat exchanger was added, as described in section 1.1, which removed the need for continuously introducing fresh water to the primary heat sink circuit. The results show that the scaling increased the pressure drop by 47 per cent at 1 m^3/h.

From the results of the 'as received' condensers it is clear than the Standard J-201 and IMI YIA CK3-10 models have unacceptably high pressure drops for the heat pump application. Thus only the Wieland KWG 3X and IMI YIA CK8-20 models were included in the thermal performance evaluation.

The relative roughness of the water side passages, as defined in Chapter 3, is calculated from figure 5.7 for the KWG 3X and CK8-20 condensers at 1 m^3/h to be 2.128 and 1.021 respectively.

3.4 Thermal Performance Tests and Test Results

Prior to the thermal performance evaluation the Wieland KWG 3X condenser was chemically treated to remove the scale deposits. The IMI YIA CK8-20 was tested in 'as received' condition. Each condenser was in turn installed in the CTR in order to measure the heat transfer characteristics for four water flow rates covering the range 0.6 to 1.2 m^3/h, and at each flow rate 4 different evaporatoring temperatures were used. The water outlet temperature was held constant at 55°C through the test series. The procedure adopted is summarised below.

a) Follow the CTR start up procedure a) to c) as described in section 1.3.

b) Set the heat source heater battery to a fixed output.

c) Adjust the heat sink temperature control to provide a water outlet temperture of 55°C. Monitor T_{w2} as equilibrium conditions are approached and make further adjustments as necessary.

d) At steady-state operation record all temperature, pressure, flow rate and power readings.

e) Adjust the water flow rate to the next desired setting and repeat steps c) to e).

f) Once the results for all four flow rates have been acquired, adjust the heat source heater to change the evaporating temperature and thus the capacity of the heat pump system, and repeat steps c) to e).

For completeness the thermal performance of the KWG 3X condenser was also measured before the scaling was removed, although these tests did not rigorously follow the procedure and conditions described above. Instead of maintaining a constant water outlet temperature of 55°C, the condensing temperature was fixed at 57°C, allowing T_{w2} to vary slightly.

The test results are presented in tables 5.5, 5.6 and 5.7 covering the KWG 3X unit before and after descaling and the CK8-20 condenser respectively. Throughout the tests the level of liquid subcooling occuring in the condenser was small, less than 1 deg C in most cases. The water flow rates measured for the KWG 3X after descaling deviate slightly from the target values of 0.6, 0.8, 1.0 and 1.2 m^3/h, hence the following graphical procedure was used in the construction of the thermal performance characteristic:

a) For each result the condenser temperature difference $T_c - T_{w1}$ and the externally measured heat transfer rate $\dot{q}_{c,e}$ were plotted against water flow rate, see figures 5.8 a) and b).

b) The thermal performance points ($\dot{q}_{c,e}$, $T_c - T_{w1}$) were read from these figures for the desired water flow rates.

The results are shown in figure 5.9. Results for the same condenser in the scaled condition are also shown. From this it is clear that water side fouling has increased the overall heat transfer resistance by about 56 per cent. Thus, for a given water inlet temperature and a nominal heat transfer rate of 6 kW, the water side fouling leads to 3.1 deg C increase in the mean condensing temperature.

The heat transfer characteristic of the CK8-20 condenser is shown in figure 5.10. At a flow rate of 1.0 m^3/h and a nominal condensing capacity of 6 kW, the CK8-20 condenser requires a lower temperature difference by 0.35 deg C compared with the KWG 3X. This benefit is attributable to the greater heat transfer area of the CK8-20 unit. An estimation of the overall heat transfer coefficient may be obtained by considering the simple condensing to refrigerant temperature difference $T_c - T_{w1}$, and referring the result to the water side area. The results for 1 m^3/h are; KWG 3X = 2060 W/m^2K, CK8-20 = 1920 W/m^2K. The smaller condenser produces a higher coefficient due to the increased water side Reynolds number at a given volumetric flow rate.

The benefits to the thermodynamic performance of the heat pump of the larger condenser are illustrated in figure 5.11. The measured heat output rate and COP_h are plotted against evaporating temperature for the two condensers operating at a water flow rate of 1.0 m^3/h and a water discharge temperature of 55°C. At T_e = -5°C the CK8-20 condenser increases both the heat output rate and COP_h by about 3 per cent.

The test results for the descaled KWG 3X and the CK8-20 condensers indicate a water outlet temperature close to, and in some cases greater than, the mean condensing temperature of the working fluid. This apparent anomaly is explained by the desuperheating function of the condenser. Figure 5.12 illustrates the point with a simplified heat exchange diagram constructed from the data from test run Y10. About 35 per cent of the heat transferred in the condenser is vapour desuperheating. The high temperature difference in the desuperheating region results in a temperature lift in the water stream of about 2 deg C. The balance of the heat transfer occurs in the condensing region (as previously described there is no appreciable liquid subcooling) in which the high refrigerant side coefficient results in small temperature differences across the exchanger.

4. EXPANSION VALVE PERFORMANCE

The data presented for the compressor test is used to illustrate the performance of the expansion valve, see table 5.2. Figure 5.13 shows the measured superheat at the evaporator outlet as functions of the mean evaporating temperature and the refrigerant mass flow rate. The results indicate a general trend towards increased superheat at higher evaporating temperatures and at lower condensing temperatures. This can be explained by the response of the valve to increases in the required refrigerant mass flow through the system. If, due to either an increased in T_e or a decrease in T_c, the mass flow rate is increased, the degree of opening of the valve must increase to accommodate the greater flow. As the degree of opening is governed by the suction superheat, the superheat must rise to bring about the required increase in value capacity. The relationship between suction superheat and the refrigerant mass flow rate is demonstrated by figure 5.13 b).

The scatter in the data reflects the fact that the valve is somewhat imprecise in its operation, which may be expected from the nature of the valve.

5. CONCLUSIONS

1. An hermetic, reciprocating compressor has been selected from a survey of commercially available compressors. The measured volumetric performance is good, although the isentropic efficiency is low at 0.44 to 0.50.

2. Detailed thermal performance tests were conducted on two coaxial tube heat exchangers, and water side pressure drop tests carried out on a further two models. The CK8-20 unit was selected primarily because of its low pressure drop. This condenser also provides a superior thermal performance, although it is recognised that this alone does not justify the increased component cost.

3. Water side fouling was found to be a problem with the test rig. While accepting that the high scale deposition rate was influenced by the operation of the heat sink cooling tower, and that this rate is unlikely to be reproduced in a leak-tight domestic central heating system, it is none-the-less a salutary indication of possible long term problems in the field.

Run No.	Heat Source Liquid (ethanediol) T_{g1} (°C)	T_{g2} (°C)	$\dot{V}_g$ (m^3/h)	Heat Sink (water) T_{w1} (°C)	T_{w2} (°C)	$\dot{V}_w$ (m^3/h)	Refrigerant State Points (R22) T_1 (°C)	T_2 (°C)	T_3 (°C)	T_4 (°C)	P_1 (bar, absolute)	P_2	P_3	P_4	$\dot{V}_r$ (l/min)
1	-4.8	-6.3	2.57	28.1	32.9	0.99	-4.8	100.2	33.2	-10.8	3.37	13.09	13.12	3.50	1.27
2	-5.2	-6.7	2.59	34.5	39.1	1.00	-5.5	111.1	39.0	-11.1	3.32	15.13	15.14	3.42	1.23
3	-5.2	-6.3	2.61	45.5	49.8	1.01	-8.4	126.9	49.6	-9.7	3.51	19.36	19.36	3.62	1.29
4	-5.0	-6.1	2.61	52.3	56.4	1.01	-7.4	137.0	55.9	-9.2	3.58	22.39	22.30	3.67	1.23
5	-0.1	-1.9	2.67	29.1	34.7	1.00	-1.1	95.3	34.8	-6.2	3.96	13.77	13.81	4.10	1.56
6	0.3	-1.5	2.67	34.8	40.3	1.01	-1.0	103.2	40.1	-5.4	4.07	15.70	15.73	4.19	1.58
7	0.0	-1.7	2.68	41.3	46.4	1.00	-1.8	113.4	46.6	-5.6	4.04	18.10	18.12	4.17	1.54
8	0.1	-1.4	2.68	47.6	52.7	1.01	-0.9	121.5	52.4	-4.7	4.16	20.71	20.74	4.29	1.58
9	-0.3	-1.8	2.67	49.7	54.7	1.01	-1.4	125.6	54.3	-5.1	4.10	21.61	21.62	4.24	1.55
10	0.2	-1.2	2.69	55.5	60.4	1.01	-0.9	134.0	59.9	-4.3	4.22	24.34	24.35	4.35	1.56
11	5.5	3.3	2.72	29.8	36.6	1.01	3.0	91.3	36.5	-1.2	4.69	14.45	14.43	4.86	1.87
12	5.2	3.1	2.71	34.0	40.6	1.01	4.0	97.1	40.5	-1.1	4.67	15.86	15.86	4.83	1.89
13	5.3	3.2	2.71	38.9	45.3	1.01	4.4	104.3	45.2	-0.8	4.70	17.65	17.64	4.88	1.87
14	5.0	3.1	2.70	43.0	49.2	1.01	4.4	110.2	48.9	-0.6	4.75	19.28	19.30	4.93	1.89
15	5.5	3.6	2.70	43.8	50.1	1.01	4.8	110.4	49.9	-0.2	4.84	19.66	19.63	5.00	1.91
16	5.4	3.5	2.72	47.5	53.6	1.01	4.5	115.8	53.2	0.2	4.89	21.19	21.18	5.05	1.92
17	5.1	3.3	2.69	50.5	56.5	1.01	3.9	120.6	56.1	0.1	4.86	22.54	22.51	5.02	1.91
18	5.4	3.7	2.70	53.5	59.2	1.01	4.7	125.2	58.7	0.6	4.95	23.88	23.85	5.11	1.92
19	10.2	7.7	2.72	33.1	40.5	1.01	9.5	92.4	41.0	3.1	5.38	16.06	16.06	5.57	2.25
20	9.8	7.4	2.71	37.4	44.5	1.01	9.2	98.8	44.7	3.0	5.36	17.56	17.52	5.54	2.19
21	10.1	7.9	2.71	43.8	50.7	1.02	8.9	107.1	50.6	3.8	5.50	20.11	20.10	5.68	2.23
22	9.7	7.8	2.72	50.7	57.3	1.01	9.1	117.1	56.9	4.1	5.56	23.07	23.08	5.68	2.21
23	14.9	11.7	2.74	22.0	30.6	1.01	14.4	78.0	31.8	6.2	5.90	12.86	12.89	6.09	2.53
24	14.9	12.0	2.76	33.8	41.9	1.02	14.4	92.0	42.8	7.4	6.08	16.72	16.74	6.26	2.59
25	15.1	12.5	2.74	39.2	47.3	1.01	14.6	98.8	47.8	7.8	6.20	18.81	18.90	6.41	2.59
26	15.0	12.6	2.76	47.3	54.9	1.02	14.6	109.9	54.9	8.5	6.30	22.13	22.15	6.50	2.59

TABLE 5.2: RESULTS OF PERFORMANCE TESTS ON MANEUROP MT32 JF5 COMPRESSOR (Table continued on next page)

Run No.	Summary of Thermodynamic Cycle					Mass Flow Rates			Externally Measured Heat and Work Transfer Rates					Internally Measured Heat and Work Transfer Rates					
	T_e	T_c	ΔT_{sh}	ΔT_{sc}	P_2/P_1	$\dot{m}_g$	$\dot{m}_w$	$\dot{m}_r$	$\dot{q}_e$	$\dot{q}_c$	$\dot{W}_c$	COP_h	η_{is}	$\dot{q}_e$	$\dot{q}_c$	$\dot{W}_c$	COP_h	η_{is}	η_v
	(°C)	(°C)	(degC)	(degC)	(-)	(kg/s)	(kg/s)	(kg/s)	(kW)	(kW)	(kW)	(-)	(-)	(kW)	(kW)	(kW)	(-)	(-)	(-)
1	-10.9	33.7	6.5	0.5	3.88	0.75	0.28	0.0245	4.00	5.53	1.86	2.97	0.46	4.04	5.65	1.61	3.51	0.54	0.67
2	-11.4	39.5	6.5	0.5	4.56	0.75	0.28	0.0232	4.03	5.33	2.03	2.63	0.45	3.64	5.34	1.70	3.14	0.54	0.65
3	-9.8	49.9	1.8	0.3	5.52	0.76	0.28	0.0233	3.11	5.04	2.28	2.21	0.45	3.27	5.25	1.99	2.64	0.52	0.60
4	-9.4	56.2	2.2	0.3	6.25	0.76	0.28	0.0216	2.98	4.83	2.36	2.05	0.44	2.84	4.81	1.96	2.45	0.52	0.55
5	-6.4	35.7	5.6	0.9	3.48	0.78	0.28	0.0299	4.98	6.59	2.05	3.22	0.47	4.90	6.68	1.78	3.75	0.54	0.70
6	-5.6	41.0	5.0	0.9	3.86	0.78	0.28	0.0296	4.98	6.50	2.19	2.97	0.47	4.63	6.54	1.91	3.43	0.54	0.67
7	-5.8	47.0	4.3	0.6	4.48	0.78	0.28	0.0282	4.59	6.16	2.34	2.64	0.47	4.17	6.18	2.01	3.07	0.55	0.65
8	-4.9	52.8	4.4	0.4	4.98	0.78	0.28	0.0282	4.17	5.88	2.52	2.34	0.47	3.94	6.07	2.13	2.85	0.55	0.63
9	-5.3	54.7	4.3	0.3	5.27	0.78	0.28	0.0270	4.02	5.76	2.54	2.27	0.47	3.74	5.89	2.15	2.74	0.55	0.61
10	-4.5	60.1	4.0	0.2	5.77	0.78	0.28	0.0268	3.77	5.63	2.72	2.07	0.45	3.46	5.70	2.24	2.54	0.55	0.59
11	-1.3	37.5	4.7	1.0	3.08	0.79	0.28	0.0357	6.20	7.83	2.19	3.59	0.47	5.82	7.76	1.94	4.00	0.53	0.71
12	-1.4	41.4	5.9	0.9	3.40	0.79	0.28	0.0354	6.04	7.69	2.31	3.33	0.48	5.60	7.61	2.02	3.78	0.55	0.71
13	-1.2	45.9	6.1	0.7	3.76	0.79	0.28	0.0345	5.76	7.44	2.43	3.06	0.49	5.26	7.37	2.11	3.50	0.56	0.69
14	-0.9	49.7	5.7	0.8	4.06	0.79	0.28	0.0342	5.32	7.14	2.57	2.78	0.49	5.03	7.24	2.21	3.28	0.56	0.67
15	-0.4	50.5	5.6	0.6	4.06	0.79	0.28	0.0345	5.32	7.40	2.59	2.86	0.49	5.03	7.25	2.22	3.27	0.57	0.67
16	0.0	53.8	4.9	0.6	4.33	0.79	0.28	0.0341	5.36	7.03	2.69	2.61	0.48	4.80	7.12	2.31	3.08	0.56	0.65
17	-0.2	56.6	4.5	0.5	4.64	0.78	0.28	0.0335	4.88	6.91	2.78	2.49	0.48	4.57	6.95	2.38	2.92	0.56	0.64
18	0.3	59.2	4.7	0.5	4.82	0.79	0.28	0.0332	4.75	6.68	2.88	2.32	0.47	4.41	6.86	2.44	2.81	0.56	0.62
19	3.0	41.9	7.0	0.9	2.99	0.79	0.28	0.0420	6.91	8.58	2.39	3.60	0.50	6.73	8.85	2.11	4.19	0.56	0.74
20	2.8	45.6	6.8	0.9	3.28	0.79	0.28	0.0404	6.75	8.20	2.48	3.31	0.50	6.26	8.46	2.19	3.86	0.57	0.71
21	3.6	51.5	5.7	0.9	3.66	0.79	0.28	0.0401	6.18	7.99	2.72	2.94	0.49	5.86	8.24	2.38	3.47	0.57	0.68
22	3.8	57.7	5.6	0.8	4.15	0.79	0.28	0.0384	5.50	7.62	2.94	2.59	0.48	5.28	7.79	2.51	3.11	0.57	0.65
23	5.8	33.0	9.0	1.2	2.18	0.80	0.28	0.0490	9.01	10.00	2.13	4.71	0.46	8.56	10.49	1.93	5.45	0.51	0.79
24	6.8	43.6	8.0	0.8	2.75	0.80	0.28	0.0482	8.30	9.53	2.55	3.74	0.50	7.71	9.97	2.26	4.41	0.56	0.75
25	7.5	48.7	7.6	0.9	3.03	0.80	0.28	0.0471	7.39	9.36	2.73	3.43	0.50	7.21	9.60	2.38	4.03	0.57	0.72
26	7.9	55.8	7.1	0.9	3.51	0.80	0.28	0.0456	6.86	8.85	3.03	2.92	0.49	6.51	9.12	2.60	3.50	0.57	0.69

TABLE 5.2: CONTINUED

MANUFACTURER	MODEL	TYPE	WATER SIDE DESIGN DETAILS bore diameter (mm)	pass length (mm)	no of passes	Water side heat exchange area (m^2)	OVERALL diameter (mm)	SIZE Height (mm)
Wieland (West Germany)	KWG3X	coaxial	18.5	7045	1	0.409	395	290
IMI YIA (UK)	CK3-10	coaxial	15.7	6126	1	0.302	320	190
IMI YIA	CK8-20	coaxial	19.7	9800	1	0.607	450	320
Standard Refrigeration Company (USA)	J-201	shell and coil	13	not known	1	not known	143	343

TABLE 5.3: DESIGN DETAILS OF TEST CONDENSERS

$\dot{V}_w$	0.296	0.384	0.567	0.702	0.828	0.944	1.000	1.097	1.185
Δp_w	1.3	2.0	4.1	6.2	8.3	10.4	11.7	13.7	15.6

Water temperature 47°C
Condenser: Wieland KWG 3X, as received

$\dot{V}_w$	0.430	0.537	0.615	0.625	0.750	0.770	0.847	0.857	0.905	0.983
Δp_w	3.1	5.1	6.7	6.9	10.0	10.7	12.9	13.1	14.8	17.6
$\dot{V}_w$	1.075	1.085								
Δp_w	21.4	21.7								

Water temperature 48°C
Condenser: Wieland KWG 3X, 966 hours operation

$\dot{V}_w$	0.375	0.555	0.695	0.79	0.88	0.91	0.935	0.972	1.03
Δp_w	2.8	6.3	9.4	12.5	14.7	15.8	16.5	17.9	20.0

Water temperature 48°C
Condenser: IMI YIA CK3-10

$\dot{V}_w$	0.30	0.45	0.59	0.74	0.90	1.02	1.10	1.18	1.21	1.33
Δp_w	0.4	1.2	2.1	3.2	4.7	5.9	6.9	7.9	8.2	9.8

Water temperature 51°C
Condenser: IMI YIA CK8-20

$\dot{V}_w$	0.40	0.49	0.557	0.595	0.682	0.73	0.81	0.847
Δp_w	7.0	10.4	13.2	15.4	20.0	22.2	27.6	29.8

Water temperature 52°C
Condenser: Standard J-201

$\dot{V}_w$ = Water flow rate in m^3/h
Δp_w = Water side frictional pressure drop in kN/m^2

TABLE 5.4: WATER SIDE PRESSURE DROP RESULTS

Run Number	Heat Sink Liquid (Water)			Refrigerant State Points (R22)					Summary of Thermodynamic Cycle				Externally Measured Heat & Work Transfer Rates			Temp. Difference Across condenser
	T_{w1}	T_{w2}	$\dot{V}_w$	T_2	T_3	P_2	P_3	$\dot{m}_r$	T_e	T_c	ΔT_{sh}	ΔT_{sc}	$\dot{q}_c$	$\dot{W}_c$	COP_h	(T_c-T_{w1})
	(°C)	(°C)	(m^3/h)	(°C)	(°C)	(bar, abs)		(kg/s)	(°C)	(°C)	(degC)	(degC)	(kW)	(kW)	(-)	(degC)
WA1	41.5	53.0	0.62	115.7	56.3	22.76	22.87	0.0395	4.2	57.2	5.8	0.9	8.25	2.96	2.79	15.71
WA2	42.5	52.7	0.64	118.5	55.9	22.61	22.63	0.0363	2.2	56.8	5.4	0.9	7.65	2.86	2.68	14.31
WA3	43.2	52.8	0.66	120.5	55.8	22.46	22.48	0.0340	0.4	56.5	6.3	0.7	7.22	2.80	2.58	13.32
WA4	47.9	56.1	0.60	132.6	57.7	23.21	23.30	0.0257	-5.7	58.0	3.3	0.3	5.69	2.60	2.19	10.13
WA5	49.3	55.7	0.63	142.7	56.8	22.58	22.65	0.0199	-11.3	56.8	3.8	0.0	4.62	2.34	1.97	7.46
WA6	50.8	56.0	0.63	152.9	56.6	22.48	22.54	0.0156	-15.9	56.5	2.4	0.0	3.74	2.13	1.76	5.74
WA7	44.7	53.0	0.83	117.4	57.1	23.22	23.24	0.0388	4.2	58.0	5.4	0.9	7.87	2.98	2.64	13.28
WA8	45.3	52.8	0.83	120.1	56.2	22.69	22.73	0.0343	0.6	56.9	5.0	0.8	7.16	2.83	2.53	11.69
WA9	47.8	53.9	0.83	131.7	56.3	22.53	22.57	0.0258	-6.0	56.6	4.3	0.3	5.70	2.26	2.53	8.83
WA10	50.0	54.9	0.82	142.4	56.5	22.60	22.64	0.0202	-10.8	56.8	4.0	0.3	4.56	2.37	1.93	6.81
WA11	51.9	55.5	0.81	156.5	56.5	22.50	22.54	0.0149	-16.6	56.6	2.4	0.1	3.32	2.09	1.59	4.72
WA12	43.1	51.7	1.00	112.5	57.2	23.29	23.30	0.0481	10.2	58.1	7.9	1.0	9.88	3.18	3.11	15.06
WA13	45.0	51.9	1.01	116.3	56.4	22.83	22.82	0.0395	4.2	57.2	5.8	0.8	7.94	2.96	2.69	12.18
WA14	46.8	52.9	1.01	121.0	56.6	22.90	22.92	0.0343	0.8	57.4	4.4	0.8	7.09	2.84	2.50	10.60
WA15	47.2	52.9	1.01	123.8	56.4	22.75	22.78	0.0314	-1.7	57.1	5.0	0.7	6.61	2.74	2.42	9.9
WA16	53.2	56.1	1.01	155.7	57.3	22.92	22.95	0.0148	-16.4	57.4	2.2	0.1	3.35	2.11	1.59	4.2

TABLE 5.5: CONDENSER THERMAL PERFORMANCE RESULTS. WIELAND KWG 3X BEFORE DESCALING

Run Number	Heat Sink Liquid (water)			Refrigerant State Points (R22)					Summary of Thermodynamic Cycle				Externally Measured Heat & Work Transfer Rates			Temperature Difference Across Condenser
	T_{w1}	T_{w2}	$\dot{V}_w$	T_2	T_3	P_2	P_3	$\dot{m}_r$	T_e	T_c	ΔT_{sh}	ΔT_{sc}	$\dot{q}_c$	$\dot{W}_c$	COP_h	(T_c-T_{w1})
	(°C)	(°C)	(m^3/h)	(°C)	(°C)	(bar, abs)		(kg/s)	(°C)	(°C)	(deg C)	(deg C)	(kW)	(kW)	(-)	(deg C)
WB1	52.1	54.8	1.20	145.0	55.1	21.73	21.78	0.0180	-13.7	55.0	3.4	0.0	3.72	2.21	1.69	2.95
WB2	51.5	54.7	1.03	144.6	54.8	21.61	21.67	0.0177	-14.3	54.8	3.5	0.0	3.82	2.17	1.76	3.32
WB3	50.6	54.7	0.81	144.8	54.6	21.47	21.56	0.0174	-14.5	54.5	4.1	0.0	3.74	2.15	1.74	3.91
WB4	49.8	55.3	0.62	143.6	54.7	21.68	21.76	0.0182	-13.6	54.9	3.0	0.2	3.90	2.20	1.77	5.13
WB5	50.4	54.3	1.20	129.6	54.5	21.57	21.62	0.0248	-7.3	54.7	4.1	0.2	5.31	2.46	2.16	4.27
WB6	49.5	54.2	1.01	129.2	54.3	21.45	21.56	0.0251	-7.2	54.5	4.2	0.2	5.42	2.47	2.19	5.03
WB7	49.1	54.8	0.82	129.4	54.6	21.66	21.73	0.0250	-7.2	54.9	3.6	0.3	5.30	2.47	2.15	5.78
WB8	47.7	55.2	0.62	129.9	54.7	21.69	21.78	0.0252	-7.0	55.0	3.9	0.3	5.33	2.48	2.15	7.32
WB9	49.0	54.3	1.20	115.4	54.7	21.84	21.87	0.0359	1.6	55.2	5.2	0.6	7.24	2.79	2.60	6.21
WB10	47.7	53.8	1.01	115.1	54.0	21.56	21.60	0.0352	0.9	54.6	5.1	0.7	7.07	2.74	2.58	6.99
WB11	46.8	54.1	0.84	116.1	54.1	21.66	21.72	0.0355	1.1	54.9	5.4	0.8	6.96	2.76	2.53	8.12
WB12	44.7	54.4	0.63	115.9	54.2	21.67	21.74	0.0352	1.0	54.9	5.3	0.8	7.01	2.75	2.55	10.20
WB13	47.0	53.4	1.21	107.3	54.0	21.64	21.66	0.0466	8.4	54.8	6.7	0.8	8.87	2.98	2.98	7.79
WB14	46.3	54.0	1.02	108.3	54.7	21.96	22.03	0.0463	8.1	55.5	6.7	0.8	8.97	3.01	2.98	9.20
WB15	44.0	53.5	0.83	108.2	54.1	21.73	21.76	0.0461	8.1	55.0	7.0	0.9	9.01	2.99	3.02	10.98
WB16	41.4	54.0	0.63	108.4	54.4	21.91	21.92	0.0462	8.4	55.3	6.4	0.9	9.08	3.02	3.01	13.93

TABLE 5.6: CONDENSER THERMAL PERFORMANCE RESULTS. WIELAND KWG 3X AFTER DESCALING

Run Number	Heat Sink Liquid (water)			Refrigerant State Points (R22)					Summary of Thermodynamic Cycle				Externally Measured Heat & Work Transfer Rates			Temperature Difference Across Condenser
	T_{w1}	T_{w2}	$\dot{V}_w$	T_2	T_3	P_2	P_3	$\dot{m}_r$	T_e	T_c	ΔT_{sh}	ΔT_{sc}	$\dot{q}_c$	$\dot{W}_c$	COP_h	(T_c-T_{w1})
	(°C)	(°C)	(m^3/h)	(°C)	(°C)	(bar, abs)		(kg/s)	(°C)	(°C)	(deg C)	(deg C)	(kW)	(kW)	(-)	(deg C)
Y1	47.6	54.9	1.20	106.2	55.3	21.92	21.86	0.0482	9.2	55.3	6.2	0	10.05	3.07	3.27	7.74
Y2	46.3	55.0	1.00	106.6	55.3	21.88	21.82	0.0469	9.2	55.2	6.2	0	9.98	3.06	3.27	8.96
Y3	44.4	55.2	0.80	106.9	55.3	21.95	21.84	0.0472	9.3	55.3	6.2	0	9.96	3.05	3.27	10.95
Y4	40.8	55.0	0.60	105.7	54.8	21.71	21.63	0.0470	8.9	54.8	4.9	0	9.78	3.02	3.24	14.03
Y5	49.3	55.0	1.21	114.9	55.2	21.92	21.83	0.0364	2.2	55.3	4.1	0.1	7.87	2.84	2.78	5.96
Y6	48.1	54.8	1.00	114.8	54.9	21.76	21.67	0.0362	1.8	54.9	4.5	0	7.70	2.82	2.74	6.87
Y7	46.7	55.1	0.80	114.9	55.1	21.79	21.70	0.0361	1.9	55.0	3.9	0	7.71	2.81	2.74	8.29
Y8	43.8	55.0	0.60	113.8	54.5	21.54	21.48	0.0362	1.7	54.5	3.4	0	7.71	2.79	2.76	10.70
Y9	50.6	54.8	1.20	127.4	54.9	21.73	21.67	0.0267	-5.7	54.9	4.1	0	5.85	2.56	2.29	4.34
Y10	49.7	54.8	1.00	127.4	54.7	21.66	21.58	0.0265	-5.6	54.7	3.1	0	5.90	2.55	2.32	5.08
Y11	48.7	55.1	0.80	127.5	54.7	21.66	21.57	0.0264	-5.9	54.7	4.7	0	5.92	2.54	2.33	6.06
Y12	46.4	55.1	0.60	126.6	54.3	21.48	21.41	0.0268	-5.6	54.4	2.8	0	5.99	2.55	2.35	7.96
Y13	52.2	55.2	1.20	144.6	55.0	21.81	21.75	0.0178	-14.2	55.1	4.1	0.1	4.13	2.20	1.88	2.92
Y14	51.3	54.9	1.01	146.0	54.6	21.59	21.53	0.0175	-14.2	54.6	3.0	0	4.09	2.20	1.86	3.30
Y15	50.7	55.2	0.80	145.2	54.7	21.67	21.60	0.0179	-14.0	54.8	2.5	0.1	4.13	2.20	1.88	4.06
Y16	48.8	54.9	0.60	144.5	54.0	21.29	21.22	0.0183	-14.2	54.0	2.9	0	4.20	2.21	1.90	5.22

TABLE 5.7: CONDENSER THERMAL PERFORMANCE RESULTS. IMI YIA CK8-20

CHAPTER 6 PROTOTYPE DESIGN AND FIELD TRIAL

1. INTRODUCTION AND OBJECTIVES OF FIELD TRIAL

A prototype heat pump design was developed based on the previously described component selection studies. This was manufactured and installed in a domestic location for a field trial. A control strategy for the heat pump was devised and implemented on the prototype unit.

The objectives of the field trial were:

1. To gain operating experience, in terms of the performance and reliability of the heat pump under realistic conditions. To this end the installation was extensively monitored.

2. To assess the suitability of the control strategy and to determine the appropriate values of the control settings.

3. To determine the environmental acceptability of the design, for example, with reference to noise levels and appearance.

4. To appraise the viability (cost/effectiveness) of the design for commercial development.

This chapter describes the design and manufacture of the prototype heat pump, the control strategy and its implementation, the field trial location and installation, and outlines the monitoring and data processing schemes. The results of the field trial are presented in Chapter 7.

2. MECHANICAL AND ELECTRICAL DESIGN OF THE PROTOTYPE HEAT PUMP

This section draws together the outcome of the earlier work on component selection by summarizing the details of the components chosen for the prototype heat pump. Table 1 contains the design specification for the complete heat pump, giving the physical size of the unit, expected performance, and additional application details.

2.1 Refrigerant Circuit

2.1.1 Circuit Configuration

The configuration of the refrigerant circuit is shown in figure 6.1. In the following description the circuit components are identified with reference to the numerical key shown in this figure.

Refrigerant evaporation is carried out by an air heated, dual circuit, dry expansion evaporator, 2, and the two outlet streams are combined by the suction header, 3. Liquid separation is achieved by the suction accumulator, 4, which prevents liquid floodback to the compressor during defrosting. The compressor, 6, is a hermetically sealed, reciprocating type protected by low, 5, and high, 8, pressure trips. In order to reduce the compressor motor starting torque requirement a bypass solenoid valve, 7, opens to equalize the pressure across the compressor. Liquid floodback from the condenser or the defrost line when the comperssor is switched off is prevented by a non-return valve, 9. The condenser, 10, is a water cooled, single pass, coaxial tube heat exchanger operated in counter flow. Surplus refrigerant liquid is stored in the receiver, 11, without which the liquid would tend to back up in the condenser and so reduce the area available for heat transfer. The liquid is filtered and dehydrated by a adsorbent type filter/dryer, 12, and a combined sight glass and moisture indicator, 13, is included in the liquid line for maintenance purposes. Refrigerant expansion and metering is controlled by a thermostatic expansion valve, 15. Each evaporator circuit is fed with equal mass flow rate of refrigerant by the action of the distributor, 16. To defrost the evaporator the solenoid valve, 14, is opened which routes the hot discharge gas from the compressor to the evaporator distributor.

As part of the performance monitoring scheme four platinum resistance thermometers were immersed in the refrigerant flow at the positions shown in the diagram, and these are discussed further in section 5.

2.1.2 Component Selection and Application

a) Compressor

The compressor selected for the prototype was the same model as that which was tested on the Component Test Rig and used on the Evaporator Test Rig, i.e. the Maneurop MT32 JF5. The specification of this model is summarized in table 6.2 (75). As the motor windings are internally cooled by the incoming suction vapour, the outer shell of the compressor was thermally insulated. The compressor was fitted with service access valves which were sealed on to the threaded suction and discharge ports. The use of screwed fittings simplifies the removal and replacement of the compressor in the event of failure, and the service valves themselves facilitate the commissioning of the refrigerant circuit. The valves also provide the pressure tappings for the pressure protection switches.

The design of the compressor electrical circuit is dominated by the requirement to contain the motor starting current to an acceptable level. Direct on-line starting of the compressor requires a current of about 80 A for two mains cycles, which may cause a noticeable voltage depression in the household mains supplies of the owner and his neighbours. The electricity supply authorities recommend that any consequent voltage drop must not exceed 1 per cent of the guaranteed supply (88). The maximum allowable current to comply with this requirement will depend upon the impedance and reactance of the local electricity supply, and so will vary from one installation to another. However, it has been indicated that starting currents of 30A and higher are likely to cause nuisance in a significant number of installations (88).

A number of proprietary devices are available to reduce the starting current of a single phase induction motor, an example being the 'static soft starter', model SPSS-004-250-05, manufactured by Hellermann Electronic Components (89). This device contains a ramp generator which applies a low but gradually increasing voltage to the motor. This reduces both the acceleration of the motor and its starting current. The results of a series of trials with this device, conducted by other members of the Group, are presented in figure 6.2. The compressor is supplied with the choice of two basic circuit types; a permanent single capacitor (PSC) version and a capacitor start and run (CSR) version in which a second capacitor is connected in series with the auxillary windings during starting. As shown in figure 6.2, the CSR version offers the greater reduction in starting current when the soft start device is included, and a maximum current of 20 A was observed for this configuration. However, the trials also demonstrated that the motor starting torque is reduced when the soft start device is used, and so to prevent stalling it is necessary to unload the compressor during the acceleration phase. This is achieved by opening a solenoid valve which equalizes the pressure difference across the compressor, as mentioned in section 2.1.1. The sequencing of the operation of the solenoid valve and the starting of the compressor is controlled by 2 delay timers.

An additional consideration in the design of the compressor electrical circuit is the provision of a crankcase heater. When the heat pump is in stand-by mode, i.e. in between heating or defrosting cycles, the compressor must be maintained at a higher temperature than the evaporator and suction pipework, otherwise refrigerant migration and condensation within the compressor shell may occur, with the result of possible damage to the motor windings and to the compressor itself (75). An externally mounted, pencil type electrical resistance heater is available, but a more satisfactory method is to arrange a trickle current to flow through the motor windings in stand-by mode, releasing heat inside the compressor shell by the ohmic resistance effect. This provision is available as a modification to the basic starting circuit and carries the manufacturer's designation "CSR 2".

The complete compressor circuit design is shown in figure 6.3. This is based on the CSR 2 version with trickle current crankcase heating, contains the Hellermann soft start device and the timer controls to sequence the operation of the unloading valve and the closing of the compressor contactor.

b) Condenser

The condenser is a modified version of the heat exchanger evaluated on the CTR, the Yorco-Ax CK8-20 manufactured by Yorkshire Imperial Alloys. The modification requires the heat exchanger to be coiled 7¾ turns instead of the usual 7½ turns, and the resultant geometry is shown in figure 6.4. This configuration simplifies the design of the interconnecting water piping between the condenser and the supplementary heater.

The heat transfer surface is a low fin Integron tube made from a 90/10 cupro-nickel alloy which offers greater erosion resistance compared with pure copper (90). Refrigerant vapour enters the annular space between the Integron tube and the outer shell at the top connection of the heat exchanger, so that the condensate drains towards the outlet connection. The water flows through the inner tube in counter current fashion. The outer surfaces of the condenser are thermally insulated. The nature and application of the heat transfer surface are illustrated in figure 6.4, and the specification of the condenser is presented in table 6.3. The information contained in this table was mainly provided by the manufacturer, but some entries, for example, the water side pressure drop, were obtained by measurement, see Chapter 5.

c) Evaporator

The design of the evaporator was derived from by the optimisation study of Chapter 4; the 30 m version was applied to the prototype heat pump. The heat transfer surface consists of an externally finned wire wound tube made by Clayton Dewandre Ltd. The refrigerant pathway comprises two parallel dry expansion circuits, coiled into two concentric helical rows, see figure 6.5. The specification for the evaporator is given in table 6.4.

d) Expansion Valve and Distributor

The expansion valve is a cross-charged thermostatic type, model TERL/A3-R22 manufactured by Egelhof (91). Some operating experience of this valve was gained with the Evaporator Test Rig. Owing to the pressure drop introduced by the refrigerant distributor the valve is externally equalized. Table 6.5 contains the component specification for the valve.

The evaporator refrigerant feed is evenly split into the 2 circuits by the action of the distributor. This is a venturi throat type made in brass, model VK00/12.5.2, also manufactured by Egelhof (91). The distributor outlets are connected to the evaporator circuits by small bore distributor tubes, see table 6.6.

e) Pipeline Components and Circuit Accessories

The selection and pertinent specifications of the pipeline components and accessories are documented in table 6.6. The function of each item has already been described in section 2.1.1.

The tube diameter and gauge of the major line sections is presented in table 6.7. Refrigeration grade soft copper was used throughout and the adoption of Imperial sizes reflects their widespread availability compared with metric tube. The distributor pipework was changed during the field trial, and the pipe sizes used are shown in the table.

Where possible tube and fittings were soldered together using refrigeration grade capillary fittings, as this method is less prone to leakage than flared compression joints. All pipe runs containing refrigerant at elevated temperatures were thermally insulated by a foam based product to a typical thickness of 12 mm.

2.2 Ancillary Components

2.2.1 Fan

A second prototype version of the 450 mm diameter Group 1 series fan was supplied by Woods of Colchester Ltd. This fan has a 5 bladed fabricated steel propeller and a plane blade geometry. The 900 rev/min motor was specially wound to increase motor efficiency. A similar prototype fan was evaluated on the ETR and was designated fan B in Chapter 4. The dimensions are shown in figure 6.6.

2.2.2 Supplementary Heater

The supplementary heater is required to augment the heat output of the heat pump when the ambient temperature falls below the balance point and when a rapid warm up of the building is required. It takes the form of an electrically powered in-line flow boiler of 6 kW maximum output. This capacity, which is the same as the nominal output of the heat pump was chosen to ensure that the building heat demand at the usual design ambient temperature of -1°C can be satisfied by the combined operation of the heat pump and flow boiler.

The flow boiler was manufactured by Redring Electric to their drawing reference number 539-1059. The outer shell is fabricated from a mild steel tube with steel end plates which are penetrated by 22 mm outside diameter copper pipes for water inlet and outlet, see figure 6.7a. The flow boiler contains two helical heating elements rated at 2 kW and 4 kW, allowing the heat output to be varied in 2 kW steps up to 6 kW. The design incorporates a thermostat pocket, and as an additional safeguard, a second thermostat is surface mounted on the inlet end plate and used as a high temperature safety cut-out.

The flow boiler is contained within the heat pump casing and is piped in series with and immediately downstream of the condenser, see figure 6.7b. In order to prevent entrained air collecting inside the flow boiler, it was positioned so that the water flow rises through it. The flow boiler and water piping were thermally insulated.

2.2.3 On-Board Controls

A number of items associated with the electrical mains supply and control circuitry are contained within the heat pump assembly, including the compressor starting circuit discussed in section 2.1.2, the mains contactors for the fan, flow boiler and solenoid valves, and a number of electrical protection devices. The mains supplies circuits are shown in figure 6.8, which traces the supplies back to the consumer unit, and the control circuits are shown in figure 6.9. The on-board electrical equipment is contained in a water tight enclosure designated 'enclosure 2' in the wiring diagrams. The electrical protection includes current overload, over temperature and excessively high or low pressure, and is summarized in table 6.8.

The details of the control strategy and its implementation are discussed in section 3.

2.3 Frame and Casing Design

A three tier mild steel space frame was designed to support and protect the heat pump components, see figure 6.10. The outside dimensions were chosen to enable the heat pump to pass through a personnel door, to simplify installation.

The lowest deck houses the compressor, condenser, flow boiler, electrical controls and most of the refrigerant circuit components. The base plate is supported by an angle-iron base. The compartment is enclosed by mild steel panels which are lined with an accoustically absorbent foam.

The central deck supports the evaporator and the suction accumulator. The air discharge grilles are constructed from a diamond shaped expanded aluminium mesh. A cone shaped air deflector made from accoustic foam supported by a wire frame is positioned over the accumulator. The evaporator support plate has a central depression in order to drain condensate towards the centre of the plate where a drain pipe is attached. This pipe runs vertically through the lowest compartment and discharges the water at ground level. The warmth of the compressor compartment prevents the water from freezing inside the drain pipe.
The upper deck supports the propeller fan and its bellmouth housing. The air intake grilles are made from the same mesh as the discharge grilles. The intake air duct is shaped by a pyramidal air deflector which is lined with accoustically absorbent foam. The top surface of the heat pump is enclosed by a folded mild steel panel.

The frame and all panels were spray coated with a chlorinated rubber paint to protect the steel from corrosion and to produce an attractive finish. The ERC Design Group reference number for the frame and casing design is W21056 A3 514.

2.4 Assembly

The layout of the major components of the heat pump is shown in pictorially in figure 6.11. The assembly scheme layout drawing carries the ERC Design Group reference number 20129 A0 472. However, this drawing does not define the design of refrigeration pipework or show the routing of the electrical wiring, as it was felt that these details would be better finalised during assembly. The complete parts list for the prototype carries the reference number W20158 A3 095.

The prototype was assembled from the base up in the following sequence:

1. Assembly of the base and frame corner posts.

2. Fitting, piping and thermal insulation of the refrigeration components and the flow boiler. Fixing of the electrical control enclosure and wiring up of compressor, flow boiler and solenoid valves.

3. Installation of the condensate drain and the evaporator support tray.

4. Fitting and piping up of the evaporator and suction accumulator.

5. Sealing and dehydration of the refrigerant circuit.

6. Assembly of the fan support plate and the installation and wiring of the fan.

7. Charging of the refrigeration circuit.

8. Completion of the casing with the fitting of the top and side panels and the intake and discharge grilles.

Figure 6.12 shows several views of the prototype unit during assembly.

3. HEAT PUMP CONTROL SYSTEM

3.1 Outline of Control Functions and Methods

The control functions are summarized as follows:

a) The control of the heat output of the heat pump by on/off cycling in response to the building heat demand.

b) The initiation, mode selection and termination of the evaporator defrosting operation.

c) The appropriate use of the supplementary heater.

d) The provision of pipework ice protection.

The functions and the methods are discussed in the following subsections. The control system is designed so that it may be implemented by an electrical control circuit containing six thermostats, seven delay timers and a clock. However, for the purposes of the prototype field trial, all the control functions described in this section were carried out using a microcomputer. This was done to provide flexibility in the design of control strategy and to enable the control settings (thermostat set points, timer settings etc.) to be easily adjusted. The measurements made by the performance monitoring system supply the temperature inputs to the control system emulator, and all timing functions are carried out by the computer's internal clock. The details of this scheme are discussed later in section 3.6.

The computer based control system was designed to simulate the behaviour of various electrical devices which could be assembled into a practical controller for use on the production version of the heat pump. A conceptual circuit diagram of a practical controller is shown in figure 6.13. In the following sections the details of the control strategy adopted for the prototype heat pump are described in terms of the electrical devices shown in the figure.

3.2 Regulation of Heat Output

The heat output of the heat pump is matched to the building requirement by on/off cycling in response to a single room thermostat. This is also a common method of controlling small scale gas and oil fired boilers. The control system simulates the action of a conventional close differential room thermostat, using the reference room air temperature measurement, T_{ar}, from a resistance thermometer located in the hallway of the house. In addition to the room thermostat the heat pump may be cycled off due to high condenser water outlet temperature. This is to protect the compressor against excessive discharge pressure. This also is common practice for domestic water heating equipment. The signal from thermometer T_{w2} is used as the basis for the simulation of a water outlet thermostat control, with a set point of 55°C and a differential of 6 deg C.

At the end of a heating cycle a procedure is initiated which involves a pump overrun, a compressor lockout timer and a defrost operation. The delay timers Dt1 and Dt2 are started when the room or condenser outlet thermostats open to end the heating cycle. Dt1, typically set to 5 minutes, controls the duration of the end of cycle defrost operation. When the field trial began this was simply a fan-only type defrost, in which the compressor is stopped but the fan continues to rotate. During the trial this was modified so that the defrost mode was determined by the status of the defrost mode selection thermostat, as explained in section 3.3. Hence at low ambient temperatures, an end of cycle hot gas defrost would be executed for Dt1 seconds. The pump overrun timer, Dt2, typically 10 minutes, maintains the water circulation through the condenser in order to recover the residual heat from the refrigerant and the heat exchanger. For the first part of the trial this feature was not used and the pump was continuously operated, in order to permit continuous monitoring of the heat sink temperatures. The timer Dt2 has the second function of preventing the heat pump from restarting. This is to protect the compressor from rapid on/off cycles which may reduce its service life. When Dt2 has counted out the heat pump is able to restart, and will do so if both the room and water thermostats are closed.

A further common feature of a domestic central heating control scheme, the time/programmer, is also simulated by the computer system. It allows two heating periods per 24 hours to be defined by the user, and contains override functions for continuous heating and permanently off.

3.3 Evaporator Defrost Initiation, Mode Selection and Termination

A common method of initiating the defrost cycle is to defrost at regular intervals based on the compressor run time. While this approach is simple and robust it leads to unnecessary defrosting at mild ambient conditions since the interval timer must be set for the worst conditions (92). An improvement is to couple the timer to a temperature sensor to inhibit defrosting in mild conditions, or as proposed by Mueller and Bonne to vary the defrost interval time in line with the ambient temperature (93). However, as demonstrated in Chapter 4, the rate of frost accumulation is influenced by additional factors, such as the ambient relative humidity and the heat pump condensing conditions, hence a method based on the detection of ice blockage of the evaporator is likely to be more satisfactory.

A number of parameters may be used as the basis of an ice detection or demand type system, including resistance of air flow through the evaporator, evaporating pressure, optical frost sensing, fan power sensing, electrical capacitance and temperature difference (94, 95). The technique chosen for the prototype heat pump is to initiate a defrost cycle when the air to refrigerant temperature difference across the evaporator, $T_{a1} - T_4$, exceeds a preset threshold. The evaporator frosting trials reported in Chapter 4 demonstrated that this temperature difference is a clear signal of the extent of ice blockage of the coil. This method has the further merit of simplicity and ease of implementation.

The value of the temperature difference at which to initiate a defrost cycle requires careful consideration. From figure 4.19 it is clear that a set point of 8 deg C would trigger a defrost cycle before the COP_h had fallen by more than 5 per cent of the initial value. However, it is possible to develop an evaporator temperature difference greater than this value with an unblocked coil under certain conditions, i.e. at high air temperatures and low condenser water temperatures. Thus the trigger level must exceed approximately 10 deg C if unnecessary defrost cycles are to be avoided. At this value the COP_h is reduced by about 10 per cent.

The temperature difference initiation scheme is implemented on the prototype heat pump by means of direct measurement of the air and refrigerant saturation temperatures, using the resistance thermometers installed for performance monitoring, labelled T_{a1} and T_4 respectively. T_4 is immersed in the refrigerant flow at the inlet of the evaporator outer circuit, which is the same position as thermometer T_{5A} of the ETR. T_{a1} is in the air intake duct. The differential temperature $T_{a1} - T_4$ is computed digitally and the result subsequently manipulated by the thermostat simulation routine of the control program. The output from this routine is the current status of the thermostat. This may be either open, meaning a defrost cycle is not required, or closed, which occurs when the measured temperature difference exceeds the set point, and causes a defrost cycle to be initiated. In a production version this scheme could be implemented using an actual differential thermostat, with the air sensing bulb held within the air intake region and the refrigerant temperature sensing bulb clamped to the evaporator inlet tubing. Suitable thermostats are commercially available at low cost and the electrical diagram of figure 6.13 contains a defrost thermostat (96). An alternative arrangement is to use thermistors or resistance thermometers in conjunction with an electronic comparator amplifier, producing a switching output related to the temperature difference. Buick et al suggest a suitable circuit and report satisfactory results with a detection system based on platinum resistance thermometers (95).

A further consideration in the implementation of this method is to avoid spurious triggering of the defrost cycle due to abnormally low evaporating temperatures during start up. This is caused by the slow response of the thermostatic expansion valve which tends to remain closed when the compressor first operates. The effect is a sudden fall in the evaporator pressure, and a corresponding reduction in the saturated temperature, as shown in figure 6.14. The prototype heat pump control system contains a simulated delay timer, Dt3, which inhibits the operation of the temperature difference measurement scheme for 10 minutes after the compressor starts. This feature could also be incorporated into an electronic comparator type circuit, but may be difficult to accomplish in the differential thermostat method.

The prototype heat pump has two methods of defrosting the evaporator; hot gas bypass and fan-only defrost, the selection dependent on the ambient air temperatures. At air temperatures substantially above freezing the fan-only defrost is appropriate, in which the compressor is stopped and the fan continues to rotate, thus exposing the evaporator surfaces to a relatively warm air stream. As the fan motor consumes only 150 W, this is a low energy defrosting method. However, this technique is unsatisfactory at low air temperatures; at ambients below about 5°C the hot gas bypass defrost mode is selected. The fan is stopped, the defrost solenoid valve is opened and the compressor continues to operate with the effect of injecting the hot discharge vapour directly into the evaporator distributor. The compressor motor power is typically 2 kW in hot gas bypass mode. Selection between these two methods is accomplished by using a simulated thermostat positioned in the air intake, of set point approximately 5°C. The status of this thermostat at the start of the defrost cycle determines the defrost method. Since the heat emitted by the evaporator during hot gas defrost tends to increase the air temperature in the region of the intake and so may cause the status of the thermostat to be reversed, it is necessary to latch the defrost mode for the duration of the defrost cycle. During the field trial it became apparant that the fan-only defrost mode was in fact a redundant feature, since the frost accumulation rate was low at the air temperatures appropriate to this mode. Thus to simplify the control design, the mode selection function was discontinued and the hot gas method was implemented when a defrost cycle was initiated.

The defrost termination procedure differs for each mode. The hot gas bypass cycle is of fixed duration, as governed by delay timer Dt4, set to about 30 minutes. For reasons of simplicity this time based termination method was preferred to a coil temperature sensing approach. The fan-only mode may be terminated by either the resetting of the simulated defrost thermostat, or by the action of a watchdog timer, Dt5. The switching differential of the defrost thermostat ensures that the saturated refrigerant temperature T_4 exceeds 0°C when the thermostat resets. The purpose of the timer is to prevent excessively long fan-only defrost cycles.

3.4 Control of the Supplementary Heater

There are three aspects to the control of the supplementary heater; enabling, sequencing and termination. The end user is provided with override options over the operation of the supplementary heater; it may be disabled entirely, or enabled only during the off-peak tariff period, or may be enabled to operate at any time. Instruction supplied to the end user would emphasise the financial penalties of operation of the supplementary heater, and recommend that for most of the heating season the flow boiler is disabled. The facility of enabling operation for a particular time of day, i.e. during the off-peak period, requires that the control system contains a clock with the appropriate period defined. The field trial control system uses the computer's internal clock to determine the tariff period.

Assuming the user has authorized the use of the supplementary heater, the remaining enabling procedure is as follows. The central theme is to allow time for the heat pump alone to meet the building heat load in order to diminish the contribution from the supplementary source. Thus the heater is disabled for a period of about one hour, as set by the delay timer Dt6, from the time when the room thermostat first closes, i.e. calls for heat, provided that the heat pump can operate. At ambient temperatures above the balance point this period is usually sufficient to allow the heat pump satisfy the heat load, in which case the supplementary heater will not be used at all. Below the balance point the heat pump output is less than the heat requirement and so the thermostat may still be open at the end of timed period, in which case the second phase of sequencing will begin.

The heater power is sequenced in the following manner. Each step up in heat output must be preceded by a period of operation at the lower heat output level. This is controlled by delay timer Dt7, typically set to 60 minutes. Thus, once Dt6 is timed out, the flow boiler is not energized until a further 60 minutes elapse, and then the 2 kW stage is activated. The power would be increased to 4 kW if the room thermostat remains closed for a further 60 minutes and the process is repeated until all three stages are in use, unless flow boiler operation is terminated.

The supplementary heater may be shut down by a variety of means:

a) The room thermostat opens, indicating the heat demand is satisfied, in which case the flow boiler is de-energized and the enabling procedure restarts.

b) The simulated flow boiler discharge water thermostat opens due to excessive water flow temperature, which causes the power input to be immediately stepped down one 2 kW stage.

c) The end user revises his override options, preventing operation during the present tariff period; the flow boiler is immediately de-energized and disabled.

d) The central heating timer control opens at the end of the heating period; again the flow boiler is immediately de-energized.

The supplementary heater power is reduced by one 2 kW stage if the heat pump is cycled off due to high condenser water temperature. This function ensures priority is given to the operation of the heat pump rather than the supplementary heater, and was discussed in Chapter 3. This function was introduced midway through the field trial as a modification to the control strategy.

3.5 Pipework Ice Protection

To obviate the risk of freezing the water in the condenser and in the outside pipework, an ice protection thermostat is simulated, based on the measured water return temperature at point of entry to the heat pump, T_{w1}. This thermostat closes at 5°C, causing the circulation pump and the heat pump to be cycled on. The thermostat opens when the water temperature reaches 20°C, whereupon the heat pump and circulator are returned to their previous states. In practice this thermostat would take the form of a bimetallic strip type clamped onto the water return pipework near to the heat pump. The ice protection system overrides all other control functions and remains active when the heat pump is switched off by the action of the central heating timer switch.

3.6 Implementation of the Control Scheme for the Prototype Field Trial

For the purposes of the prototype field trial the heat pump control functions described above were carried out using a micro-computer system. This computer system was also the basis of the performance monitoring equipment, and is discussed in section 5. Here the sub-routines associated with the control function are described along with the details of the actuation system.

3.6.1 Program DAC - Control and Actuation Sub-routines

The Data Acquisition and Control program, DAC, was specially written to carry out the joint tasks of heat pump performance monitoring and the control system simulation. Three sub-routines are concerned with the control functions.

a) Sub-routine Hardware Control Simulator

Temperature measurements made by the monitoring system are converted to 6 thermostat positions (i.e. open or closed) by reference to the thermostat set points and differential settings and to the immediate past history of the temperatures. The thermostats are represented as flags with only two possible values, 0 for open and 1 for closed. In addition this sub-routine simulates the central heating timer switch, by reference to the computer's internal clock and pre-defined heating periods. Again a two value flag is used to indicate the timer's status. The signals generated by this sub-routine are passed to sub-routine Controller.

b) Sub-routine Controller

The control decisions, as described above, are taken in this sub-routine. The inputs are the thermostat and main timer flags, the outputs are six hardware actuation flags which govern the status of the principal items of equipment in the heat pump; compressor, fan, water circulation pump, defrost valve, and the 2 kW and 4 kW flow boiler elements. The seven delay timers are initiated, read and reset by statements contained within this sub-routine, and are simulated using the computer's internal clock. The hardware actuation flags are passed to sub-routine Actuator.

c) Sub-routine Actuator

This sub-routine instructs the HP3421A data acquisition and control unit to open or close any combination of 6 reed relay contacts, depending on the status of the hardware actuation flags.

In addition, program DAC contains a series of functions to enable the operator to change any of the control settings; thermostat set points and differentials, timer settings, central heating timer settings and overrides and the supplementary heater options. All revised control settings are stored on disk together with the time and date of the latest change, for later analysis.

3.6.2 Hardware Actuation System

In order to protect the HP3421A, its relays are used to activate 12 V DC circuits rather than the 240 V AC mains circuits, as shown in figure 6.15. The low voltage circuits contain the coils of a set of intermediate relays which activate the mains control circuits. They also contain manual override toggle switches which enable the operator to control the heat pump independently of the computer based controller. A pair of lamps indicate, for each hardware function, the status of the HP3421A actuator and of the intermediate relay. The role of the intermediate relays in the 240 V AC control circuits is shown in figure 6.9.

4. INSTALLATION DETAILS

4.1 Location

The author's own house hosted the field trial. This is a 3 bedroom, 2 storey, semi-detached house of 80 m^2 total floor area, built around 1936 and located in the Glen Parva district of Leicester. The floor plans are shown in figure 6.16. The external walls of the house are 9 inch brick with internal plaster, and all internal walls are brick. The floors are wooden floor boards on joists. All the windows are single glazed and most have timber frames. The pitched roof is constructed of tile on battens with roofing felt and the loft space is insulated with a 100 mm layer of fibre glass. As a result of its construction it is expected that this house would have a relatively high heat loss and high thermal mass, compared to present construction standards.

4.2 Estimated Heating Requirement

The heating requirement of the building was calculated for the following case:

a) Outside air temperature -1°C.

b) Negligible solar gains, i.e. night-time conditions are assumed.

c) Comparatively high ventilation rates are assumed, particularly for the two main ground floor rooms as they have open chimneys.

Table 6.9 records the design room temperatures, ventilation rates, casual gains and the heat transmission coefficients used in the calculation, along with the results (97, 98). Notice that of the total building heat loss rate of 8.2 kW, some 2.7 kW is attributable to ventilation, and that of all the parameters contained in the calculation it is the air infiltration rate which is the most difficult to estimate. When incidental heat gains are considered the building heating requirement is estimated to be about 7.3 kW at -1°C.

As an independent check on the building heat loss estimates, the details of the dwelling were submitted to the computer based radiator sizing service operated by the radiator manufacturer (99). This assessed the total building heat loss as 8.6 kW, which is close enough to give confidence.

4.3 Original Central Heating System

The original gas fired central heating system comprised a Baxi back boiler of about 12 kW output, a gas fire and five single panel Stelrad radiators connected as a fully pumped, two-pipe network. The sizes and heat dissipation capacity of the radiators are shown in table 6.10. Notice that the original radiator capacity is in general less than the estimated heat requirement, i.e. that the radiators were undersized.

The boiler also feeds an indirect, self priming, domestic hot water cylinder via a gravity circulation system.

4.4 Radiator Sizing for a Heat Pump Installation

It was demonstrated in Chapter 3 that the minimum requirement of a radiator system when operating with the heat pump was to dissipate the building design heating load at a water flow temperature of 55°C. Clearly the original radiator installation did not satisfy this requirement. For the purpose of the trial the original radiators were replaced by larger, double panel convector type radiators with an output of approximately 1.5 times the heat demand of individual rooms, as detailed in table 6.10. This provided a total output at 55°C of 10.7 kW, or 1.47 times the estimated building heating load at an ambient of -1°C.

4.5 Heat Sink Design

In most cases the replacement radiators were selected to be of the same length as the originals in order to reduce the alterations to the pipework. However, as the double panel radiators have the connecting ports midway between the panels, some pipework reconstruction was needed in every case. The two rooms which were not heated by the original radiator network were also fitted with double panel convector type radiators.

Each radiator was fitted with a 'lockshield' regulating valve and, with the exception of the radiator in the hallway, with a thermostatic radiator valve. The purpose of fitting the thermostatic valves was to ensure satisfactory regulation of the heating system once the trial was completed and the heating source reverted to the higher output temperature gas fired boiler. A thermostatic valve was not fitted to the hallway radiator because that was the location of the house heating system control thermostat.

The arrangement of the revised heat sink is shown in figure 6.17. Priming and thermal expansion of the primary circuit took place inside the DHW cylinder. Pump P2, Grundfos UPS 18-60, was installed to circulate water around the heat pump and radiator network. In order to produce the highest flow rate the pump was set to operate at its maximum speed of 2100 rev/min. In case of complete failure of the heat pump, the heat sink was designed to enable the gas boiler to provide space heating via the original circulator P1, by altering the valves 1, 2 and 3. However, this design did not allow space heating from the combined efforts of the heat pump and the gas boiler.

The pipe run from the house to the heat pump was a total length of approximately 16 m, and was mainly 28 mm outside diameter copper tube. The pipe diameter was reduced to 15 mm near the point of entry to the building, for the purposes of installing the turbine flow meter and to reduce the disruption to the house wall. The pipe size was also reduced to 22 mm at the point of connection to the heat pump. The pipe run contains two electromagnetic heat meters, two resistance thermometers to measure the flow and return temperatures close to the heat pump, a pressure relief valve, isolation and drain valves, and a filter to protect the flow meter. The outside pipe run was thermally insulated to a high standard, so that the calculated heat loss at 50°C mean water temperature and 0°C air temperature was about 100 W, i.e. less than 2 per cent of the nominal heat pump output.

4.6 Electrical Supplies

The electricity supply to the house is a single phase 240 V, 100 A on the Domestic Economy Seven tariff. A control and monitoring station was set up in the garage. A 60 A supply was installed in the garage and two consumer units were fitted to provide separate supplies for the heat pump and the control/instrumentation systems. Each consumer unit was fitted with a residual current circuit breaker of 30 mA sensitivity.

4.7 Delivery and Installation of the Heat Pump

The heat pump was transported to site in a 22 cwt van. Lifting from the loading deck required the services of 4 men, but movement along the ground was facilitated by using a trolley to support the weight. The value of restricting the width of the product was demonstrated when it had to be manoeuvred through a narrow doorway.

The short distance between the Laboratory and the field trial location, and the relative ease of handling the heat pump, meant that the time taken to load, deliver, unload and position the heat pump was about 2 hours. The manpower requirements could be reduced by using a delivery van equipped with a tail lift.

The heat pump was positioned on an existing concrete slab about 8 m from the south face of the house. A photograph of the heat pump after delivery to site is presented as figure 6.18.

5. PERFORMANCE MONITORING

5.1 Monitoring Scheme

The monitoring scheme was based on a computer controlled data logging system for instantaneous measurements of various air, water and refrigerant temperatures, air relative humidity, water flow rate and electrical power. All of the thermometers were four wire, platinum resistance types to B1904 : 1984 class A. The three thermometers immersed in the water circuit were matched and calibrated to reduce uncertainty in the temperature difference measurements, $T_{w2} - T_{w1}$ and $T_{w3} - T_{w2}$, so that the heat output rates from the heat pump and supplementary heater could be accurately calculated. Four air temperatures were measured; the ambient air temperature inside a Stevenson pattern screen, the reference room air temperature (in the hallway of the house), and the evaporator air-on and air-off temperatures. Four refrigerant temperatures were measured; evaporator outlet, compressor discharge, receiver outlet and evaporator inlet. The monitoring scheme is illustrated in figure 6.19, and table 6.11 lists the complete set of the measured variables.

The computer program to control the logging instrumentation, program DAC, contains a number of features. The main tasks were to control the data acquisition unit, convert the measured raw data to engineering units, store the data on a flexible disk for later analysis, and display the results on the monitor. The program also contains the heat pump control sub-routines described in section 3.6.1.

In order to provide an independent check on the results from the computer based data logging system, a set of self-contained integrating instruments were installed and the indicated totals were manually recorded at regular intervals. These instruments were two domestic type kWh meters, one in the supply to the heat pump and the other in the supply to the supplementary heater, and two electromagnetic heat meters, both of which recorded total heat delivered to the building. Two heat meters were used to provide some redundancy in case one were to fail.

Appendix A contains a full description of the field trial monitoring scheme, including the specifications of the transducers and instrumentation, analysis of errors, and details Program DAC.

5.2 Data Processing

5.2.1 Overview of the Data Processing Scheme

An overview of the data processing scheme is presented in figure 6.20. The raw data consisting of the instantaneous transducer voltage measurements are reduced in two stages. Firstly, they are converted to physical quantities (temperature, flow rate etc.) and then averaged and totalled over a period of four hours. This process is carried out by program FHA, 'Four Hourly Analysis' and leads to the production of a compact data file FHD 'Four Hourly Data'. Secondly, the 4 hourly results are further reduced to 24 hour means and totals by program DAYTOTAL, which generates the DAYTOTALS data file. As shown in the diagram, printed output is raised from all three data files; instantaneous, 4 hourly and daily. A suite of graph plotting programs (not shown) has also been developed for diagrammatic presentation of the contents of the three data files.

The usual pathway through the data processing scheme is as follows. The full data disk arriving from the field trial is supplied to program RDSS REPORT which retrieves the voltage data, converts it to engineering units and prints each record on the line printer. Printed output of the control settings is also produced. This printout is useful for troubleshooting and is filed in chronological sequence. The data are reduced by program FHA to a series of averages and totals for all the 4 hour periods covered by the data. The results are stored on a separate disk in file FHD, the original raw data disk is then archived. Two printed reports are made from the 4 hourly data file, a summary of the contents so arranged to contain one days worth of results per page, and a table of the 4 hourly mean operating conditions specifically for the hot gas defrost mode. Graphical output may also be generated from the 4 hourly data file. The next stage compacts the data further by deriving the mean daily results from the 4 hourly period data, and the resultant daily data file also generates a printed report and is accessed by the graph plotting programs. The final processing stage is the reduction of all the records on the 24 hour file into one; this produces a file in which the data is sorted by daily mean ambient air temperature, and produces a concise summary of the results of each part of the trial.

Appendix B describes the data reduction processes in detail, specifies the contents and structure of the data files which are generated and provides output samples of each of the major report generating programs.

5.2.2 Period Numbering Conventions

The prototype heat pump first produced useful heat on 7 February 1985, and so this date is the starting point for the period numbering conventions:

a) Four hourly period number 1 is 00:00 to 04:00 GMT on 7 February 1985.

b) Day number 1 is 7 February 1985.

c) Week number 1 represents the period Sunday 10 February, 00:00 GMT to Saturday 16 February 1985, 24:00 GMT.

PROTOTYPE HEAT PUMP AT FIELD TRIAL LOCATION
IMMEDIATELY AFTER DELIVERY

FIG. 6·18

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Air flow rate through evaporator	$1.3m^3/s$	
Water flow rate through condenser	0.5 to $1.5m^3/h$	
Water side pressure drop at 1 m^3/h	8.8 kN/m^2	
Water side total capacity	4 litres	
Water pipework size	22 mm Outside Diameter	
Refrigerant	R22 ($CHClF_2$)	
Charge	5 kg	
Refrigerant flow rate	0.02 to 0.05 kg/s	
Refrigerant circuit pressure protection		
minimum suction pressure	1 bar gauge	
maximum discharge pressure	24 bar gauge	
Performance		
Heat output capacity (coefficent of performance in brackets)	water discharge temperature	
air temperature	50°C	55°C
-1°C	5.87 kW (2.34)	5.62 kW (2.12)
+7°C	7.57 kW (2.73)	7.31 kW (2.51)
Maximum water delivery temperature with heat pump operating	55°C	
Electrical supply	240 V AC, 50 Hz, 1 ph, 40A	
Compressor	2.5 kW nominal	
Fan	0.15 kW	
Flow boiler	2, 4 and 2+4 kW	
Size	1352 mm high by 705x705 mm plan	
Weight	150 kg (estimated)	

TABLE 6.1: DESIGN SPECIFICATION AND RATING OF PROTOTYPE HEAT PUMP

Manufacturer	Maneurop S.A. (France)
Model Reference	MT32 JF5 VEH
Type	Hermetically sealed, single cylinder, single acting, reciprocating.
Displacement	$53.8cm^3$/revolution, or $9.3m^3$/h at the nominal shaft speed of 2900 rev/min.
Motor	2 pole, split phase induction, 220-240V/1/50Hz
Insulation Class	F, suction cooled.
Power input	2.5kW nominal
Internal Overload	Temperature and current sensing bimetallic cut out incorporated in the windings. Cut out temperature = 110°C, cut in = 69°C.
Max Suction Pressure	8 bar, gauge
Max Discharge Pressure	27 bar, gauge
Shell Test Pressure	26 bar, gauge
Lubricant	Maneurop White Oil (based on Shell Clavus 32)
Lubricant Charge	0.92 litres
Shell Dimensions	343 mm high x 225 mm diameter
Net Weight	25 kg
	Source: Maneurop S.A.

TABLE 6.2: COMPONENT SPECIFICATION - COMPRESSOR

Manufacturer	IMI Yorkshire Imperial Alloys (UK)
Model Reference	Yorco-ax CK8-20-Special
Type	Water cooled coaxial tube condenser
Materials	
Inside tubing	Integron 197049 low fin tube made from 90/10 cupro-nickel alloy, 'Kunifer 10'
Outside shell	Plain bore copper tube
Length of inside tubing	9.8m
Heat exchange areas	
Water side	0.604 m^2
Refrigerant side	2.017 m^2
Internal volumes	
Water side	3 litres
Refrigerant side	5.5 litres
Maximum shell pressure	26 bar, gauge
Water side pressure drop	5.7 kN/m^2 at $1m^3/h$
Maximum water side velocity	2.5m/s
Dimensional details	see figure 6.4
Dry weight	30 kg

Source: Derived from data supplied by IMI Yorkshire Imperial Alloys

TABLE 6.3: COMPONENT SPECIFICATION - CONDENSER

Manufacturer	Clayton Dewandre (UK)
Model Reference	CTGA 1134
Type	Dry expansion, air heated, wire wound tubing, 2 helically arranged refrigerant circuits
Materials	swg 22 copper wire fin on swg 21 copper tube
Tube size	0.5 inch outside diameter
Fin loop winding formula	Manufacturer's reference 22/½/20/1.515/30.8/52.6/0.05 No 150
Tube lengths	
inside row	12.61m
outside row	17.70m
Heat exchange areas	
air side	$16.24m^2$
refrigerant side	$1.04m^2$
Internal volume	2.2 litres
Air side face area	$0.78m^2$
Dimensional details	see figure 6.5
Dry weight	30kg

Source of tube surface details:
Clayton Dewandre

TABLE 6.4: COMPONENT SPECIFICATION - EVAPORATOR

Manufacturer	Egelhof GMBH and Co. (West Germany)
Model Reference	TERL/A3-R22
Type	Externally equalized thermostatic expansion valve
Element Charge	Liquid
Factory set static superheat	4 deg C, adjustable
Materials	Diaphragm valve with stainless steel head, brass valve body.
Test pressure	28 bar, gauge

TABLE 6.5: COMPONENT SPECIFICATION - EXPANSION VALVE

COMPONENT	MANUFACTURER	MODEL REFERENCE	SPECIFICATION
Suction Accumulator	PARKER (USA)	PA410-5-5	U tube type with metering orifice for oil return Internal volume 2 litres Design max pressure 24.5bar gauge
Liquid Receiver	STANDARD (USA)	L-514	Upright type Internal volume 4 litres Design max pressure 27.5 bar gauge
Filter/Drier	SPORLAN (USA)	C-164	Porous core type containing $16in^3$ of various desiccants
Sight Glass	SPORLAN	SA14FM	Full view sight glass with colour-calibrated moisture indicator
Non Return Valve	BAILY GILL PRODUCTS (UK)	GBI08F	Valve body and piston in brass Synthetic rubber seal
Shut off Valve	VIRGINIA CHEMICALS INC (USA)	RB4S	Brass ball valve maximum pressure 34.5 bar gauge
Solenoid Valve	SPORLAN	B10F2	Normally closed, electrically operated. Coil type: MKC1,240V, 50Hz
Pressure Switch			
a) low	FANAL (West Germany)	FF31-RLP/C-FL	Trip pressure 1 bar gauge Manually reset
b) high	FANAL	FF31-RHP/C-FL	Trip pressure 24 bar gauge Manually reset

TABLE 6.6: SPECIFICATIONS OF REFRIGERANT PIPELINE COMPONENTS AND CIRCUIT ACCESSORIES

Line section	Tube Outside Diameter (inch)	Tube Gauge (swg)
Suction	7/8	19
Discharge	1/2	21
Liquid	1/2	21
Defrost	1/2	21
Distributor tubing		
a) Initial design	3/16 x 1.37m long	22
b) Modified design	1/4 x 1m long	22

TABLE 6.7: REFRIGERANT CIRCUIT PIPELINE SIZES

Item	MCB trip current (Amps)	External Overload current (Amps)	Over-temperature Limit (°C)	Other Protection
Compressor	15	12.5	Windings IOL 110°C cut out 69°C cut in	Low pressure limit 1 bar High pressure limit 24 bar
Fan	2			
Solenoid Valves	2			
Flow boiler				
2kW element	10		Internal thermostat 50 - 90°C and high temperature cut out at 90°C	
4kW element	20			

TABLE 6.8: SUMMARY OF ELECTRICAL PROTECTION FEATURES

ROOM	DESIGN TEMPERATURE (°C)	VENTILATION RATE (No of air changes per hour)	ESTIMATED HEAT LOSS RATE IN WATTS AT -1°C						TOTAL HEAT LOSS RATE (W)	ESTIMATED INTERNAL GAINS (W)	ROOM HEATING REQUIREMENTS (W)
			FABRIC LOSS					VENTILATION LOSS			
			External Wall U=2.1	Party Walls U=2.2	Roof U=0.35	Floor U=1.4	Windows and Doors U=3.5				
Kitchen	21	2	270	-	-	80	250	170	770	770	0
Lounge	21	3.5	240	0	-	160	490	780	1670		1670
Living room	21	3.5	300	0	-	160	160	760	1380		1380
Hallway/ Landing	17	1.5	700	0	40	120	210	290	1360		1360
Bathroom	21	2.0	350	0	30	-	110	120	610		610
Bedroom 1	18	1.5	260	150	100	-	140	270	920	230	690
Bedroom 2	21	1.5	260	180	100	-	160	260	960		960
Bedroom 3	18	1.5	350	-	40	-	90	90	570		570
Total			2730	330	310	520	1610	2740	8240	1000	7240

NOTE: Units for U-values are W/m^2k

TABLE 6.9: ESTIMATED HOUSE HEAT LOSS AND HEATING REQUIREMENTS

ROOM	EXISTING RADIATOR CAPACITY			REPLACEMENT RADIATORS				
	length x height, type (mm) (mm)	Output[2] at 80°C (W)	Room heat demand at -1°C (W)	length x height, type (mm) (mm)	Output[2] at 80°C (W)	Output at 80°C as a proportion of room heat demand (%)	Output[2] at 52°C (W)	Output at 52°C as a proportion of room heat demand (%)
Kitchen			0					
Lounge	1920 x 600, P1	1548	1670	1920 x 750, K2	4746	2.48	2183	1.31
Living Room		0	1380	1920 x 750, K2	4746	3.44	2183	1.52
Hallway/ Landing	1440 x 600, P1	1175	1360	1760 x 750, K2	4355	3.20	2003	1.47
Bathroom	640 x 600, P1	541	610	800 x 750, K2	2001	3.28	920	1.51
Bedroom 1		0	690	1440 x 450, K2	2279	3.30	1048	1.52
Bedroom 2	1280 x 600, P1	1049	960	1280 x 750, K2	3186	3.32	1465	1.53
Bedroom 3	1070 x 620, P1	867	570	960 x 600, K2	1966	3.45	904	1.59
Total		5180	7270		23279	3.05	10706	1.47

NOTES: 1 Radiator types refers to Stelrad's classification: P1 = single panel, K2 = double convector
2 Mean of flow and return water temperatures. A mean of 80°C implies a flow temperature of 85°C. A mean of 52°C implies a flow temperature of 55°C.

TABLE 6.10: SUMMARY OF ORIGINAL AND REVISED RADIATOR CAPACITIES

TEMPERATURE MEASUREMENTS (all°C)

a) Air Temperatures

T_a	ambient inside Stevenson Pattern Screen
T_{a1}	evaporator intake
T_{a2}	evaporator discharge
T_{ar}	reference room air

b) Water Temperatures

T_{w1}	return (condenser inlet)
T_{w2}	condenser outlet (supplementary heater inlet)
T_{w3}	flow (supplementary heater outlet)

c) Refrigerant Temperatures

T_1	suction line
T_2	compressor discharge
T_3	condenser outlet
T_4	evaporator outer circuit inlet

ADDITIONAL INSTANTANEOUS MEASUREMENTS

$\dot{V}_w$	water flow rate	(m^3/h)
$\dot{W}$	heat pump input power	(kW)
Rh	ambient air relative humidity	(%)
ACV	reference mains voltage	(V)

LONG RUN INTEGRATING INSTRUMENTATION

W	electrical energy absorbed by heat pump	(kWh)
W_b	electrical energy absorbed by supplementary heater	(kWh)
$Q_c + Q_b$	heat delivered by the heat pump and supplementary heater combined	(kWh)

TABLE 6.11: VARIABLES MEASURED BY THE PERFORMANCE MONITORING SYSTEM

CHAPTER 7 FIELD TRIAL RESULTS

1. INTRODUCTION

The purpose of this chapter is to present and discuss the detailed findings from the field trial. The main results are given in section 2, in which the performance of the heat pump is examined in detail along with its interactions with the heat distribution system and the building. The results from the trial are compared with the predictions from program DSI, which is described in Chapter 3. Details of the reliability difficulties experienced during the trial and the adopted solutions are given in section 3. Finally, the conclusions are presented in section 4.

1.1 Definition of Two Parts of the Field Trial

As the trial straddled two heating seasons, and because various modifications were carried out between the heating seasons, it is helpful to the presentation of the results to define two parts of the field trial. Part 1 covers the 140 day period 10/2/1985 to 29/6/1985. Part 2 encompasses the entire 1985/86 heating season, from 1/9/1985 to 29/6/1986, a 301 day period.

1.2 Summary of the Major Modifications to the Heat Pump and its Control System

a) During part 1 steps were taken to silence the compressor. These involved modifying the compressor mounting arrangements and installing vibration isolators in the water piping, and are discussed in section 3.2

The modifications b) to e) were carried out during the period between the two parts of the trial:

b) The evaporator distributor tubes were replaced by larger bore versions to improve the defrosting performance, see section 3.1

c) The control system was modified to introduce the end of cycle, short duration, hot gas defrost operation at low ambient temperatures.

d) Rather than being run continuously, the pump was stopped 10 minutes after the end of the heating cycle. It was also stopped during hot gas defrost operations.

e) The power of the crankcase heater was reduced from 60 W to 30 W. This was achieved by replacing the 25 μF crankcase heater capacitor, A, by a 15 μF capacitor, and replacing the 10 μF run capacitor, E, by a 20 μF unit, figure 6.3 refers.

f) In the early stages of part 2 of the trial, two further changes were made to the control system. The first was the abandonment of the selection between the fan-only and hot gas defrost methods when the defrost cycle was initiated by the high evaporator temperature difference detection system. Instead the hot gas mode was always selected when a defrost was demanded during a heating cycle.

g) The second was an alteration to the sequencing of the supplementary heater. This resulted in the heater power level being stepped down one 2 kW stage when the heat pump was cycled off due to high condenser water temperature.

The representative values of the control settings used in each part of the field trial are given in table 7.1. The main difference is the increase in the set point of the room thermostat, from 17.0°C in part 1 to 17.7°C in part 2.

Before the start of part 2 of the field trial the data acquisition system was overhauled and the calibration of the transducers was checked.

1.3 Integrity of the Final Totals of the Heat the Heat Produced and Electricity Consumed

The overall totals of the heat delivered and the energy consumed by the heat pump can be obtained by three separate means:

a) From the self-contained integrating instrumentation.

b) From the running totals of W, Q_c and Q_b which were calculated by the computer controlling the data logger, the 'on-line' integrated totals.

c) From the data processing programs which apply a Trapezium Rule integration on the data stored in the raw data files, the 'post-processed' integrated totals.

A difference between the results from routes b) and c) might be expected since in c) the data sampling frequency is significantly less than in b) and so some detail in the data is lost. For part 2 of the trial the data sampling and recording strategies were changed to minimise this loss of data, as explained in Appendix A.

Table 7.2 presents the results of W, W_b and Q_c + Q_b from the three approaches, for defined periods within both parts of the trial. Examining first the differences between methods b) and c) it is clear that the reduced sampling frequency which applied to the data processing scheme resulted in discrepancies of +0.7 per cent in W, -0.6 per cent in Q_c in part 1, and that the improved methods eliminated these errors for part 2. The difference between the two estimates of Q_b for part 1 arises because the on-line integrating program disregarded negative values of $\dot{q}_c$ and $\dot{q}_b$, whereas the post-processing scheme does allow negative values. Hence the results from the on-line integration tend to overestimate the total of Q_b. However, this explanation does not hold for part 2 of the trial, since the on-line integration was modified to accept negative values. The difference between the on-line and post-processed values of Q_b is 15.6 kWh, which is 2.2 per cent of the gross total 717 kWh of indicated heat lost from the supplementary heater and associated pipework. The reason for the discrepancy in Q_b between the two methods is not clear, but must be a consequence of the reduced data sampling rate.

The results from the post-processing scheme are used to present the detailed findings from the field trial, so it is salutary to compare the totals from the post-processing scheme with those from the independent integrating instrumentation. From table 7.2 the totals for W from the two sources agree to 1.8 per cent for part 1, and 1.2 per cent for part 2. The tolerance on the kWh meter alone is $\pm$ 2 per cent, so both results are within the meter tolerance. Furthermore the slight under-measurement of W by the kWh meter in part 2 of the trial is explained by the observation that the kWh meter was unable to register the 30 watt crankcase heater power. The results of the combined output of the heat pump and supplementary heater, $Q_c + Q_b$, agree with the ISS Clorius heat meter totals to -2.7 per cent for part 1 and -4.0 per cent for part 2. The manufacturer's specification for the heat meter suggests a systematic under-reading by 3.3 per cent for this application, see Appendix A. Furthermore, the heat meter records the heat delivered to the house, rather than the heat produced by the heat pump hence a difference of about 1.5 per cent is to be expected to account for the heat lost from the outside pipe runs. Thus the agreement between the computer totals and the heat meter is within the meter tolerance when allowance is made for the expected pipework loss. The computed totals of the total supplementary heater electricity consumption, W_b, are under-estimates when compared to the kWh meter readings, by 3.3 per cent for part 1 and 2.4 per cent for part 2. These are outside the meter tolerance, and arise because the power consumption of the supplementary heater is not directly measured by the data logger but rather it is deduced from the measured rate of heat output from the supplementary heater. It is to be expected that this indirect method is less precise than the direct kWh meter measurement. Thus in the calculation of the heat pump seasonal COP_h, the kWh meter readings of the energy consumption of the supplementary heater are used in preference to the computer estimated totals, in order to provide the greater accuracy in this key result.

To conclude this discussion on the integrity of overall totals for the trial, it has been shown that there is in general good agreement between the on-line and post-processed integrated totals, and with the only exception of W_b, that the totals from the post-processing scheme agree with the results from the independent integrating instruments to within the accuracy tolerances of these instruments.

2. PRESENTATION AND DISCUSSION OF THE RESULTS

2.1 Overall Performance

The overall totals for the heat produced and the electricity consumed are:

		Part 1	Part 2	
heat produced by heat pump	Q_c	6011	17490	kWh $\pm 3\%$
net heat transferred by supplementary heater	Q_b	618	-76	kWh $\pm 3\%$
total of heat produced	$Q_c + Q_b$	6629	17414	kWh $\pm 3\%$
electricity consumed by heat pump	W	2529	6792	kWh $\pm 1.5\%$
electricity consumed by supplementary heater*	W_b	582.6	656.7	kWh $\pm 2\%$
total electricity consumption	$W + W_b$	3111.6	7448.7	kWh $\pm 1.6\%$
seasonal $COP_h = \dfrac{Q_c + Q_b}{W + W_b}$		2.130	2.338	$\pm 4.6\%$

* As explained in the previous section, for greater precision the W_b results are taken from the kWh meter readings.

Hence the achieved seasonal COP_h was 2.13 for part of the 1984/85 heating season, and 2.34 for the complete 1985/86 season. The difference between the two results partly reflects the differences in timing of part 1, during which, because it did not start until February, the heat pump was exposed to a greater proportion of cold days than was the case in part 2. A further contributory factor was the failure of the defrosting system to consistently remove all the evaporator ice blockage, which occurred during low ambient temperature conditions in part 1 only. This initial failure of the defrost system resulted in a greater proportion of supplementary heating than would otherwise have been the case, hence the lower COP_h result. The defrost system is discussed further in section 3.

The relationship between the supplementary heater electricity consumption, W_b, and its total heat output, Q_b, merits comment. The results show that in part 1 the heat output slightly exceeded the electricity consumption, while in part 2 an apparent heat loss from the supplementary heater was observed, which when taken across the entire heating season, was greater than the electricity consumption. It is thought that this difference was caused by the combination of two modifications to the prototype unit. The first was the decrease in the crankcase heater output, from 60 to 30 W, which was implemented between the two parts of the trial. This would lead to a decrease in the air temperature within the compartment which houses the compressor and flow boiler, and so reduce the opportunity to recover a small amount of heat by the water flowing through the flow boiler. Secondly, midway through the first part of the trial, the pipework connecting the heat pump to the radiator network was altered by the addition of a 1 m flexible hose link in each of the flow and return pipes at the point of connection to the heat pump, in order to reduce the vibration transmitted to the building. The flexible hoses were positioned between the water thermometers T_{w1} and T_{w3}, and the heat pump, such that any heat loss from these hoses would affect the indicated heat output rate from the condenser and the supplementary heater. Furthermore, it was difficult to thermally insulate the hoses to high standards. Hence it is suggested that the slight gain in energy due to heat recovery from a warm compartment was converted into a measureable heat loss because of both the reduction in crankcase heater power and the insertion of poorly insulated vibration isolators into the pipework.

The major energy flows for part 2 of the trial are represented in Sankey diagram form in figure 7.1. The proportions of the total electrical input which were consumed by supplementary heating, the compressor stand-by function (which is mainly the energy consumed by the crankcase heater) and defrosting are 8.8, 2.7 and 5.6 per cent respectively. The equivalent proportions for part 1 were 19.0, 5.7 and 6.8 per cent respectively. Thus, compared with part 1, the later results demonstrate a significant reduction in the proportion of supplementary heating, a reduction in stand-by losses following the down-rating of the crankcase heater, and a small reduction in defrost losses following the modification to the evaporator distributor tubing. However, the apparent reduction in the contribution of the supplementary heater is partly a factor of timing, since in both parts of the trial the supplementary heating was mainly required in the month of February, and since part 1 started in this month it will clearly show a greater proportion of supplementary heating than part 2 which encompassed the entire heating season.

By subtracting the energies involved in the stand-by, defrost and supplementary heating functions, and incorporating the known pipework heat loss into the heat pump output total, it is clear that for part 2 6177 kWh of electricity was required to produce 17490 kWh of heat. Thus the COP_h of the vapour compression cycle alone, based on the electricity consumed by the compressor and fan in heating mode, is 2.83. This drops to 2.48 when the pipework heat loss and the parasitic energy consumption for the stand-by and defrosting functions are included. Finally, when supplementary heating is included, the COP_h falls to the overall result of 2.34.

A detailed breakdown of the heat output, electricity consumption and operating patterns against ambient temperature is presented in table 7.3. Some of the results from this table, for part 2, are summarised in figures 7.2 and 7.3. Figure 7.2 shows the heat delivered by the system against ambient temperature interval. The greatest heat demand occurred at temperatures between 3°C and 4°C, whereas the analysis in Chapter 3, based on the 12 year average temperature variations for the region, suggested this peak would occur between 4°C and 5°C. The supplementary heater operated at ambient temperatures of 2°C and below, with its highest heat output in absolute terms occurring between -3°C to -2°C, and the greatest contribution to the total heat delivered was 38.4 per cent for the lowest daily mean temperature of -7.5°C.

Figure 7.3 shows the electricity consumed by the heat pump for each ambient temperature interval, for the four modes of operation of heating, supplementary heating, defrosting and stand-by. The consumption patterns for heating mode and supplementary heating resemble those of the heat output discussed above. Defrosting activity is distributed about 0°C, in line with the predictions given in Chapter 4, although the greatest energy consumption occurred between 1°C to 2°C. The energy consumed in stand-by mode, as might be expected, is skewed towards the higher ambient temperatures. Table 7.3 also shows the run-hours in heating and defrosting modes, along with the number of heating and defrost cycle starts. From these figures, for part 2 of the trial, the average run time per heating cycle is 41.4 minutes and the average starting frequency is 13 starts per day. The heat pump was in operation, i.e. in either heating or defrost mode, for 25.2 per cent of the total time.

The table also indicates the proportion of electricity which was consumed while the low cost off-peak tariff was in force. In part 1 this proportion was 38.2 per cent, and the corresponding figure for part 2 is 35.0 per cent. In both parts of the trial more than half the total consumption of electricity by the supplementary heater was off-peak, demonstrating the usefulness of the control option which permits the user to specify the time of day during which the supplementary heater may be activated.

An indication of the ambient temperatures which prevailed across the two heating seasons of the trial is given in table 7.4, which presents the monthly degree data for the 1984/85 and 1985/86 winters along with the 20 year average data (100). The table shows that both winters were more severe than the typical winter, the 1985/86 season particularly so, with a 13 per cent higher degree day total.

Both heating seasons were characterised by very cold periods in February. At the very start of the field trial, in February 1985, the daily mean ambient temperature remained below freezing for eleven days, and five of these days had mean temperatures of less than -4°C. February 1986 was even more severe. According to the Meteorological Office, for England and Wales as a whole, February 1986 was the second coldest this century, and was the sixth coldest since detailed records began in 1659 (101). For the Midlands region, the average air temperatures for this month were 4.9 deg C below the 30 year average figure of 4.0°C. How the prototype heat pump coped with these conditions is illustrated in figure 7.4, which is a plot of the daily mean room (hallway) and ambient temperatures during the trial. Throughout the trial the heat pump demonstrated its ability to maintain comfortable room temperatures. Any departures from the set point conditions were due to the way the system was operated rather than any shortcomings in the equipment. The lower mean room temperature during the period from day 58 to 76 was a result of the heating system being turned off overnight because of initial difficulties with noise disturbance. This is the only period of the trial for which the heating system was not operated on a continuous basis. The drop in room temperature between day 325 and 328 occurred while the house was unoccupied and the control option prohibiting the supplementary heater was in force. Thus the sudden drop in ambient temperature led to a situation in which the output capacity of the heat pump was insufficient to maintain the room temperatures and yet the supplementary heater was disabled. Under normal occupancy the supplementary heater would have been enabled, at least for off-peak operation, and comfort conditions would have been maintained. The mean room temperature for February 1986 shows a slight depression of about 1 deg C in response to the low ambient temperatures. The heat supplied to the house was 119.0 kWh per day, averaged across the month, and the supplementary heater supplied 18.7 per cent of this. Some 85.1 per cent of the electricity used by the supplementary heater in part 2 was consumed during this month.

Hence the heat pump has demonstrated its effectiveness as a house heating source and produced an overall COP_h of 2.34 for an unusually harsh heating season. This result compares favourably with the outcome of a field trial of 16 heat pumps of UK origin, funded by the Department of Energy and carried out by the Electricity Council (102). The average COP_h of all 16 heat pumps taken across the two heating seasons 1983/84 and 1984/85 was 1.9. The best unit achieved a COP_h of 2.2 over these periods.

2.2 Heat Pump Performance and Operational Characteristics

The COP_h of the heat pump is plotted against the daily mean ambient temperature in three different forms in figures 7.5 to 7.7, as follows:

a) The device $COP_h = \dfrac{Q_c + Q_b}{W + W_b}$

i.e. the ratio of the combined daily heat output of the heat pump and supplementary heater to the total electricity consumption, figure 7.5.

b) The heat pump $COP_h = \frac{Q_c}{W}$

i.e. the ratio of the daily heat output from the heat pump only to the total electricity consumption of the heat pump, figure 7.6.

c) The cycle $COP_h = \frac{Q_c}{W_h}$

where W_h is the electricity consumed per day while the heat pump is producing useful heat, i.e. stand-by and defrosting losses are excluded, figure 7.7.

The device COP_h exhibits a more marked fall at low ambient tempertures compared with the heat pump COP_h, because of the rapid fall in device COP_h when supplementary heating is used. Figures 7.5 and 7.6 exhibit a maximum COP_h at about 11°C, above which the stand-by loss becomes a significant proportion of the total electricity consumption, with the result that the COP_h declines. It is apparent from figure 7.5 that the COP_h results for part 1 at temperatures below 5°C are lower than those for part 2. This is thought to be related to the defrosting difficulties which beset the first part of the trial, see section 3.1. The inability of the defrost system to completely clear the evaporator ice blockage led to a reduction in the device COP_h by the combined effects of three influences; the reduction in the available evaporator surface area and so fall in heat pump COP_h and output, the increase in supplementary heating to offset the shortfall in output, and thirdly, the increase in energy consumed in defrost mode by attempts to clear the evaporator.

The cycle COP_h, figure 7.7, shows a linear rise with ambient temperature, from just over 2.0 at -7.5°C to nearly 4.0 at 15°C.

The combined mean heat output rate of the heat pump and supplementary heater is plotted against the daily mean ambient air temperature in figure 7.8. The mean heat output rate is calculated by dividing the daily total of $Q_c + Q_b$ by the total time in heating mode for the day. For ambient temperatures above 2°C the results confirm the expected rise in heat output capacity with temperature. Below 2°C the additional heat output from the supplementary heater is evident.

The compressor starting frequency is plotted against the daily mean ambient temperature in figure 7.9. The maximum frequency is about 25 starts per day and occurs at 4°C. At higher temperatures the starting frequency falls in line with the reduced activity of the heating system. At lower temperatures the starting frequency falls because of the longer heat pump run times. The observation of low starting frequencies at low ambient temperatures is evidence that the heat pump was not cycled on/off by the condenser water outlet thermostat to any significant extent.

The mean heat pump run time per heating cycle is plotted against ambient temperature in figure 7.10. As expected long run times are observed at low temperatures. Above 2°C, however, the mean run time rapidly falls to a minimum of about 24 minutes. Clearly this minimum run time is related to the combined response time of the heat distribution system and the building structure.

The load factor is defined as the ratio of the total time in a day for which the heat pump is active, i.e. in either heating or defrosting modes, to the total time of 24 hours. The load factor results are plotted against the daily mean ambient temperature in figure 7.11. This shows that the load factor is less than 50 per cent at temperatures of 5°C and above, and that the equipment is fully utilized at -1°C and below.

2.3 Interactions Between the Heat Pump, Heat Distribution System and the Building

2.3.1 Performance Characteristics of the Heat Distribution System

The measured heat output characteristics of the radiator system are illustrated in figure 7.12 which is a plot of the daily mean heat output rate from the heat pump and supplementary heater against the daily mean of the avarage heat sink water temperature, $(T_{W1} + T_{W3})/2$. Only the results from part 1 of the trial are included in this presentation because for that part the water circulation pump was operated continuously and so the water temperatures were sampled at all times by the heat sink thermometers. The expected heat output characteristic is also shown in the figure. This is calculated from the known total installed radiator area and the relationship between water temperature and heat output given by the radiator manufacturer (38). A constant room air temperature of 20°C is assumed in this calculation. The figure indicates that the achieved output is slightly less than expected at the higher water temperatures. This may in part be due to the action of the thermostatic radiator valves which are capable of reducing the output of individual radiators. The heat output rate is approximately 9.2 kW at a mean water temperature of 52°C.

The figure also highlights the generally low mean heat sink temperatures which were achieved in this installation, for example the highest mean water temperature is less than 44°C. However, the instantaneous water temperatures were at times noticeably higher than this, for instance at low ambient temperatures the water discharge temperature from the device was often in the range 50°C to 60°C towards the end of a heating cycle.

Figure 7.13 compares the daily mean of the average heat sink water temperature with the average water temperature evaluated over the time the heat pump is producing heat. The deviation between the long run mean and the heating mean is small at low ambient temperatures when the heat pump is in heating mode almost continuously, but becomes more pronounced at high ambient temperatures when the heat pump is cycled on/off. For example, at an ambient temperature of 5°C the mean water temperature to which the heat pump must deliver heat is approximately 6.5 deg C higher than the long run water temperature at which the radiator network can dissipate sufficient heat to satisfy the building heat load. This difference rises to 10 deg C at an ambient temperature of 10°C. This effect reduces the COP_h of the heat pump under part load conditions, as it requires the device to produce a higher output temperature than is strictly necessary. However, this effect will always occur in installations employing the simple on/off method of output modulation, although as indicated in Chapter 3, its consequences may be reduced by the combination of a high heat sink thermal capacity, low building thermal capacity and a small switching differential on the room thermostat.

A development of the present on/off control strategy would be to limit the on-period under part load conditions. This would tend to increase the starting frequency but reduce the run time per cycle and so reduce the temperature elevation of the heat sink. Against this suggestion is the deterioration in the quality of control over the house internal temperatures, which the occupants may find objectionable, and possibly, reliability problems arising from higher starting frequency of the compressor. A better approach is to contrive direct capacity control over the heat pump so that the instantaneous heat output can be matched to the building heat demand. Under such a scheme the heat pump could run continuously and so the water temperature would not deviate from the long run requirement. Experimental capacity controlled heat pumps have been built. For example, Wilson et al used an open compressor driven by a speed regulated d.c. motor, a speed controlled fan, a motor operated expansion valve together with a microprocessor based control system (103). Unfortunately the performance benefits accruing from such a development tend to be outweighed by the increase in cost of the system and the decrease in reliability.

2.3.2 Building Heating Requirement and System Balance Point

The average rate of heat transferred to the building for each day of the trial is plotted against the ambient temperature in figure 7.14. As expected the house requires very little active heating above about 15°C, because incidental heat gains are sufficient to maintain comfortable internal conditions. The scatter in the data illustrates the influence of the uncontrolled variables; air infiltration rate, wind speed and direction, incidental heat gains, etc. The solid line drawn through the points attempts to define the representative heat requirement characteristic of the building. Also shown is the expected heat requirement obtained by calculation, see Chapter 6. At -1°C the actual requirement is approximately 5.2 kW, compared with the calculated 7.3 kW, a difference of 29 per cent. It is thought that this discrepancy is due to a combination of the following factors:

a) Solar gains. The calculation of the house heating requirement included incidental heat gains of 1 kW in total, but as the calculation was performed with the objective of selecting the appropriate radiator sizes, night-time conditions were assumed. In practice an appreciable benefit from solar gains was obtained owing to the orientation and the high thermal capacity of the house.

b) Ventilation rates. A high air change rate was assumed in the calculation, such that at the design temperature about 2.7 kW was needed to heat the incoming air to room temperature. While no further assessment of the ventilation rate has been made, clearly the discrepancy between the measured and expected heating requirement may in part be attributable to an overestimation of the ventilation rate.

c) Internal temperatures. The room temperature control was located in the hallway, which has the lowest design room temperature, and it was noticed that in mild ambient conditions the air temperatures of the other rooms in the house tended to drift down towards the hallway temperature. In other words the hallway temperature was maintained above the 'thermostat' set point by passive heating with the result that the heating system did not operate to maintain the temperature elevation of the other rooms. This effect, although only relevant at ambient temperatures above 12°C, contributes to a reduction in the heating requirement, at the expense of a slight lowering of occupancy comfort levels.

Figure 7.15 is the actual balance point diagram for the installation. It is constructed from the actual heating requirement of the building, obtained from figure 7.14, and the heat output characteristics of the heat pump, as shown in figure 7.8. The balance point diagram indicates that the output capacity of the heat pump is sufficient to satisfy the building requirement down to an ambient temperature of -1.8°C. However, in practice the supplementary heater is brought into service by the control system at the higher ambient temperature of 2°C due to two factors. The first is the long response time of the heating system which occurs when the heat pump output is only just greater than the building heating requirement. Supplementary heating is required under these circumstances to achieve comfortable room temperatures within a reasonable period of time, and is brought into effect by the time delay control. The second factor is the effective reduction in heat pump heat output capacity which occurs as a result of defrosting. This extends the time that the room thermostat calls for heat and so increases the likelihood that the supplementary heater will be used.

2.3.3 Comparison Between the Results for Part 2 of the Field Trial and the Predictions from Program DSI

Program DSI is a simulation exercise which models the performance of the heat pump as a component in a house central heating system. Full details of the analysis are given in Chapter 3. The level of detail in the results of the field trial offers an opportunity to verify the predictive accuracy of this program.

The program was executed for four sets of input data as follows:

Run A The input data were the results of the design calculations on the installation, as presented in Chapter 6, e.g. the heating demand of the building at -1°C was set to 7.3 kW, the combined heat output from the installed radiators was 10.7 kW at a water flow temperature of 55°C. The ambient temperature frequency data were the long run average figures for the region, i.e. a typical heating season is simulated.

Run B As A except the actual house heat loss at -1°C of 5.2 kW was used in preference to the calculated figure.

Run C As B except the measured radiator output rate, as shown in figure 7.12, was used in place of the estimated output.

Run D As C except the typical ambient temperature frequency data were replaced by the actual data for the 1985/86 heating season.

These input data are summarised in table 7.5 along with the results of the simulation runs and the relevant measurements from part 2 of the field trial. In comparing the predictions with the actual measurements the simplifying omissions of the simulation must be taken into account. These are; stray heat losses from the water pipework are negligible, and the consumption of energy in the stand-by and defrosting modes is not assessed. Thus, for a valid comparison, the measured results must be adjusted in two senses; the known pipework heat loss must be added to the condenser heat output to determine the total condenser heat output, and only the electricity consumed by the heat pump in the heating mode must be considered.

The heat output predictions from run A are considerably in excess of the actual totals because of the overestimated house heating requirement. Run B overestimates the COP_h of the heat pump because the actual performance of the radiator system fell short of the expectations. Run C, using the measured values of the house heating requirement and radiator characteristics, still overestimates the COP_h because the actual heating season was atypical in relation to the greater incidence of low ambient temperatures. Run D demonstrates good agreement with the measured results; Q_c is 1.3 per cent higher than the measured value, Q_b is 35.1 per cent higher giving a heat output total which is 2.5 per cent higher than the measured total. The program predicts the heat pump heating mode electricity consumption to an accuracy of 0.1 per cent, and the cycle COP_h is estimated to be 1.4 per cent greater than the achieved value. The good agreement between run D and the measured results is a demonstration of the predictive accuracy of the simulation program; with the exception of Q_b the agreement is within the measurement uncertainties of the measured values. However, this validation exercise also illustrates the importance of the quality of the input data. In the absence of the measured characteristics of the building and the radiator system, the results from the design calculations which are used in run A are a defendable set of figures, but based on these the predicted cycle COP_h is almost 8 per cent in error and the predicted total heat output is 36 per cent too high.

The predictions from run C provide an indication of the seasonal COP_h of the heat pump in an average heating season. The results of this run can be adjusted to allow for water pipework heat loss and heat pump stand-by and defrosting electricity consumption by applying factors derived from the part 2 measurements. For example the pipework heat loss is 4.0 per cent of the total heat output, and the electricity consumption in the stand-by and defrost modes together represents 10.0 per cent of the heating mode consumption. Thus the adjusted results from run C suggest that, for a typical heating season, the total heat delivered by the device would be 17261 kWh, of which 1.9 per cent is provided by the supplementary heater. The electricity consumption of the heat pump, including stand-by, defrost and heating operations would be 6563 kWh. Thus the COP_h of the device, including supplementary heating, would be 2.51.

3. FURTHER FINDINGS

3.1 The Hot Gas Defrost System

As mentioned previously, the hot gas defrost system was found to be ineffective in the first part of the field trial. This prompted a closer examination of the defrosting method, including further experimental studies, which eventually led to a modification to the evaporator distributor as well as to changes in the control system. These modifications were carried out before part 2 of the trial started, and the revised arrangement proved satisfactory throughout the remainder of the trial.

This section reports the observations made on the defrost system during part 1 of the trial, summarises the results of the analytical and the experimental studies, and details the modifications to the heat pump and its control system.

3.1.1 Observations Made During Part 1 of the Field Trial

Soon after the start of the field trial, in Feburary 1985, a gradual accumulation of ice developed on the lower half of the evaporator. In hot gas defrost mode the frost and ice on the top turns of the coil, near the refrigerant inlet, were rapidly melted by the flow of hot refrigerant vapour, but the lower turns were not cleared, even after the 30 minute defrost cycle time. Under the worst conditions, it was estimated that 40 per cent of the evaporator area was blocked by the ice. The consequence of this blockage was a reduction in heat pump heat output capacity and therefore an increase in the contribution from the supplementary heater, which partly explains the lower device COP_h result for part 1.

Examination of the transient data records indicated that the frost accumulation rate, as indicated by the rate of increase in the air to refrigerant evaporating temperature difference, was consistent with the previous measurements on the ETR, as reported in Chapter 4. The frost accumulation rate was generally quite high, which is a consequence of the low condenser water temperatures brought about by the large radiator surface area in this installation. The transient data also demonstrated that the defrost initiation and mode selection functions of the control system were operating correctly. The problem appeared to centre on the excessive fall in temperature of the refrigerant vapour as it was circulated through the evaporator in hot gas defrost mode. The refrigerant inlet temperature was generally 65°C to 85°C. However, this fell to about -5°C at the outlet, which is clearly of little value in melting ice. The problem was most acute at sub-zero ambient temperatures, as a low air temperature tends to promote heat loss from the vapour in transit through the evaporator circuits and so further depresses the outlet temperature.

3.1.2 Analysis

Suppose at the start of the defrost cycle the evaporator is at a temperature of -10°C. The evaporator itself has a mass of 30 kg and suppose there is a further 2 kg of ice attached to it. Then the energy required to raise the temperature of the coil, ice and refrigerant to 0°C and then to melt the ice is roughly 800 kJ. The hot gas defrost cycle is capable of supplying energy to the evaporator at a rate of about 2 kW, hence ignoring subsequent heat losses from the coil to the surrounding air, a complete defrost would be accomplished in about 7 minutes. Heat losses, of course, extend this period, but clearly the pre-set duration of 30 minutes used during the trial ought to have been sufficient.

The reason for the inadequacy of the defrost system is the low mass flow rate of the refrigerant vapour through the evaporator in defrost mode. At low vapour flow rates the enthalpy supplied to the working fluid during compression appears in the evaporator as a large change in the fluid temperature between the inlet and outlet. In practice the temperature change was such that at a position roughly 60 per cent along the evaporator circuits the refrigerant vapour temperature fell below 0°C, and so the remaining downstream circuits were not defrosted. Under low vapour flow conditions a significant proportion of the supplied enthalpy appears as heat lost in the inlet areas of the evaporator, because of the high refrigerant inlet temperatures.

The refrigerant mass flow rate may be increased by elevating the compressor discharge pressure. Provided liquid refrigerant exists in the high pressure regions of the system, the compressor discharge pressure during hot gas defrosting is fixed by the saturated pressure in the condenser, which in turn is governed by the water temperature in the condenser. At high water temperatures a high compressor discharge pressure is generated and this increases the mass flow rate and so improves the defrosting performance. However, while this effect is important, merely ensuring high condenser temperatures by itself does not guarantee successful defrosting.

The key reason for the generally low vapour mass flow rates in defrost mode is the influence of the small bore distributor tubes, which split the incoming flow evenly between the two evaporator circuits. No operational problems arose with two phase flow through these tubes in heating mode, however, in defrost mode for pure vapour flow with a considerably lower density the acceleration of the flow is high enough for Mach numbers of unity to occur within the distributor tubes. Analysis of compressible gas flow through these restrictors indicated that the flow is choked under hot gas bypass conditions. The mass flow rate is therefore limited, and in an effort to balance the flow through the restriction the action of the compressor will tend to reduce the suction pressure. Low suction pressure leads to a marked reduction in the mass throughput of the compressor because of the ensuant reduction in vapour density. The solution is to reduce the effect of the restriction by changing its pressure drop characteristics, and this was further investigated experimentally.

3.1.3 Experimental Results

In order to determine a suitable design of distributor tubing the Evaporator Test Rig was operated with several different sizes of distributor tubing, designated A to F and defined in table 4.8. The original distributor tubes used on the prototype are designated F.

The operating conditions in defrost mode are shown in figure 7.16. In all cases the ambient conditions were -5°C, 95 per cent relative humidity, and the water inlet temperature was 30°C, which represent the worst case conditions for defrosting. As the distributor of a given bore is reduced in length the pressure drop falls and the mass flow rate increases, but this effect is less pronounced than that attributable to increasing the bore of the tube. The mass flow rate for configuration D, which has a bore of 4.9 mm, was 3.5 times that for the original distributor used on the prototype, which had a bore of 3.3 mm.

The observed defrosting rate in these experiments confirmed the importance of achieving a high vapour flow rate. While distributor D would take about 15-20 minutes to remove the frost layer, version C would take about twice as long, and the other configurations did not achieve a complete defrost even after 90 minutes of operation. Long run endurance testing over several days of low air temperature operation confirmed the effectiveness of distributor D.

3.1.4 Modifications to the Prototype Heat Pump Defrost and Control Systems

The distributor tubes were replaced by the $^1/_4$ inch outside diameter, 1m long version, i.e. distributor D. The control system was modified as follows:

a) At low ambient temperatures a short duration hot gas defrost operation would be implemented at the end of each heating cycle. This is an extension of the fan-only end of cycle defrost feature. Its purpose is to alleviate the defrost problem by adding extra defrost cycles when the heat pump is not required to heat the building. The duration of the end of cycle defrost operation was set to 5 minutes. Selection between the hot gas or fan only modes for this end of cycle defrost operation was made by the ambient air 'thermostat', set to 4°C.

b) The control logic was changed so that whenever a defrost was initiated during the heating cycle then the hot gas bypass mode would always be selected, independently of the air temperature. Experience indicated that a defrost was initiated in those circumstances which demanded the more positive defrosting method.

c) The water circulation pump was stopped during the defrost cycle, in order to maintain the condenser at high temperature.

3.1.5 Subsequent Operation of the Defrost System

As indicated previously the revised defrosting arrangements proved successful throughout part 2 of the field trial. Even during the unusually low ambient temperatures of February 1986 the evaporator emerged with only a trace of ice accumulation on the lowest turn. The refrigerant temperatures at the evaporator inlet and outlet during hot gas defrosting are illustrated in figure 7.17. The ordinate scale for all four plots is the number of four hour periods for which the time-averaged refrigerant temperature in hot gas defrost mode fell within particular intervals, expressed as a percentage of the total number of four hour periods which contained defrosting activity. The figure demonstrates that the result of the modification to the distributor is a decrease in the refrigerant evaporator inlet temperature and an increase in the evaporator outlet temperature. The most frequent mean evaporator outlet temperature for part 2 was -2°C, and the transient data indicates that, in the majority of cases, this temperature was well above zero by the end of the defrost cycle.

3.2 Noise

Difficulty was experienced with the noise generated by the compressor at the initial stages of the first part of the field trial. A low frequency noise was audible inside the nearby dwellings which was a nuisance during night-time operation.

Noise measurements of the heat pump in operation at the test site location were taken by ERC's Acoustics and Vibration Group. The measurements were taken before the heat pump was coupled to the radiator system. The results are summarised in table 7.6. The table contains measurements for the ambient noise levels, for the fan and compressor operating together, and when a layer of sandbags was positioned around the base of the unit in an attempt to reduce the noise levels. The measurements taken inside the downstairs room facing the heat pump indicate that the compressor noise was being transmitted to the room, but the general levels were low and the sandbag layer appeared to provide some attenuation. Figure 7.18 shows the frequency and spatial analysis of the recordings made 1m from the unit with the fan and compressor running. The sandbags were not in place for these readings. The peaks at the low frequencies are caused by the compressor.

Following the connection between the heat pump and the radiator system, the sound pressure levels inside the house were measured by the author at times during the night when the noise nuisance was apparent to the occupants, using a portable single octave analyser. The overall sound pressure level was low, around 27 to 30 dB(A), and at this level it was difficult to take precise readings. However, the instrument did indicate an increase in low frequency sound when the heat pump was running.

The solution to the compressor noise problem had two components:

a) Revised compressor mounting arrangements. Three brass posts were machined and screwed onto the protruding threads of the compressor mounting bolts on the underside of the base plate. These posts had the effect of grounding the compressor mountings to the concrete surface beneath the heat pump.

b) Pipework vibration isolators. Lengths of flexible tubing were installed in the water pipe runs at the connections to the heat pump. This was to prevent vibration being transmitted through the pipework to the building.

These measures were successful in eliminating the noise nuisance. The wall of sandbags around the compressor compartment was removed once the these steps had been implemented. The heat pump operated for the remainder of part 1 and for all of part 2 without further complaints, as it was no longer audible from inside the nearby houses.

3.3 Electrical Trips

3.3.1 Compressor Internal Overload (IOL)

This is a current and temperature sensing bimetallic strip located in the common connection in the compressor windings. It operated on 6 separate occasions during part 1 and for some of these it opened several times, generally soon after the compressor started. The switch is self resetting but has a large switching differential, hence once opened it could take 1 to 2 hours to re-close. The compressor is disabled while the IOL switch is open. The solution, at least in part, was to change the orientation of the compressor starting relay, as suggested by the compressor supplier. Apparently this relay has been known to occasionally stick in the start position unless gravity assists the opening of the contacts. This change, which was carried out 6 weeks into the first part of the trial, put an end to the spate of IOL operations.

However towards the end of part 1, the IOL operated again on two occasions. The circumstances of these two events differed from the earlier experience in that they were characterised by the compressor starting following an extended period in stand-by mode. Ambient temperatures were reasonably high and the compressor had received continuous crankcase heating for in excess of 15 hours in each case. Although the compressor case temperature was not measured, the winding resistance had increased significantly, indicating high internal temperatures which would lead to the IOL operating. A further contributory factor to these later IOL events is thought to be the effect of the soft start device on windings temperature. For about 2 seconds a current flows through the auxiliary windings prior to the compressor rotating, due to the time taken for the voltage applied to the primary windings to reach a sufficient level to turn the rotor. The auxiliary current causes a further increase in the windings temperature, and this coupled with the already hot compressor due to excessive crankcase heating caused the IOL to operate.

The solution to this problem was to halve the crankcase heater power input, to 30 W, by changing the capacitors in the compressor starting circuit. This had the additional benefit of reducing the parasitic energy consumption in stand-by mode. No further IOL trips of this kind occurred once the crankcase heater power was reduced.

Two further IOL trips occurred in part 2 of the trial, both following extended compressor run times at low ambient temperature. Since the events occurred when the compressor had been running for at least one hour, the compressor starting relay or soft start device are unlikely to be responsible, but the actual cause is not known.

3.3.2 Low Pressure Switch (LP)

The LP trip operated once in part 1 and twice during part 2 of the trial. In all three cases the ambient temperatures were extremely low. However, the transient data show that the trip was operating at a higher pressure than was specified, at 1.4 bar, gauge (-20°C saturation temperature), compared to 1.0 bar, gauge (-25°C).

3.3.3 Residual Current Circuit Breaker (RCCB)

This trip interrupts the supply to the heat pump if the inbalance between the current detected in the live and neutral lines exceeds 30 mA. It operated seven times in the first part of the field trial, and each trip occurred when the fan motor was switched off at the end of cycle. The problem was solved by rewiring the fan contactor so that it interrupted both the live and neutral connections to the fan motor. No further RCCB trips occurred once this alteration had been made.

4. CONCLUSIONS

1. The heat pump achieved an acceptable seasonal COP_h of 2.34 for the 1985/86 heating season which contained an unusually long period of low ambient temperatures. The essential characteristics of the installation are a building heat load of 5.2 kW at -1°C and a large radiator area capable of dissipating 9.2 kW at a mean water temperature of 52°C. Comfortable internal temperatures were

demanded for 24 hours per day, throughout the heating season. The total heat delivered to the building in part 2 was 17414 kWh, of which 3.8 per cent was produced by the supplementary heater. A heat loss from inadequately lagged water pipework vibration isolators amounted to 732.7 kWh, and this is, in principle, recoverable. Some 85.1 per cent of the electricity used by the supplementary heater was consumed during February 1986. Advantage was taken of the favourable off-peak electricity tariff; 33 per cent of the heat pump consumption and 57.2 per cent of the supplementary heater consumption were obtained at the off-peak rate.

2. The integrity of the heat pump as the heating source for domestic central heating has been demonstrated. The device was able to maintain comfortable house temperatures in all ambient conditions virtually throughout both parts of the field trial.

3. The main shortcoming of the initial design was the failure of the hot gas defrost system to keep the evaporator clear of ice. This was overcome by increasing the bore and decreasing the length of the evaporator distributor tubes. The modified arrangement proved successful in part 2 of the trial.

4. Compressor noise levels were found to be a nuisance to nearby residents during night-time operation. This was resolved by modifying the compressor mountings and by installing vibration isolators in the water pipes.

5. Reliability problems centred on a number of operations of the electrical trips. Compressor internal overload trips were caused by the incorrect orientation of the start relay, and by overheating produced by the crankcase heater. The crankcase heater power was halved as a result. Two further internal overload trips occurred in part 2 of the trial for unknown reasons. Switching off the fan motor caused the residual current circuit breaker to operate on several occasions, and this was solved by rewiring the fan contactor so that both the live and neutral connections to the fan motor were interrupted.

6. The control strategy and its implementation worked well. Revisions to the strategy were centred on the defrost feature:

   a) End of cycle, short duration hot gas defrost operation as a matter of routine at low ambient temperatures.

   b) The automatic selection of the hot gas defrost mode when a defrost is initiated during the heating cycle.

   c) Stop the water circulation pump during hot gas defrost operations.

7. The high level of detail in the results from the performance monitoring system has allowed the performance and behaviour of the heat pump together with the heat distribution system and the building to be studied. It has been shown that defrosting and stand-by losses together amounted to 10 per cent of the electricity consumed by the compressor and fan in heating mode. The maximum compressor starting frequency was 25 starts per day which occurred at an ambient temperature of about 4°C. The average run time for the heat pump was 41.4 minutes.

8. The predictive accuracy of the simulation program DSI has been demonstrated. However, the shortcomings in the quality of input data supplied to this program have also been highlighted. For example, the measured building heat loss characteristics and the radiator output performance differ from the design values.

CONTROL	Part 1 10/2/85 to 29/6/85	Part 2 1/9/85 to 29/6/86
a) delay timers	settings in minutes	
Dt1 end of cycle defrost duration	5	5
Dt2 Compressor end of cycle disable time and pump end of cycle overrun time	10	10
Dt3 defrost status check disable time	10	10
Dt4 hot gas bypass defrost duration	30	30
Dt5 fan only defrost watchdog timer	20	20
Dt6 supplementary heater disable time	60	60
Dt7 supplementary heater power increase disable time	15	60
b) thermostats	set point[1] (°C)/differential (deg C)	
T_{ar} room air	17.0/0.2	17.7/0.2
T_{w2} condenser water outlet	55.0/6.0	55.0/6.0
T_{w3} supplementary heater outlet	85.0/6.0	65.0/6.0
T_{w1} ice protection	20.0/15.0	20.0/15.0
T_{a1}-T_4 defrost initiation	14.0/12.0	12.0/7.0
T_{a1} defrost mode selection	6.0/1.0	4.0/1.0

NOTE:

1. The set point is the temperature at which the thermostat opens.

TABLE 7.1: REPRESENTATIVE VALUES OF THE CONTROL SETTINGS USED IN EACH PART OF THE FIELD TRIAL

	W Electricity consumed by heat pump (kWh)	W_b Electricity consumed by supplementary heater (kWh)	$Q_c + Q_b$ Heat produced by heat pump and supplementary heater (kWh)
Period of comparison	10/2/85, 10:00 GMT to 30/6/85, 9:14 GMT		14/4/85, 8:05 GMT to 30/6/85, 9:14 GMT[1]
Results from independent integrating instrumentation (kWh meters and ISS Clorius[2] heat meter)	2467.1	582.6	2031
Results from on-line computer integration	2495	-	Q_c[3] 2096 Q_b 40 2136
Results from data processing scheme	2512.1	568.5	Q_c 2083.5 Q_b 3.9 2087.4

NOTES:
1. Reduced period as heat meter did not initially operate continuously.
2. Results from the HG heat meter are not shown because this meter suffered from obvious malfunction.
3. Known to be an overestimate, as the on-line integration disregarded negative values of $\dot{q}_c$ and $\dot{q}_b$.

a) Part 1 of the Field Trial

	W Electricity consumed by heat pump (kWh)	W_b Electricity consumed by supplementary heater (kWh)	$Q_c + Q_b$ Heat produced by heat pump and supplementary heater (kWh)
Period of comparison	1/9/85, 00:00 GMT to 29/6/86. 00:00 GMT		
Results from independent integrating instrumentation	6708.5	656.7	16721
Results from on-line computer integration	6790.5	-	Q_c 17497.7 Q_b -60.4 17437.3
Results from data processing scheme	6792	635	Q_c 17490 Q_b -76 17414

b) Part 2 of the Field Trial

TABLE 7.2: COMPARISON BETWEEN THE HEAT OUTPUT AND ELECTRICITY CONSUMPTION TOTALS FROM THE INTEGRATING INSTRUMENTATION, ON-LINE COMPUTER INTEGRATION AND THE POST-PROCESSING SCHEME

Daily mean ambient temperature interval T_a (deg C)	HEAT OUTPUT (kWh) From heat pump Q_c	From supple-mentary heater Q_b	ELECTRICITY CONSUMPTION (kWh) Stand-by	Heating	Defrosting	Heat pump TOTAL W	Supple-mentary heater W_b	No of heating cycle starts	No of defrost cycle starts	Time in heating mode (hours)	Time in defrost mode (hours)
-5 to -4.1	352	284	5	156	32	193	297	90	61	74	16
-4 to -3.1	82	29	1	38	4	43	31	18	8	18	2
-3 to -2.1	-	-	-	-	-	-	-	-	-	-	-
-2 to -1.1	251	90	1	109	28	138	91	43	51	51	13
-1 to -0.1	148	45	2	64	19	85	44	34	25	30	9
0 to 0.9	312	28	2	130	32	164	19	53	34	59	16
1 to 1.9	297	49	4	114	41	159	44	74	60	51	20
2 to 2.9	222	15	3	83	17	103	12	60	20	37	9
3 to 3.9	601	34	10	220	20	250	26	202	25	98	11
4 to 4.9	468	22	8	172	5	185	16	154	5	73	3
5 to 5.9	554	16	11	192	13	216	10	189	16	84	7
6 to 6.9	555	6	11	190	0	201	1	190	0	80	0
7 to 7.9	546	3	12	184	1	197	1	163	3	76	1
8 to 8.9	565	-1	15	182	0	197	1	180	0	76	0
9 to 9.9	215	-1	7	65	0	72	0	76	0	28	0
10 to 10.9	264	-1	15	80	0	95	0	94	9	34	5
11 to 11.9	331	-1	21	99	0	120	0	120	4	42	2
12 to 12.9	93	1	17	26	0	43	2	34	1	11	0
13 to 13.9	108	-1	17	30	0	47	0	38	0	13	0
14 to 14.9	27	0	7	8	0	15	0	9	0	3	0
15 to 15.9	6	0	1	2	0	3	0	2	0	1	0
16 to 16.9	6	0	1	2	0	3	0	2	0	1	0
-5 to 16.9	6011	618	173	2145	211	2529	595	1825	308	941	107
Proportion of electricity consumed at off-peak rate			27.7%	33.9%	24.0%	33.9%	56.3%				

TABLE 7.3: a) RESULTS FROM PART 1 ANALYSED BY DAILY MEAN AMBIENT TEMPERATURE

Daily mean ambient temperature interval T_a (deg C)	HEAT OUTPUT (kWh) From heat pump Q_c	HEAT OUTPUT (kWh) From supplementary heater Q_b	ELECTRICITY CONSUMPTION (kWh) Stand-by	ELECTRICITY CONSUMPTION (kWh) Heating	ELECTRICITY CONSUMPTION (kWh) Defrosting	Heat pump TOTAL W	Supplementary heater W_b	No of heating cycle starts	No of defrost cycle starts	Time in heating mode (hours)	Time in defrost mode (hours)
-8 to -7.1	86	33	0	41	4	45	39	12	20	20	3
-7 to -6.1	0	0	0	0	0	0	0	0	0	0	0
-6 to -5.1	286	48	1	125	16	142	59	17	50	59	10
-5 to -4.1	198	31	1	82	8	91	39	9	17	38	4
-4 to -3.1	398	36	0	168	15	183	54	41	52	78	9
-3 to -2.1	1028	97	1	424	49	474	138	90	133	194	28
-2 to -1.1	935	47	2	384	44	430	77	106	130	172	24
-1 to -0.1	1185	89	6	475	61	542	138	137	198	214	34
0 to 0.9	528	1	0	208	34	242	19	58	85	91	18
1 to 1.9	1369	3	3	529	66	598	56	189	212	228	36
2 to 2.9	1431	-52	6	527	48	581	3	301	221	228	28
3 to 3.9	2378	-93	13	841	36	890	2	556	245	358	27
4 to 4.9	1015	-36	14	339	10	363	2	270	44	147	5
5 to 5.9	1627	-65	14	540	15	569	2	432	81	230	9
6 to 6.9	787	-31	10	250	3	263	0	245	25	107	2
7 to 7.9	1023	-42	14	316	1	331	0	331	13	133	1
8 to 8.9	902	-38	16	268	0	284	1	315	17	115	2
9 to 9.9	755	-31	16	222	3	241	1	259	9	96	2
10 to 10.9	658	-30	17	189	3	209	2	225	10	80	2
11 to 11.9	369	-17	10	104	0	114	2	124	1	44	0
12 to 12.9	281	-14	13	78	1	92	1	101	1	33	1
13 to 13.9	191	-8	13	53	0	66	0	67	1	22	0
14 to 14.9	47	-3	8	13	0	21	0	15	0	5	0
15 to 15.9	2	0	7	0	0	7	0	2	2	0	1
16 to 16.9	3	0	4	1	0	5	0	1	0	0	0
17 to 17.9	3	-1	5	0	0	5	0	2	0	0	0
18 to 18.9	0	0	1	0	0	1	0	0	0	0	0
19 to 19.9	0	0	1	0	0	1	0	0	0	0	0
20 to 20.9	5	0	2	0	0	2	0	2	6	1	0
-8 to 20.9	17490	-76	198	6177	417	6792	635	3907	1573	2694	246
Proportion of electricity consumed at off-peak rate			26.3%	33.1%	34.8%	33.0%	57.2%				

TABLE 7.3: b) RESULTS FROM PART 2 ANALYSED BY DAILY MEAN AMBIENT TEMPERATURE

DEGREE DAYS	1984/85	1985/86	Average over 20 years to 1979
Sept	79	64	94
Oct	149	162	171
Nov	239	352	186
Dec	340	298	360
Jan	467	386	379
Feb	406	482	343
March	344	355	320
April	227	296	238
May	166	142*	156
Total	2417	2537	2247

* Provisional figure

Source: Department of Energy and BSIRA, published in reference 100

TABLE 7.4: DEGREE DAYS FOR THE MIDLAND REGION FOR THE TWO TRIAL SEASONS AND THE 20 YEAR AVERAGES.

	Actual results for part 2	Predictions from simulation program DSI			
		A	B	C	D
INPUT DATA					
House heat demand at -1°C (kW)	5.2	7.3	5.2	5.2	5.2
radiator heat output at 55°C (kW) (flow water temp.)	9.3	10.7	10.7	9.3	9.3
ambient temperature distribution across the heating season	1985/86 heating season	typical[1] heating season	typical data	typical data	1985/86 data
RESULTS					
Heat output from heat pump, Q_c (kWh) (ignoring pipework heat losses)	17490	23248	17677	17652	17717
Heat output from supplementary heater[2] Q_b (kWh)	656.7	1492	285	328	887
Heat output total $Q_c + Q_b$ (kWh)	18146.7	24740	17962	17980	18604
Electricity consumed by compressor and fan in heating mode only W_h (kWh)	6177	7623	5631	5966	6168
cycle $COP_h = \frac{Q_c}{W_h}$	2.83	3.05	3.14	2.96	2.87

NOTES: 1. The average of 12 years data for the Midlands region
2. In order to compare the actual results with the predictions, the heat output of the supplementary heater is assumed to be equal to the electricity consumed by the heater

TABLE 7.5: COMPARISON BETWEEN THE RESULTS FOR PART 2 AND THE PREDICTIONS FROM PROGRAM DSI

Test No.	Microphone Location	Test Condition	dB(A)
16	1m from unit at mid height	Ambient Noise	49.6
36 - 40	1m from unit at mid height and above	Fan + Compressor	63.5
24 - 28	as above	Fan + Compressor with sand bags around the base of the unit	61.3
34	Outside of house near window	Ambient Noise	48
35	" " " " "	Fan + Compressor	53.6
22	" " " " "	Fan + Compressor sand bags around the base of the unit	50
8	Centre of downstairs room	Ambient Noise	29.6
9	" " " "	Fan only	30.1
41	" " " "	Fan + Compressor	31.4
10	" " " "	Fan + Compressor sand bags around the base of the unit	28.2

All measurements were taken during daylight hours.

TABLE 7.6: SUMMARY OF SOUND PRESSURE LEVEL MEASUREMENTS OF THE HEAT PUMP AT THE FIELD TRIAL SITE.

CHAPTER 8 DISCUSSION OF THE COMMERCIAL PROSPECTS

This chapter discusses the commercial prospects for the domestic heat pump design developed in this project. It is assumed that the motivation of a potential customer in purchasing the heat pump is to realize a financial saving as a result of replacing a high running cost heating plant by a more economical system. In order to achieve this the customer must apportion some of his capital resources to purchase and install the heat pump. Calculating the time taken for the initial cost to be repaid by the savings in running cost, the simple payback period, is a graphic and straight-forward method of assessing the wisdom of the investment. The payback period method based on present fuel prices is applied in the following analysis, in favour of more sophisticated financial appraisal techniques, because it is the most likely method to be used in practice. To perform this calculation the data required are the capital outlay and the expected annual monetary savings, both of which are discussed below.

Table 8.1 illustrates how the final installed price to the customer is likely to be structured. The total component cost for a production rate of 500 units per year is estimated to be £818, based on the experience of assembling the prototype unit and test rigs. The labour charges and gross margin figures were obtained from discussions with the GEC production companies involved with this project. The ex-works selling price is thus £1947, on to which a 40 per cent mark up is assumed to be added by the installer/distributor giving a final installed price of £2726. This price does not include the cost of a major increase in the radiator area, which may add up to a further £500. Thus, depending on the adequacy of the original radiator network, the customer must invest in the region of £2700 to £3200 in the installation of the domestic heat pump.

The purchase scenario supposes that the customer owns an oil fired central heating boiler which is approaching the end of its useful life and does not have access to the natural gas supply. The customer must decide whether to simply replace the boiler with a new unit or to substitute the electric heat pump. Thus the installed cost of the heat pump is partially offset by the cost of a replacement oil boiler, which is of the order of £700 installed. If no enhancement to the radiator network is needed, the effective capital cost of the heat pump is then about £2000.

The running costs of whole house space heating systems are summarized in table 8.2 for the principal fuels used in the UK domestic market. The price data refers to September 1985. The right hand column contains the annual running costs of each fuel for a heating requirement of 18000 kWh/a which is typical of a three bedroom semi-detached or a well insulated four bedroom detached house with a heating regime of 16 hours per day.

The assumed appliance seasonal efficiencies for gas, oil and solid fuel fired boilers represent a consensus of reported results (104, 105, 106, 107, 108). The 'efficiency' of electric unit storage heaters is based on tests carried out at the ERC and reflects the inability to properly control the heat emission from such heaters (109).

As expected, the greatest fuel cost savings occur when the heat pump replaces an oil fired boiler, amounting to £207 per year. Hence the simple pay back period for this investment is £2000/£207, or approximately 10 years. Under the less favourable circumstances in which significant enhancement of the radiator area is required, the payback period could increase to £2500/£207, i.e. 12 years. The judgement within GEC was that these payback periods would be regarded as unattractive by potential customers.

In the comparison between the heat pump and the oil-fired boiler it should be remembered that the heat pump provides only space heating, whereas the boiler would normally be also used to produce domestic hot water. It is assumed that a heat pump user would install a 50 gallon hot water cylinder heated by off-peak electricity via an immersion heater. This results in a hot water production cost slightly less than that achieved by the oil boiler.

Interestingly, it is possible to reduce the payback period of the heat pump by introducing design changes which lower the COP_h, provided that the ensuing decrease in installed cost exceeds the reduction in fuel cost savings.

An example of this is shown in table 8.3, in which the sizes, and therefore the costs, of the condenser and evaporator are gradually reduced to illustrate the effect on the payback period. The selling price quoted is the estimated final installed price to the customer, not including any significant enhancement to the radiator system. It is calculated using the price structure described above. It is assumed that reductions in the evaporator size also lead to reductions in the cost of the casing as well as the material involved in the heat exchanger, as indicated in the table. The change in the COP_h of the reduced versions is calculated from program VCCM and expressed as a proportion of the measured seasonal COP_h of the prototype unit. For each size of evaporator, the air side face velocity was chosen to optimize the COP_h, as explained in Chapter 4. The payback period shows the greater dependency on the evaporator size, because of the higher material cost of this heat exchanger. Indeed, the payback period exhibits a reduction for decreases in the condenser size only for the small evaporator sizes. If the goal is to minimise the payback period, the optimum design would appear to be an evaporator of approximately 10 m in length and a condenser of 8 m. The COP_h of this configuration is 92 per cent of that achieved by the prototype design, but the payback period has fallen from 9.8 to 8.6 years. It should be made clear that a 10 m evaporator is outside the range of experience of this present study, as the smallest experimental evaporator was 20 m, so this financial optimisation analysis should not be interpreted as a demonstration of the engineering validity of the smaller evaporator. Indeed the adoption of a smaller coil may lead to design compromises which are ultimately of greater concern to the customer than the payback period, for example greater noise levels due to the higher face velocity requirements, and a higher frosting rate resulting from the lower coil surface temperatures.

To achieve a more reasonable payback period of, say, five years requires one of the following changes:

a) A reduction in the installed cost of the heat pump to about £1700, for the same fuel prices and equipment performance which pertained to the field trial. This represents a 38 per cent reduction in the installed cost of the heat pump. It should be emphasised that the component cost data presented in table 8.1 were derived from the experience of manufacturing one-off experimental systems and the prototype. No detailed commercial negotiations with the suppliers have taken place, nor has a value analysis of the design been conducted, both of which could yield significant cost savings. However, it is difficult to imagine the 38 per cent reduction being achieved without some reductions in the manufacturing and installation margins, and the indications are that significant reductions in these margins are unlikely.

b) An increase in the seasonal COP_h to 5.80, which is, to say the least, unlikely.

c) An increase in the price of heating oil of 38 per cent for the same electricity price. The changes in prices of the domestic heating fuels from 1974 to 1985 are shown in figure 8.1 (110). Historically the price of heating oil has at times suddenly and dramatically risen, but the price increase is then gradually eroded by corresponding adjustments in the price of the other fuels, particularly electricity. It is interesting to calculate how the simple payback period for the heat pump would have varied in recent times, and this is shown in table 8.4. The fuel prices are based on the 1985 values shown in table 8.2, and are adjusted by the appropriate indices in order to calculate their average levels for each year in the period 1974 to 1985. The difference between the installed cost of the heat pump and the oil boiler are based on the cost estimates for 1985 and adjusted by the general index of retail prices (110). The payback period for each year is calculated from the price difference and saving in running costs based on the prices applying to each particular year. A wide variation in the payback period is evident in the results, although a trend of a gradually reduced payback period is apparent from 1979 onwards. The minimum occurs in 1985, and while the official statistics for 1986 are not yet available, it is clear that the significant decrease in the crude oil price of early 1986 has lead to a reducion in the cost of heating oil and thus to an increase in the payback period.

Therefore there is little prospect of 38 per cent relative increase in the price of oil in the immediate future. In the longer term, however, there remains the expectation of substantial rises in the oil price as this is a finite resource.

The decision taken by GEC on the future of the domestic heat pump was to delay the product launch until the market prospects are regarded as more favourable.

Two final observations to conclude this discussion. The first is a reflection on the difficulty facing the potential customer in coming to a rational decision on the purchase of the heat pump, in view of the uncertainty in future fuel prices. The case presented above is based on the assumption that the present fuel prices remain constant throughout the lifetime of the product. This approach can lead to errors. For example, the annual savings and differential capital cost historical data presented in table 8.4 illustrate that had a customer purchased the heat pump in 1976 for £840, on the expectation of a 78 year payback time, he would have found by 1985 that the accumulated annual savings in fuel bills were £903.6, which, in simple terms, would have paid for the investment. This would have occurred because of the relative increase in the price of heating oil. Thus the customer must exercise his own judgement about the future price of heating oil, and this may tip the balance in favour of the heat pump.

The second observation is the success of the GEC Nightstor 100 product, described in Chapter 1, which has been well received in the market place and is selling well. This product has roughly the same capital cost and payback period as the heat pump. The local area Electricity Boards are responsible for the marketing and installation of the Nightstor, and the Boards are enthusiastic about its prospects. This indicates that the payback period does not necessarily dominate the customer's purchasing decision, and that well targeted marketing can greatly assist in the sales success of a high capital cost domestic product.

Components Cost Estimate for 500 systems produced per year

	£	
a) Major Items		
Compressor	100	
Condenser	100	
Evaporator	250	
Fan	60	
Flow boiler	40	
Casing (including assembly)	80	estimate
Control System	100	
Sub total	£730	
b) Minor Items		
Expansion Valve	10	
Receiver	10	
Accumulator	10	
Filter/Dryer	3	
Solenoid Valves	25	
Misc. (pipe, fittings, refrigerant)	30	
Sub total	£88	
Total component cost	£818	

Determination of the Ex-works Price

Direct labour cost	100
Overhead recovery on direct labour (250%)	250
Labour plus material cost	£1168
Gross margin of 40% on ex-works selling price	779
Ex-works selling price	£1947

Price to Customer

Assume installer applies a margin of 40% on the ex-works selling price	779
Price to customer	£2726

TABLE 8.1: ESTIMATION OF THE INSTALLED PRICE OF THE HEAT PUMP

Fuel	Price per Unit	Heat Content per Unit	Appliance Seasonal Efficiency	Cost per Useful kWh (p/kWh)	Annual Cost for 18000 kWh
Electric heat pump (35% night rate)	4.38p	1kWh/kWh	2.34	1.87	£337
Natural Gas fired boiler	37p/therm	29.3kWh/therm	0.70	1.80	£324
Solid fuel fired boiler (Anthracite grains)	660p/50kg	452.3kWh/50kg	0.65	2.24	£403
Electric unit storage heaters (100% night rate)	2.04p/kWh	1kWh/kWh	0.80	2.55	£458
Oil fired boiler (28 sec)	20p/litre	10.18kWh/litre	0.65	3.02	£544

TABLE 8.2: FUEL COST COMPARISON FOR SEPTEMBER 1985 PRICES.

length of evaporator	cost of evaporator	cost of casing	length of condenser (m) cost of condenser	2 £20	4 £40	6 £60	8 £80	10 £100
5	£42	£55	a) Heat pump selling price	£1948	£1995	£2042	£2088	£2135
			b) % of prototype COP_h[1]	73.1	81.1	83.7	84.9	85.7
			c) Annual fuel cost saving[2]	£83	£128	£141	£147	£151
			d) Simple payback period[3] (years)	15.0	10.1	9.5	9.4	9.5
10	£83	£60	a)	£2102	£2149	£2196	£2242	£2301
			b)	75.8	86.5	90.6	92.4	93.5
			c)	£99	£154	£172	£179	£187
			d)	14.2	9.4	8.7	8.6	8.7
20	£167	£70	a)	£2322	£2368	£2415	£2462	£2508
			b)	76.8	89.5	94.4	97.0	98.3
			c)	£105	£167	£187	£197	£201
			d)	15.4	10.0	9.2	8.9	9.0
30	£250	£80	a)	£2539	£2585	£2632	£2679	£2726
			b)	77.1	9.04	95.8	98.4	100
			c)	£107	£171	£192	£202	£207
			d)	17.2	11.0	10.1	9.8	9.8

NOTES: 1 Predicted COP_h from program VCCM expressed as a percentage of the value measured for part 2 of the prototype field trial.

2 For an annual heat demand of 18000kWh. The saving is compared with the cost of running an oil fired boiler to produce this quality of heat, estimated to be £544.

3 Assuming a replacement oil fired boiler would cost £700 to install.

TABLE 8.3: SENSITIVITY ANALYSIS OF PAYBACK PERIOD TO CHANGES IN THE SIZES OF THE CONDENSER AND EVAPORATOR

Year	Cost per useful kWh using an electric heat pump[1] (p/kWh)	Cost per useful kWh using an oil fired boiler[2] (p/kWh)	Annual fuel cost saving for 18,000 kWh/a	Difference in installed cost between the heat pump and the oil boiler[3]	Simple payback period (years)
1974	0.42	0.56	£25.2	£580	23.0
1975	0.62	0.64	£3.6	£721	200.3
1976	0.76	0.82	£10.8	£840	77.8
1977	0.87	1.02	£27.0	£974	36.1
1978	0.96	1.03	£12.6	£1054	83.7
1979	1.05	1.30	£45.0	£1196	26.6
1980	1.33	1.74	£73.8	£1411	19.1
1981	1.60	2.07	£84.6	£1578	18.7
1982	1.75	2.36	£109.8	£1714	15.6
1983	1.82	2.65	£149.4	£1792	12.0
1984	1.84	2.68	£151.2	£1882	12.4
April 1985	1.87	3.02	£207.0	£2026	9.8

NOTES: 1 Heat pump COP_h is 2.34, 35% of electricity used is off peak

2 Seasonal efficiency is 0.65

3 Prices adjusted using the general retail price index

TABLE 8.4: VARIATION IN THE PROJECTED PAYBACK PERIOD FOR 1974 to 1985

CHAPTER 9 CONCLUSIONS

1. The primary objective of this project was to develop and assess an air to water heat pump system of 6 kW nominal capacity for the UK domestic space heating market. This has been achieved, and the integrity of the design has been demonstrated by a field trial of a prototype heat pump. The particular design features of the heat pump include:

   a) Attractive, compact unit for outdoor siting. A 6 kW electrical supplementary heater is included in the unit.

   b) Ease of installation as it is compatible with conventional radiator systems and operates on single phase electricity. It is, however, recognised that in many installations some enhancement of the radiator area is required to attain an acceptable COP_h.

   c) Low noise levels. Following the modifications to the compressor mounting arrangements, the heat pump was not audible from inside the nearby houses.

   d) Low starting current requirement by virtue of the on-board soft start device, so there is no noticeable disturbance to the local supply voltage.

   e) A COP_h of 2.34 was achieved for the entire 1985/86 heating season, which compares favourably with competitive domestic heat pumps.

2. The major components of the vapour compression circuit, the compressor, condenser, expansion valve and the evaporator, were selected using the two approaches of detailed measurement of the performance of the individual components, and analytical modelling of component behaviour as a part of the complete heat pump system. The performance studies required the design, construction and operation of three major experimental facilities, the Component Test Rig, the Evaporator Test Rig and the Environmental Chamber.

3. The evaporator was designed from first principles. It differs from the conventional tube-in-plate type heat exchanger, used in air conditioning and refrigeration equipment, in that it is formed from a wire wound heat transfer tubing. The experiments carried out on two prototype evaporators of similar geometry but different surface area demonstrated that the design performed well under all conditions. The influence of increasing evaporator size was found to increase the heat output of the heat pump rather than the COP_h. The most significant effect of greater area was found to be the resultant ability to operate at lower fan speeds and therefore lower noise levels.

   The evaporator demonstrated an ability to operate satisfactorily under frosting conditions for extended periods. A systematic experimental examination of the factors which influence the frost accumulation rate, including air temperature, relative humidity, condenser water temperature, and fan characteristic, resulted in a simple equation to describe the behaviour of the final evaporator design. Frost fouling effects were found to be most significant at an air temperature of 0°C, high relative humidity, coupled with a low condensing temperature.

4. Detailed measurements were made on an hermetic compressor and two coaxial tube condensers. An assessment of the thermodynamic irreversibilities associated with the major system components has shown that the greatest losses occur in the compressor. Improvements in the isentropic efficiency of the compressor would lead to significant increases in the COP_h of the heat pump. The condenser was selected on the consideration of the thermal performance and the water side pressure drop. A low pressure drop is desired to minimise the installation effort, to facilitate the attainment of a high water flow rate which improves the COP_h of the system, and to provide an allowance against subsequent water side fouling.

5. The methods and results of a mathematical representation of the vapour compression cycle are presented. Given the component design details and the external conditions, i.e. the ambient air

temperature and the condenser water inlet temperature, the model determines the refrigerant conditions and the heat and work transfer rates for the system. The model is encoded as an iterative computer program, VCCM, which is written in the style of a design tool. Thus, the performance consequences of changes to the design can be readily predicted, as can the behaviour of a given design under a range of operating conditions. The model was used to vindicate the selection of R22 as the working fluid, to indicate the consequences of changes in the performance of the compressor, and to explore the effects of changes in the size and operation of the two heat exchangers.

The analytical methods could find application in the design of related, vapour compression cycle products, for example refrigeration and air conditioning systems.

6. A method to predict the seasonal performance of a heat pump in actual use has been developed. This includes a simplified equation set to describe the heat pump heat output and electrical input characteristics, and involves a transient analysis of a two element, lumped parameter model. The two elements are the water filled radiator system and the building. This model is contained in an interactive computer program, DSI. A sensitivity analysis revealed that the heat dissipation rate of the radiator network is a critical parameter; it is recommended that the installed radiator network has an output rate at a flow temperature of 55°C at least equal to and preferably greater than the heat demand of the building at -1°C.

7. A prototype heat pump was manufactured and installed in a house central heating system. The heat pump and the installation were highly instrumented; refrigerant, water, ambient and room air temperatures, water flow rate, heat transfer rates and power consumption were recorded at roughly 5 minute intervals. The field trial extended over 1½ heating seasons, and data were collected for a total of 441 days. Analysis of the results revealed the gross energy flows, the heat pump output and COP_h characteristics, the starting frequency and run time of the heat pump, and the operating nature of the heat distribution system and the building.

8. The field trial highlighted a weakness in the hot gas defrost system which was resolved by reducing the pressure restriction produced by the evaporator distributor tubing. Compressor noise levels proved to be a nuisance during night-time operation, and this was solved by modifying the compressor mounting arrangements and by installing vibration isolators in the water piping.

9. The field trial was also used to develop and evaluate the heat pump control system. It was demonstrated that the device can be controlled by a room thermostat and a water thermostat to limit the condenser outlet temperature. The defrost initiation system performed well, based on detecting an abnormally high difference between the air and refrigerant temperatures across the evaporator. The simple timer based control over the supplementary heater also performed well, and the provision of user overrides to completely inhibit supplementary heating or to enable it only during off-peak periods was found to be a useful and economical feature.

10. The payback period against an oil-fired boiler is estimated to be 10 years, based on September 1985 fuel prices, the achieved COP_h and an estimated installed price. This payback period is regarded by the Company as being unattractive from the customer's point of view and so plans to launch the product are being deferred.

11. The main achievements of the work are the component selection and design methodology of the heat pump itself, the development of the control strategy, the vapour compression cycle design tool and the in-service performance prediction program. The field trial of the prototype unit was an in-depth case study of the behaviour and performance of a domestic heat pump. This study involved the development of sophisticated data monitoring and analysis techniques which may find application elsewhere. The detailed results obtained from the trial form a comprehensive data base of the transient behaviour of the system which may be used to refine further the in-service performance prediction methods. Finally, useful experimental facilities have been developed in the project for component and system performance measurements, namely the Component Test Rig, the Evaporator Test Rig and the Environmental Chamber.

CHAPTER 10 FURTHER WORK

1. COMMERCIAL REALISATION OF THE DOMESTIC HEAT PUMP

The following topics need to be addressed to transform the prototype heat pump into a saleable product:

a) Development of an electronic version of the present micro-computer based control system.

b) Industrial design of the heat pump, giving attention to ease of manufacture and installation, long term durability of the casing, and product appearance.

c) Value analysis of the present design and negotiations with the component suppliers to reduce the bought in material costs.

2. IMPROVEMENT IN THE OVERALL ISENTROPIC EFFICIENCY OF AN HERMETIC COMPRESSOR

There is considerable scope for improvement in the overall isentropic efficiency of hermetic compressors from the present level of about 50 per cent. Such an improvement would increase the COP_h and hence the attractiveness of the heat pump, and would also lead to reduced running costs in other compressor applications, e.g. air conditioning systems. Attention should be paid to the following areas:

a) Motor efficiency.

b) The influence of compressor rotational speed.

c) Motor cooling and the feasibility of recovering the waste heat from the motor directly by the heat sink fluid rather than via the refrigerant vapour.

d) Pressure losses in the refrigerant flow path through the compressor.

e) Valve design.

APPENDIX A FIELD TRIAL PERFORMANCE MONITORING SCHEME

The field trial data logging system was centred on Hewlett Packard's HP3421A data acquisition and control system and the HP86B micro-computer from the same company. The details of the computer hardware are shown in table A.1, and its operation is discussed in section 2.

1 TRANSDUCER SELECTION, CALIBRATION AND OPERATION

1.1 Thermometry

A total of eleven thermometers were used, all of which were platinum resistance thermometers with a resistance of 100Ω at 0°C and a fundamental interval of 38.5Ω . The thermometers were manufactured by Nulectrohms Ltd to BS 1904 : 1984, tolerance class A, which permits a tolerance of $\pm$ (0.15 + 0.002 $|T|$) deg C, where $|T|$ is the modulus of temperature in degrees Celcius (64). To eliminate lead resistance errors the thermometers were connected as a 4 wire circuit, as shown in figure A.1. For each thermometer one pair of leads carried the energizing current and the other provided the voltage sense across the element. The current for all the PRT's was supplied by a constant current source, model 1021 manufactured by Time Electronics. The source was set to deliver a current of 1 m A which is sufficiently small to obviate self-heating errors and developed an acceptably high voltage drop of about 0.1 V across each thermometer. As there was the possibility of long term drift in the value of the reference current, it was prudent to employ a means of directly measuring the current. This was done by connecting a 4-wire, low temperature coefficient, high precision reference resistor into the current loop and measuring the voltage drop across it at about the same time that the thermometers were scanned. Then the resistance of each thermometer element may be simply calculated from the voltage readings as:

$$R_n = \frac{V_n}{V_{ref}} \times R_{ref}$$

where R_{ref} is the known resistance of the reference resistor. For the temperature range appropriate to the field trial the temperature/resistance relationship is (64):

$$R_T = R_o\ (1 + AT + BT^2)$$

where R_T = resistance at temperature T

R_o = resistance at 0°C = 100Ω

T = element temperature in °C

A = 3.90802×10^{-3} °C^{-1}

B = -5.802×10^{-7} °C^{-2}

Thus the element temperature may be obtained by direct solution of this quadratic equation.

1.1.1 Uncertainties in the Absolute Temperature Measurement

The following reasoning determines the highest possible uncertainties in any temperature measurement at any time in the field trial. There are four sources of error to consider:

a) The PRT tolerance, which is ±0.15 deg C at 0°C.

b) The absolute uncertainty in the value of the resistance of the reference resistor (R_{ref}) at constant temperature. This is ± 0.019Ω.

c) The uncertainty in R_{ref} due to temperature variation. The measured temperature coefficient of the resistor is 8.9×10^{-4} Ω/deg C. Assuming a maximum possible temperature change of 20 deg C at the measurement station, this gives a change in resistance of 0.0178Ω. Combining this with the absolute uncertainty as shown above, gives a total uncertainty in R_{ref} ± 0.03 per cent.

d) The errors in the measurement of the two voltages V_n and V_{ref}. The one year (worst case) specifications for the HP3421A instrument give an uncertainty of ± 0.0254 per cent in each of the two voltage measurements (67). In practice, as the two voltage measurements were carried out within 15 seconds of each other it is unlikely that the reading errors specified in the one year figures would apply. However, for the worst case assessment, assume the above

Thus the combined reference resistance and voltage measurement errors become $\pm$ 0.03% $\pm$ 2 x (0.0204%) = $\pm$ 0.081% which corresponds to an uncertainty in R_n of $\pm$ 0.081Ω or $\pm$ 0.21 deg C. When this reading error is combined with the PRT tolerance, the worst case estimation of the uncertainty in the temperature measurement is $\pm$ 0.36 deg C.

1.1.2 Evaporator Air Intake and Discharge Thermometers, Measurements T_{a1}, T_{a2}

These were single element averaging bulbs designed to fit horizontally across the air intake and discharge grilles of one face of the heat pump. The stainless steel outer casings were not shielded against solar radiation effects and so some additional error may have resulted.

1.1.3 Room Air Thermometer, Measurement T_{ar}

This was a thin film type PRT enclosed in a plastic ventilated housing. It was located in the hallway of the house, in a suitable position to also act as the room thermostat for the purposes of control. It was attached to an outside wall of the hallway.

1.1.4 Ambient Air Thermometer, Measurement T_a

This was a robust, stainless steel sheathed element, specially designed for outdoor location. The manufacturer's model number was A1200. It is located in a Stevenson's Pattern radiation shield, positioned on a lawn about 5 m from the heat pump.

1.1.5 Heat Sink Thermometers, Measurements T_{w1}, T_{w2}, T_{w3}

Three water thermometers were required to record the water return temperature from the radiator network (T_{w1}), the condenser outlet temperature (T_{w2}), and the flow boiler outlet temperature (T_{w3}), which was also the flow temperature to the radiators. In conjunction with the measurement of the water flow rate, these temperatures were used to calculate the rate of heat output from the heat pump and the supplementary heater, as follows:

$$\text{condenser} \quad \dot{q}_c = \rho_w \dot{V}_w Cp_w (T_{w2} - T_{w1})$$
$$\text{flow boiler} \quad \dot{q}_b = \rho_w \dot{V}_w Cp_w (T_{w3} - T_{w2})$$

Since the temperature rises $(T_{w2} - T_{w1})$ and $(T_{w3} - T_{w2})$ were small, typically 6 deg C, a rigorous calibration procedure was required to reduce the uncertainty in the heat transfer rate calculations. The platinum elements were initially matched by the manufacturer to reduce the isothermal variation in resistance between the three thermometers. Then a calibration exercise to measure the residual resistance variation was carried out. All three thermometers were immersed in a well stirred water bath, and for each of five bath temperatures in the range 25 to 85°C the differences in coil resistances of each pair were measured. The resistance variations were converted into equivalent temperature errors and look up table of the temperature offsets across the measurement range was assembled for each thermometer pair. The tables were incorporated in the Data Acquisition and Control program to enable the offset corrections to be made on line, e.g.

$$T_{w21} = T_{w2} - T_{w1} - \delta\epsilon_1$$

where $\delta\epsilon_1$ is the offset correction for the pair T_{w2} and T_{w1} and was obtained by entering the look up table at the mean water temperature $(T_{w2} + T_{w1})/2$ and performing linear interpolation on the appropriate values. It is estimated that this technique reduces the uncertainty in the differential temperature measurements to about $\pm$ 0.05 deg C due to mis-matching. An additional uncertainty arose due to random count error in the voltage measurement system, which is given as 5×10^{-6} V in the HP3421A specifications. This corresponds to an uncertainty in each resistance of $\pm 0.005\Omega$, or in each temperature of $\pm$ 0.013 deg C. Thus the differential temperature measurement has a reading error of twice this amount, and so the total uncertainty in the differential measurement system is likely to be:

$\pm$ 0.05 deg C (mis-match) + $\pm$ 0.026 deg C (reading)

= $\pm$ 0.076 deg C,

which is 1.3 per cent of the typical temperature rise of 6 deg C.

Notice that, compared with the previous calculation of the uncertainty in the absolute measurement of temperature, considerations of the effects of uncertainty in the values of R_{ref} and V_{ref} do not apply in the case of the differential measurements as these errors cancel out when the difference is taken.

These thermometers were sheathed in stainless steel and immersed to a length of 100 mm in the water flow. T_{w1} and T_{w3} were located in tee fittings in the heat sink pipework near to the heat pump. T_{w2} was positioned in a tee fitting contained in the interconnecting pipework between the condenser and the supplementary heater.

1.1.6 Refrigerant Thermometers, Measurements T_{1-4}

The refrigerant temperature was measured at four positions; compressor suction (T_1), compressor discharge (T_2), liquid line (T_3) and evaporator inlet (T_4). The positions are illustrated in figure 6.1. The thermometers were of the same design as those used in the heat sink circuit, but they were not specially matched.

1.2 Heat Sink Flow Meter

A pelton wheel type flow meter was installed in the water flow pipe near the point of entry to the building. The transducer was manufactured by Litre Meter, model MM12.5. The pulsed output generated by the transducer was converted to a 0 to 20 mA output proportional to flow rate by a LMD 3DI signal processor and display unit made by the same company. The current signal was further converted to a voltage to be sensed by the HP3421A instrument using a 100Ω precision resistor.

The complete flow rate measurement system was gravimetrically calibrated to determine the constants of the linear characteristics:

$$\dot{V}_w = m_w V + C_w$$

where $\dot{V}_w$ = heat sink flow rate in m^3/h

V = signal voltage sensed by the logger

m_w, C_w = calibration constants

The accuracy of the flow rate measurement is estimated to be $\pm$ 1.5 per cent.

1.3 Power Consumption

The electrical power absorbed by the heat pump, including both the compressor and the fan, was detected by an electronic wattmeter. The transducer, Inwatt NES, measured the current and the supply voltage to the heat pump. The current was first transformed by an IMO ARF12 5 to 1 current transformer, and the secondary current was sensed by the wattmeter transducer. The voltage sense was simply the live to neutral voltage across the heat pump supply.

The transducer generated a 0-10 V dc output corresponding to a 0 - 6 kW power range, with a declared accuracy of $\pm$ 0.5 per cent of full scale. Since the actual power absorbed by the heat pump was of order 2.5 kW, then the representative uncertainty of this measurement is $\pm$ 1.2 per cent. The measuring system was checked by comparison with a high precision digital wattmeter with the result that the achieved accuracy appeared to be superior to that declared.

The voltage signal was converted to the power reading by the equation:

$$\dot{W} = m_p V$$

where m_p = 0.6 kW/V

1.4 Relative Humidity

The ambient relative humidity was measured by a capacitive type sensor, Lee Integer model CH12, located alongside the ambient thermometer inside the Stevenson's Pattern screen. The transducer was energised by a 12 V dc stabilized power supply and produced a 0 to 1 mA output proportional to relative humidity. A precision 100Ω resistor converted the current signal into a voltage which was sensed by the data logger. This system was calibrated with the aid of capsules of known relative humidity which were supplied by the manufacturer.

The relative humidity was determined from the measured voltage (V) by:

$$Rh = m_R V + C_R$$

where Rh = relative humidity in per cent

m_R, C_R = calibration constants

1.5 Electricity Consumption Meters

The electricity absorbed by the heat pump and the supplementary heater were recorded on two domestic type kWh meters manufactured by GEC Measurements Ltd. The electrical connections for the heat pump kWh meter are shown in figure 6.8. The accuracy class of both meters was 2, which for the typical currents used in the heat pump and flow boiler is interpreted as an uncertainty of $\pm$ 2 per cent (111). These meters were regularly manually logged.

1.6 Heat Meters

Two electromagnetic heat meters were employed to provide ample back up to the important measurement of the heat delivered by the heat pump. Both meters are manually logged.

a) ISS Clorius Combimeter

This instrument is made in 3 parts; a PRT installed in the flow pipeline, an integrating unit installed in the return line containing the second PRT, an electromagnetic flow meter and the signal processing circuitry to determine the thermal energy consumption, and a display unit which has two electromechanical counters indicating kWh of heat and hours of operation. The accuracy of this instrument is influenced by the magnitude of the difference between the flow and return temperatures. For differences between 2 and 10 deg C the instrument specification suggests a systematic under-reading of

$$\frac{0.2}{\Delta T} \times 100\%$$

where ΔT is the actual temperature difference.

Hence for this application an error of about -3.3 per cent is to be expected.

b) HG Type 100-6

This consists of two pocket installed PRT probes, an electromagnetic flow meter installed in the return line, a signal amplifier local to the flow unit and an integrator and display unit. The rolling counter display shows MWh to two decimal places, flow volume in m^3, and hours run. The manufacturer's specifications indicate a measurement accuracy of $\pm$ 4 per cent for the temperature differences appropriate to this application.

2. COMPUTER CONTROLLED MONITORING SYSTEM - METHOD OF OPERATION

The operation of the computer based monitoring system is controlled by the Data Acquisition and Control program (DAC); in this section the method of operation of this system is discussed by reference to the features of program DAC.

Program DAC is written in Hewlett Packard's Extended BASIC and calls on functions supplied by the Input/Output and the Advanced Programming enhancement Read-Only-Memories (112, 113, 114).

The method of operation is explained with the help of the flowchart shown in figure A.2. The initialization phase reads in the transducer calibration constants, thermal properties and other items of data, and instructs the operator to load empty data disks. Following the initialization phase the program executes a continuous cycle of instructions, shown as ABCA. Immediately following position A the computer sends instructions to the HP3421A data logger to scan the 15 dc voltage transducer signals and permits the logger to interrupt the computer once these readings have been taken. The time and date at which these instructions were sent are noted. The logger is set to maximum resolution (5½ digits), autorange and autozero configuration in order to operate at the highest possible accuracy, but this also reduces the scan rate to less than 2 readings per second. Hence while these readings are in progress the computer proceeds with task of processing the results from the previous measurement cycle. The voltage data is converted to temperatures, flow rate etc. according to the calibration equations, heat transfer rates are computed from the temperature differences and the water flow rate and the instantaneous COP_h is calculated. The program carries out a Trapezium Rule* integration over time of the heat pump power level, heat pump heat output rate and supplementary heat output rate in order to estimate the running totals of electricity consumed and heat delivered. The integration method is illustrated for the example of heat pump heat output:

$$Q_C = \int \dot{q}_C \, dt \simeq \sum_{n=1}^{\text{nitems}} \tfrac{1}{2} \, (\dot{q}_{C,n} + \dot{q}_{C,n-1}) \, (t_n - t_{n-1})$$

Following data processing, sub-routines associated with the control system are executed, Hardware Simulation and Controller, as discussed in Chapter 6. The screen is updated with the results from the data processing including the running totals, along with details of the current control settings and the status of the heat pump. The program waits at B for the interrupt signal from the HP3421A, upon which control moves to C. The voltage data from the logger is transferred to the computer under the supervision of a sub-routine which also checks for obvious data errors. Then sub-routine Actuator is executed which implements the control decisions taken on data from the previous measurement cycle. An AC line voltage measurement may be requested by the program; this reading takes a relatively long time to carry out and so is not taken every cycle.

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* In part 1 of the trial a simpler integration method was applied which assumed the measured variable remained constant across the time step.

The final operation is the storage of the data on disk. The time interval between raw data records is variable and depends on the activity of the heat pump, according to the following regime*:

Rapid changes, for example when the heat pump is starting or stopping, or initiating or terminating a defrost cycle. Four complete raw data records per minute are formed.

Heat pump active, meaning it is either producing heat or is in defrost mode. One raw data record is formed every five minutes.

Heat pump inactive, either in stand-by mode or off. Data is recorded at hourly intervals.

The data stored are the voltage readings as obtained by the logger, together with the time and date of the readings and two binary words which indicate the status of the heat pump (SYSTEM$) and of the control system (CTRL$). In addition to the raw data, the control settings are recorded on disk whenever a setting is revised by the operator. The program contains counters to assess when a disk is full, in which case subsequent data is stored on the disk in the second drive and a message to replace the full disk is displayed on the screen. In this way the system can operate continuously with manual intervention every four days or so to re-new the data disks.

Following data storage execution moves to point A, the start of the next cycle. The overall cycle time is approximately 15 seconds. Program DAC is also equipped with a number of routines to enable the system to recover from unexpected events. Examples include an Error Recovery sub-routine which intercepts and handles all computer generated errors,

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* This operating regime was developed during the first part of the trial, the above description defines the operation of the final version of program DAC which was used throughout part 2 of the trial. At the start of part 1 data was recorded at two minute intervals, with some additional records formed when the status of the heat pump changed, for example when it started or stopped.

and an interface timeout sub-routine which detects and acts upon an interface failure event.

3. DIFFICULTIES OF OPERATION

With a monitoring system as complex as that just described some difficulties of operation must be expected. Trouble was experienced with both heat meters and with the HP3421A data logger, as described below.

a) ISS Clorius Heat Meter

This heat meter failed completely soon after the field trial started. It was replaced by an identical unit supplied by the manufacturer which subsequently performed well, but the initial failure meant that for a period of about one week there was no independent check on the computer generated totals of heat output.

b) HG Heat Meter

This meter under-read the heat delivered by 10 per cent, when compared to both the computed totals and to the Clorius heat meter. Towards the end of the field trial the error increased to about 50 per cent. This fault was not rectified.

c) HP3421A

The logger scans the input channels using a reed relay type multiplexer. During the first part of the trial some difficulty was experienced with one of the reed relays, on the channel connected to the heat pump wattmeter transducer, which took the form of an intermittent contact. Once the nature of the problem was understood the wattmeter signal was connected to a previously unused channel, and subsequently operated normally, but by this time some corruption of the power level recording had occurred. This has consequences for the data processing scheme discussed in Appendix B. At the end of the first part of the trial, three other reed relays were showing signs of intermittency, affecting the measurements of T_2, T_3 and T_4.

Computer:	Hewlett Packard 86B 124 kbytes of Read/Write Random Access Memory Integrated HP-IB and monitor interfaces
Enhancement Read Only Memories:	
	1. Input/Output Revision A 2. Advanced Programming Revision B
Disc Memory:	Hewlett Packard 9121D dual disk drive 3½ inch single sided disks providing 270 kbytes storage per disk Connected by HP-IB
Data Acquisition and Control Unit:	
	Hewlett Packard 3421A 6 channels of actuation, 24 transducer channels multiplexed to a 5½ digit, digital volt meter Connected by HP-IB
Monitor:	Hewlett Packard 82913A, monochrome 12 inch cathode ray tube Connected by monitor interface

TABLE A1: DETAILS OF COMPUTER HARDWARE USED IN PERFORMANCE MONITORING

APPENDIX B FIELD TRIAL DATA ANLAYSIS METHODS

1. DATA REDUCTION STAGE 1 - FOUR HOURLY ANALYSIS

The main function of program FHA is shown in flowchart form in figure B1.

The preparation stage involves reading the transducer calibration constants and the determination of the period number from which to begin the analysis. This is done by reading a data file on the FHD disk which contains the current period number under consideration, this item is incremented each time the program moves to the next period number. The period number uniquely identifies every 4 hour period in the trial.

The program then prompts the user to load the relevant raw data disk and commences to extract all the data records for the period under analysis. In addition, whenever a raw data disk is first loaded, the program branches to point C, which determines the contents of the disk, retrieves all the control data files and proceeds to sort the control data into period order. When more than one set of control data apply to a single period, the control settings which prevailed for the greatest length of time within the period are taken as representative of the period as a whole.

Processing of the raw data has eight distinct stages, each performed by a sub-routine.

a) Data Conversion

The raw voltage data is converted to engineering quantities (temperature, flow rate, etc.) by the application of the transducer calibration equations.

b) Data Validation

As described in Appendix A some corruption of the raw data occurred as a result of the failure of the multiplexer in the data logger. This sub-routine carries out a simple check on each data item in the form of a comparison between pre-set low and high limits. If a data item fails this test, then the previous reading is substituted for it and a printed statement is issued specifying the variable name, timing of the invalid reading, the value of the invalid reading and the value of the substituted reading. Unfortunately the variable most affected by the multiplexer fault was $\dot{W}$, the instantaneous power consumption of the heat pump, which is a critical measurement. The procedure to check the validity of this measurement is thus more rigorous. Initially it is tested to see whether it falls within three pre-defined, narrow bands of acceptable values, corresponding to the heat pump activities of stand-by, fan only operation, and compressor operation for heating or defrosting duties. If this test is failed, before a substitution can be made, an examination of some of the contents of the raw data record is made to determine the true status of the heat pump. This includes assessing the product of water flow rate and condenser temperature rise to determine whether the heat pump is producing heat, and examining the evaporator refrigerant inlet temperature T_4 to investigate whether a hot gas defrost cycle is in progress. Having determined the mode of operation of the heat pump in this way, the value of $\dot{W}$ from the previous, valid, record is assessed for suitability. If its value is not consistent with the current mode of operation then a default value is substituted. As with any other variable, whenever a substitution is made a printed statement is produced, and these statements are then manually cross-referenced to the actual operating circumstances by the programmer.

c) Calculate Heat Transfer Rates

The instantaneous heat transfer rates $\dot{q}_c$ and $\dot{q}_b$ are calculated from the water flow rate and temperature rises in the same manner as program DAC, see Appendix A.

d) Data Categorisation

Each data record is examined and flags as set corresponding to the following categories:

Heat pump status - flag HP

- Standby
- Heating
- Hot gas defrosting
- Fan only defrosting

Supplementary heater status - flag FB

- Off
- On, 2 kW input
- On, 4 kW input
- On, 6 kW input

Electrical tariff period - flag TF

- Night rate
- Day rate

Heat pump heating and start up transient effect over - flag H

- Not heating, or within transient period
- Heating, start-up transient is over

In performing this categorization it is not sufficient to examine only the heat pump system status word SYSTEM$, because manual override actions are not represented by this variable. Hence it is necessary to examine the actual contents of the data record.

Heat pump status is defined by the value of heat pump power, $\dot{W}$; defrost status by a combination of $\dot{W}$ and T_4. The supplementary heater power is deduced from the measured heat output rate of the flow boiler, $\dot{q}_b$, by comparison with three narrow band lower and upper power limits. The electrical tariff period is obtained directly from the recorded time of day.

The final category defines whether a data record is relevant to the heat pump operating in heating mode at quasi-steady-state conditions. The heat pump undergoes a transient perturbation of all system temperatures during the initial operating phase immediately after start up, which lasts about 2 minutes. Beyond this phase, the flag H is set to 1 to identify the record as suitable for inclusion in the process of averaging over the periods of time when the heat pump delivers heat.

e) Event Register

This sub-routine enumerates the heating and defrost cycle starts throughout the 4 hour period, within the following categories

actual number of heating cycle starts
actual number of hot gas defrost cycle starts
actual number of fan only defrost cycle starts
number of computer controlled heating cycle starts
number of computer controlled defrost cycle starts

The distinction between actual and computer controlled actions arises from the possible use of the manual override control facilities. The numbers are obtained by reference to the values of the heat pump status flag HP, which were established by the previous sub-routine. The number of computer controlled actions is obtained by examination of the recorded control status byte, CTRL$ and the system status byte, SYSTEM$.

f) Minimum, Maximum

This sub-routine determines the minimum and maximum values of a number of variables for the four hour period. The variables include all the temperatures, the ambient relative humidity and the AC line voltage.

g) Data Integration

The purpose of this routine is to calculate the time integral across the period for all the variables. The integral is later used to obtain the time averaged value for some of the variables, or as a measure of the totalized quantity of heat delivered and energy consumed.

As the time step between each data record is variable the Trapezium Rule is applicable. Thus for variable V the integral is computed as

$$\int V dt = \sum_{n=1}^{nitems} \tfrac{1}{2} (V_{n+1} + V_n) \times (t_{n+1} - t_n)$$

where nitems is the total number of entries of V in the four hourly period under analysis.

The integration of the heat pump power consumption result receives special treatment. Firstly this quantity is categorised according to the 3 heat pump modes of operation (i.e. standby, heating etc.) and according to the electrical tariff period, so that in all six integrals are computed. This is to allow a detailed analysis into the pattern of heat pump energy consumption. Secondly the on/off control of the heat pump gives rise to a large step change in power consumption at the beginning and end of cycle which causes a difficulty for the integration technique. It was found that applying the Trapezium Rule across these discontinuities caused a systematic overestimation of the electrical consumption for the initial period of part 1 of the field trial.

The error arose because of the data sampling pattern of the logging and control program DAC. Prior to program version 4, implemented during week 3, data was recorded routinely every 2 minutes and immediately prior to any step change in operation. Version 4 caused an additional record to be created immediately following system start up, so that the step up in power consumption is accurately defined, but this version did not generate a record immediately following a system shut down. This deficiency was not corrected until version 7, implemented in week 10. Hence applying the Trapezium Rule to the data led to an overestimation of the electrical consumption for the data produced during weeks 5 to 10 as a consequence of the inadequate definition of the end of heating cycle.

This difficulty is handled by the program in the following manner, figure B2 refers. Whenever an end of cycle is detected the time interval between the record prior to shut down, A, and the first record following shut down, B, is examined. If $t_B - t_A$ is below a pre-defined maximum of 30s then no correction is introduced and the Trapezium Rule is applied as normal between A and B. If the time interval exceeds this limit then this indicates that high frequency sampling has not been applied during shut down and so an additional dummy data point is introduced, C. The power level for C is set to that at B, and C is positioned at 20s from A, which is the typical interval when high frequency sampling is in operation. The Trapezium Rule is then applied twice, between A and C and between C and B. This correction technique removes the error associated with the application of the Trapezium Rule between points A and B when they are widely separated in time. Furthermore, the slight overestimation attributable to 20s separation between A and the dummy point C is roughly equivalent in magnitude to the underestimation which occurs at the beginning of the heating cycle, hence the two errors tend to cancel.

This correction technique is applied only to the heat pump power consumption integration, and is relevant to about half of the first part of the field trial. Part 2 was not effected because of the improvements made to the data recording procedures in program DAC.

For all the measured temperatures except the ambient, T_a, and the room temperature, T_{ar}, an integral is calculated which represents the temperature conditions which apply when the heat pump is producing useful heat; this integral is later used to determine the heating mode averages. Those data records which do not apply to this mode of operation are excluded from the integration. In addition the first two minutes of operation in heating mode are excluded from the sample, since large transient changes occur during this initial stage, as discussed in d) above. The flag H is used to indicate that the heat pump is in heating mode and the start up transient is over. In addition to calculating the time integral of the specified variables for the remaining heating cycle, the time period over which the integral acts is also calculated, to enable the average values to be obtained.

Finally a set of temperatures is integrated across the periods when the heat pump is in hot gas defrost mode. They include the water return temperature T_{w1} and the four heat pump refrigerant temperatures T_{1-4}.

h) Data Averaging

The time averaged results are obtained by dividing the integrated totals by the appropriate time periods. The averages calculated include the four hourly period averages of all of the water and air temperatures, and of the ambient relative humidity and the AC line voltage, the heating mode averages of all the refrigerant, water, evaporator air intake and discharge temperatures, and the thermal defrost mode averages of the refrigerant and the water return temperatures.

The remaining parts of program FHA handle the storage of the two result records and the management of the data files. File FHD is a direct access data file of 300 records per disk with 700 bytes per record. The DEFROST file is similarly designed but has the reduced storage requirement of 100 bytes per record.

2. DATA REDUCTION STAGE 2 - DAILY ANALYSIS

The four hourly data base is further compressed by the application of program DAYTOTAL which determines the 24 hourly totals and averages of a selected number of variables. The result file is named DAYTOTALS and consists of 600 records per disk each containing 50 alpha numeric elements. The daily mean results are calculated as the arithmetic mean of the results from the 6 relevant four hourly periods. The heating mean results are weighted by the time in heating mode for each period. The daily totals are simply the summation of the values across the 6 periods. The control setting data are reduced to a smaller number of critical values, and the numerical values determined for the daily file represent the daily mean values.

3. DATA REDUCTION STAGE 3 - AMBIENT TEMPERATURE PROFILE

The final stage in the data reduction exercise is the organisation of the DAYTOTAL file into ambient temperature intervals. In this way the results for each day of the trial are categorized by the daily mean ambient temperature, and the totals of hours of operation, heat produced and electricity consumed are determined for the trial as a whole for each ambient temperature. This categorization process is carried out by program PROFILE.

4. PRINTED OUTPUT

A variety of programs have been written with the objective of producing hardcopy output of the contents of the major data files.

a) RDSS REPORT produces a tabulation output from the raw data file, and includes details of the control settings in force, see sample in figure B3.

b) DEFROST REPORT generates a tabulation output of the contents of the hot gas defrost file, see figure B4.

c) FHD REPORT summarizes the contents of the four hourly file, and arranges all 6 of the period reports to one page with a summary of the results for the day, see figure B5. In producing the 24 hour results this program is similar to DAYTOTALS, the main difference being FHD REPORT directs the results to the line printer instead of to the disk drive.

d) DAY REPORT summarizes the results of 7 days worth of data from the day totals file an includes the totals and averages for the week. It also calculates the number of degree days for the week from the minimum and maximum ambient temperatures to the usual base temperature of 15.5°C (115). Sample output is shown as figure B6.

Aston University

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