A NOVEL MECHANICAL VENTILATION HEAT RECOVERY/HEAT PUMP SYSTEM

by

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CONTENTS

ABSTRACT

The trend towards improving building airtightness to save energy has increased the incidence of poor indoor air quality and associated problems, such as condensation on windows, mould, rot and fungus on window frames. Mechanical ventilation/heat recovery systems, combined with heat pumps, offer a means of significantly improving indoor air quality, as well as providing energy efficient heating and cooling required in buildings.

This thesis is concerned with the development of a novel mechanical ventilation heat recovery/heat pump system for the domestic market. Several prototypes have been developed to provide mechanical ventilation with heat recovery. These systems utilise an annular array of revolving heat pipes which simultaneously transfer heat and impel air. The devices, therefore, act as fans as well as heat exchangers. The heat pipes have wire finned extended surfaces to enhance the heat transfer and fan effect. The systems use environmentally friendly refrigerants with no ozone depletion potential and very low global warming potential. A hybrid system was developed which incorporated a heat pump to provide winter heating and summer cooling.

Tests were carried out on different prototype designs. The type of finning, the working fluid charge and the number and geometry of heat pipes was varied. The prototypes provide up to 1000m^3/hr airflow, have a maximum static pressure of 220Pa and have heat exchanger efficiencies of up to 65%. At an operating supply rate of 200m^3/hr and static pressure 100Pa, the best performing prototype has a heat exchanger efficiency of 53%. The heat pump system used the hydrocarbon isobutane as the refrigerant. Heating COPs of up to 5 were measured. Typically the system can heat air from 0°C to 26°C at 200m^3/hr with a whole system COP of 2.

The contribution to knowledge from this research work is the development of a novel MVHR system and a novel MVHR heat pump system and the establishment of the performances of these systems.

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NOMENCLATURE

Symbol	Meaning	Units
A	Area	m^2
B	Plate fin diameter	m
b	Fin height	m
C	Specific heat capacity	J/kgK
COP	Coefficient of performance	
D	Diameter of heat pipe	m
d	Diameter of cylindrical fin	m
f	Friction factor	
H	Specific enthalpy	KJ/kg
h	Heat transfer coefficient	W/m^2K
k	Thermal conductivity	W/mK
L	Latent heat	KJ/kg
l	Length	m
M	Heat pipe merit number	kW/m^2
M'	Rotating heat pipe merit number	
m	Mass flow rate	kg/s
MVHR	Mechanical ventilation heat recovery	
P	Pressure	Bar
p	Power consumption	kW
Q	Rate of heat transfer	kW
R	Radius of revolution	m
r	Radius of heat pipe	m
Re	Reynolds number	
T	Temperature	K
v	Velocity	m/s
W	Work	kW

ε	Effectiveness	%
η	Efficiency	%
μ	Viscosity	kg/ms
ν	Kinematic viscosity	m^2/s
ρ	Density	kg/m^3
σ	Surface tension	N/m
ω	Angular velocity	rad/s

Sub-scripts

C	Condenser
c	Cooling
E	Evaporator
e	Exhaust
h	Heating
s	Supply

1. INTRODUCTION

The current concerns over global warming, due to greenhouse gas emissions, have brought about an increased awareness of energy use in the built environment. This has prompted the building of homes which are more energy efficient and consequently more airtight. In the past, homes were draughty with adequate levels of ventilation. This was due to poorly fitting doors and windows, chimneys, and air leakage through gaps in the structure. In recent times, the construction of modern doors, windows, and floors usually provides better seals against the entry of outside air than in the past. Some modern homes no longer have chimneys and often in older homes the chimneys have been blocked-up.

The sealing up of homes, however, has greatly contributed to an increased incidence of poor indoor air quality. Inadequate ventilation can cause problems such as condensation on windows, mould, rot and fungus on window frames, damp patches on walls and dust mites in mattresses and carpets. These problems are detrimental to the fabric of a dwelling and can be damaging to the health of the occupants.

Studies in the USA and elsewhere indicate that approximately 93% of time is spent indoors, 5% of time is spent in transit and only 2% outdoors [Otson, 1992]. The combination of greater concentrations of indoor air pollutants and the amount of time spent indoors, suggests that the exposure to airborne pollutants, with the consequent health risks, is far greater now than in the past. It is therefore necessary to provide acceptable levels of ventilation to remove pollutants and maintain a healthy environment. This should be achieved in a manner that does not compromise energy efficiency.

Since opening a window to improve ventilation defeats the purpose of tightening the building's thermal envelope, an alternative means of providing adequate ventilation is required. Alternatives are passive ventilation through trickle vents on windows or by passive stack ventilation, which relies on stack and wind effects to push air through the dwelling. However, heat is lost with the exhaust air in both these systems and neither can provide a response to periods of high moisture production.

Simple extract fans could be used but, again, heat is lost with the outgoing air and they generally only service individual rooms, which results in poor air distribution. A more efficient method of removing pollutants, and adequately ventilating dwellings, is to use whole house mechanical ventilation (MVHR) systems. A supply fan and duct system provides fresh air to living areas and bedrooms, whilst an extract fan and duct system exhausts stale, moist air from the kitchen and bathrooms. A heat exchanger is used to transfer heat from the exhaust air to the supply air. Space heating bills are reduced by preheating the supply air using recovered heat. The cost of electricity used to run the systems is offset by the space heating savings. Many studies have shown that MVHR systems can significantly improve the indoor air quality of a home [DOE 1995, 1996, Korsgaard *et al,* McIntyre 1986, 1989, 1992].

More advanced MVHR systems incorporate heat pumps to provide the heating and cooling required in buildings. Heat pump MVHR systems can supply warm air at temperatures of up to 50°C. Cool air can be provided by reversing the heat pump cycle. In a well insulated airtight house, such a system can supply up to 80% of the seasonal space heating needs at a coefficient of performance of 3 [Kenny, 1994]. The heat pump's high performance means it consumes much less fuel than conventional heating boilers and so emits only a low quantity of CO_2, the principal contributor to the greenhouse effect. The importance of this is highlighted by the commitment made by the UK Government at the Rio Earth Summit, to return CO_2 emissions to 1990 levels by the year 2000. Although heat pumps are frequently employed for industrial and commercial applications, the domestic market in the UK for these systems has been limited due to their high capital cost and maintenance requirements.

The work contained in this thesis is concerned with the development of a novel domestic sized MVHR system using revolving heat pipes. Several systems were designed, built and tested. Each system is based on the same concept of using revolving heat pipes to both impel air and transfer heat. The dual function of the heat pipes minimises the number of components and size of the system. The rotation of the heat pipes enhances heat transfer both within the heat pipes and externally between the air and finning. By virtue of their rotation the accumulation of dirt on the pipe surfaces will be less. This is a feature which has obvious maintenance benefits.

The first group of prototypes developed are heat pipe only systems. These systems are used only for ventilation and heat recovery. The latter prototype incorporates a heat pump to provide additional heating or cooling. All the systems employ environmentally friendly refrigerants with zero ozone depletion potential and zero, or very low, global warming potential. The heat pipes use water as the working fluid, and the heat pump system uses the hydrocarbon isobutane otherwise known as R600a or CARE 10. The University of Nottingham holds two patents on the systems (Patent Nos. GB9522882.1 and GB9507035.5). The patent details and publications which have resulted from this research are shown in Appendix J.

This thesis is organised as follows. Chapter 2 covers the reasons for using MVHR/heat pump systems. Their energy benefits are reviewed, as are the problems associated with poor indoor air quality. Chapter 3 covers the different ventilation strategies used for dwellings and reviews MVHR systems currently available along with the legislation which governs their performance and use. Chapters 4 and 5 review heat pipe and heat pump technologies. In particular, Chapter 4 covers revolving heat pipes as these are used in the prototype systems. The remainder of the thesis is concerned with the development of the prototype MVHR systems. Chapter 6 contains the design work involved in developing different variants of the revolving heat pipe MVHR system. The testing, both in terms of heat recovery efficiency and fan performance, is contained within Chapter 7. Chapters 8 and 9 cover the design and testing of the MVHR heat pump prototype. The project conclusions are contained in Chapter 10. The test results and some ongoing development work are located in the Appendices.

2. WHY USE MVHR SYSTEMS

The change in climate due to greenhouse gas emissions, energy use and the provision of ventilation for acceptable indoor air quality are all reasons for using mechanical ventilation heat recovery (MVHR) systems with or without heat pumping. This chapter discusses these reasons, all of which prompted the research contained in this thesis.

2.1 Climate change

During the last two decades, there has been increasing concern about world energy consumption and resources. Twenty years ago, during the first wave of global concern about energy use and the environment, the main concern was that non-renewable resources such as oil and gas would be exhausted within a few decades. However, those fears proved unfounded because new reserves were found or substitutes developed. More, recently we have begun to understand how the use of energy itself is damaging the environment.

Global warming, due to the greenhouse effect, has emerged in recent years as one of the most urgent environmental problems humankind faces. Scientists are now in agreement that the balance of evidence suggests a discernible human influence on global climate. The world is about 0.6°C warmer than it was 100 years ago, with the three warmest years globally being 1998, 1997 and 1995 [The Times, 1999].

The composition of the atmosphere affects the amount of radiation absorbed and emitted. Some naturally occurring atmospheric gases, including water vapour, carbon dioxide, methane, nitrous oxide and ozone, are more efficient at absorbing terrestrial (outgoing long wave) radiation than solar (incoming short wave) radiation. Because of this they trap outgoing radiation, keeping the surface and lower atmosphere warmer than it would be without these gases present. This is the natural greenhouse effect. Without these greenhouse gases, the surface of the Earth would be 20°C to 30°C colder and inhospitable. Figure 2.1 shows a simplified diagram illustrating the greenhouse effect.

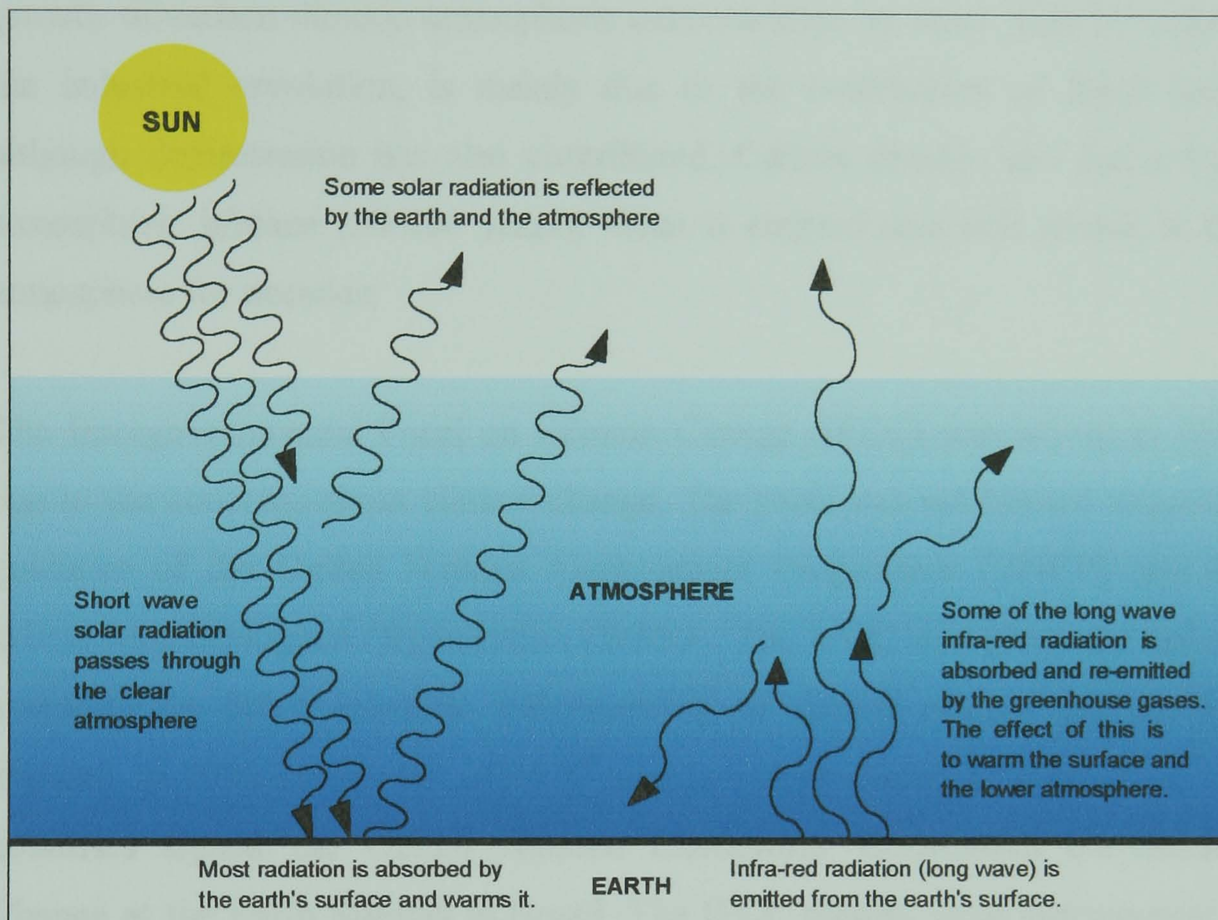


Figure 2.1: A Simplified Diagram Illustrating The Greenhouse Effect [IPCC, 1990]

Human activity has contributed to the greenhouse effect by increasing the atmospheric concentration of some of the naturally occurring greenhouse gases, such as carbon dioxide and methane. In addition, mankind has been adding new greenhouse gases such as chlorofluorocarbons (CFCs). This is an enhanced greenhouse effect but is usually just referred to as the greenhouse effect. Humankind is therefore capable of raising the global average temperature, a phenomenon known as global warming.

Carbon dioxide has made the largest contribution to the enhanced greenhouse effect. Although the level of carbon dioxide has varied naturally over time, the current concentration is probably higher than at any time in the last 160,000 years. Increases over the last 200 years have been dramatic. Pre-industrial (1750-1800) atmospheric concentration of carbon dioxide was 280 parts per million by volume (ppmv). The 1990 value was 353 ppmv [IPCC 1990]. The growth of carbon dioxide atmospheric concentration by more than 25% since the industrial revolution, is mainly due to the combustion of fossil fuels, although deforestation has also contributed. Carbon dioxide also has a long atmospheric lifetime (50-200 years). What is emitted now will remain in the atmosphere for decades.

The Intergovernmental Panel on Climate Change (IPCC) was set up in 1988 due to the concerns about climate change. The panel was established under the guidance of the United Nations Environment Programme (UNEP) and the World Meteorological Organisation (WMO). The IPCC was given the task of assessing the latest scientific understanding of climate change and possible impacts. In 1990 the first set of IPCC reports was produced. In June 1992, 155 countries signed the United Nations Framework Convention on Climate Change at the Earth Summit in Brazil. The IPCC reports were instrumental in the negotiations which led to the signings of the Convention. Initially, the Convention committed all developed country parties to take measures aimed at returning carbon dioxide emissions to 1990 levels by the end of the century. Since then, developed countries have agreed to cut their greenhouse gas emissions by 5%, relative to 1990 levels, by 2008-2012. This is a clear indication of the concern around the world.

Climate change in the future depends on how quickly and to what extent the concentration of greenhouse gases in the atmosphere increases. Complex mathematical models have been developed to predict climatic response to the increase of carbon dioxide and other greenhouse gases. These models have predicted variations in regional climates and significant rises in sea level, through changes in temperature and precipitation. It is thought people would be affected due to the impact on water resources, food production and health.

A new report by the Met Office warns that if nothing is done to reduce rates of carbon dioxide emission now, 290 million people could be put at risk of malaria and another 80 million could face the annual threat of flooding in just three generations time. An estimated three billion people in Africa, the Middle East and the Indian subcontinent will struggle to find adequate supplies of clean drinking water and large parts of South America could lose their rainforests for good [Met Office 1999].

Because of the threat of climate change, it is important that greenhouse gas emissions are reduced. It is therefore the responsibility of architects and engineers, working in the built environment, to design buildings and environmental systems that use energy efficiently. Such systems not only benefit the environment, but also consumers due to reduced energy consumption and associated running costs.

2.2 Energy use in dwellings

The building stock accounts for almost 50% of the total UK delivered energy use and a similar percentage of carbon dioxide emissions. By far the greatest part of this is from the energy used by dwellings (almost 60%). This accounts for 28% of the UK primary fuel consumption, with 56% of this used in space heating and 25% in water heating. This energy use results in the emission of 33 million tonnes of carbon each year [Shorrock (1992)]. The Department of the Environment, Transport and the Regions (DETR) has estimated that a 30% reduction in domestic energy use is possible through energy conservation and efficiency, without a drop in living standards [DoE 1996].

Dwellings represent the largest single category of buildings in terms of numbers and energy consumption and, therefore, present the greatest potential for energy savings in buildings. The long lifetimes of most dwellings mean that the design decisions made at any point in time, effectively represent a long term commitment to energy consumption. For example, the dwellings which were built before any thermal regulations existed still make up the majority of the housing stock. Thus, design decisions which were made in the past have a strong influence on energy use now, and continue to have an influence into the future. Similarly, design decisions made now set the baseline performance of the stock in the future. Because of this it is important that legislation ensures that houses are built with energy efficiency in mind.

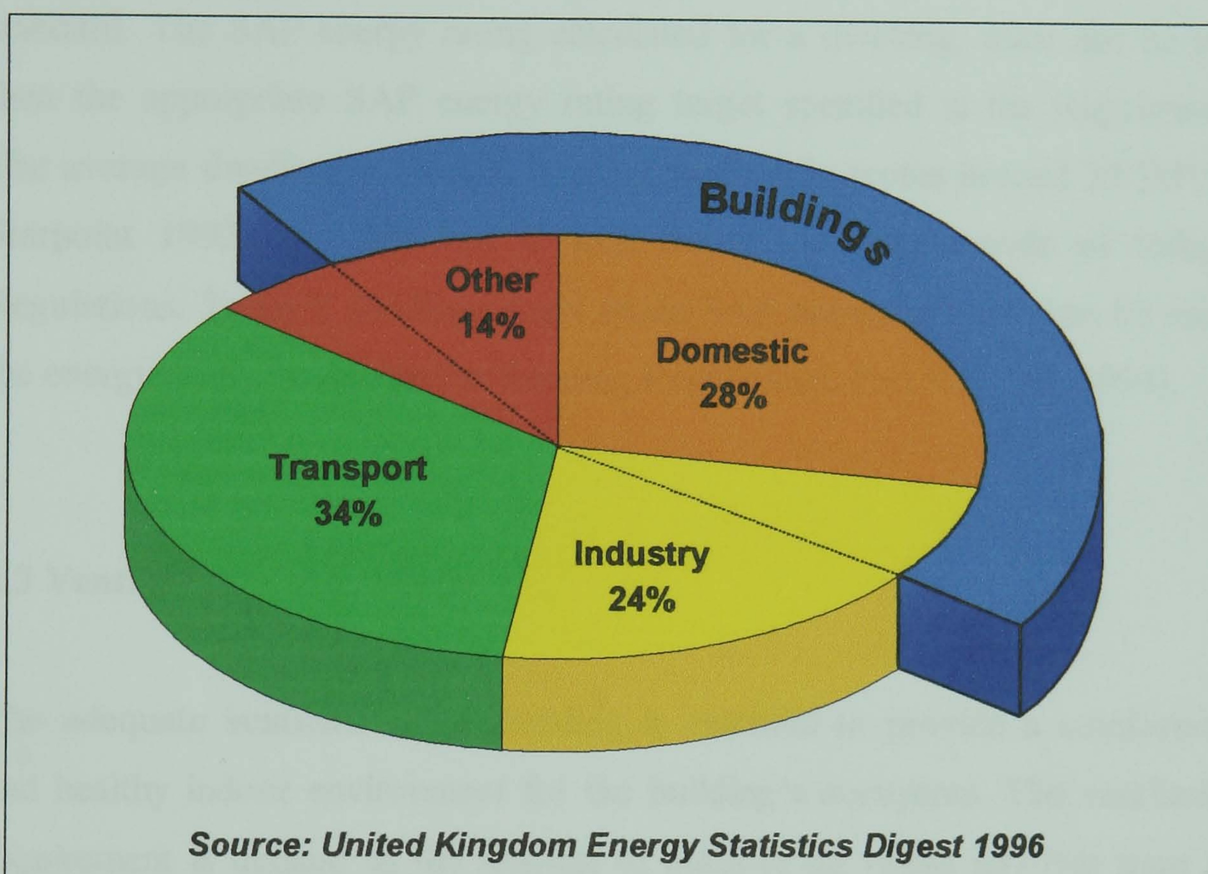


Figure 2.2: UK Energy Use [Wyatt, 1997]

The UK is already fulfilling its commitment to the Convention signed at the Earth Summit, by considering a series of measures to secure reductions in carbon dioxide emissions. One measure already implemented that is aimed to reduce domestic energy use, is the addition of VAT on domestic fuel and power. In addition, the last revisions to Part L of the Building Regulations (1995) ensure that energy consumption in dwellings is reduced, by enforcing more rigorous thermal specifications. All new homes in the UK are built conforming to guidelines that reduce fabric heat losses and limit infiltration. Every new dwelling also requires a Standard Assessment Procedure Energy Rating (SAP rating). The rating is calculated in accordance with the Standard

Assessment Procedure given in Appendix G of the Building Regulations, Part L. The SAP rating is calculated by working out the cost of heating a home and providing hot water, based on a standard heating pattern and typical occupancy of that home (energy use for lights and appliances are not included). This cost is then converted to a rate per unit area and converted into an efficiency score on a scale of 1 to 100. A SAP rating of 1 represents a poor standard of energy efficiency, while rating of 100 represents a very high standard. The SAP energy rating calculated for a dwelling, must not be less than the appropriate SAP energy rating target specified in the Regulations. The average dwelling in the UK housing stock only scores around 39 [MVM Starpoint 1992], a value that is well below the requirements of today's Regulations. Today's dwellings must attain SAP ratings greater than 80 using the energy rating method in the Building Regulations, Part L [DOE, 1994].

2.3 Ventilation requirements

The adequate ventilation of a building is essential to provide a comfortable and healthy indoor environment for the building's occupants. The ventilation requirement is defined as the amount of outdoor air (fresh air) that must be supplied to a space, to meet criteria associated with the use of that space.

The provision of a fresh air supply is necessary for the following purposes:

- Human respiration.
- Dilution and removal of indoor air pollutants.
- Control of relative humidity.
- Thermal comfort.
- Combustion appliances.

2.3.1 Human respiration

In any occupied space, ventilation is essential for the provision of oxygen and removal of contaminated air. Fresh air contains about 21% oxygen and 0.04% carbon dioxide, while expired air contains about 16% oxygen and 4% carbon dioxide [McMullan 1993]. The ventilation rate required for a given space is based on the need to dilute carbon dioxide, rather than to supply oxygen for breathing. It is desirable to keep the concentration of carbon dioxide within an occupied space down to about 0.15%. The maximum allowable carbon dioxide concentration, for 8 hours exposure, for healthy adults is 0.5% [Health & Safety Executive 1985]. Table 2.1 shows the minimum ventilation rates for various activities to prevent this limit being exceeded. It is also common practice to specify ventilation rates by the removal of body odours rather than CO_2 concentrations. Higher rates will result from using the body odour criteria.

Activity	**Minimum ventilation requirement for 0.5% CO_2 limit (litres/second per person)**
Seated quietly	0.8
Light work	1.3-2.6
Moderate work	2.6-3.9
Heavy work	3.9-5.3
Very heavy work	5.3-6.4

Table 2.1: Ventilation Rates Required To Limit CO_2 Concentration

[CIBSE, 1986]

2.3.2 Indoor air pollutants

As new houses are built more airtight and existing houses are being 'tightened up' in an effort to reduce energy costs, the natural air leakage in homes is no longer sufficient to reduce indoor air pollutants. Until fairly recently, there was so much air leakage in and out of most houses that air pollutants were diluted and removed. Now, without that air leakage, the pollutants can build up inside the house to levels where they are damaging to human health.

The average adult spends 90% of their time indoors but this is where the air can be most polluted, particularly in a home if it is well sealed [ECD]. Air of a high quality is required to create good conditions for the building structure and the people inside. This can be achieved by good ventilation.

Poor ventilation is a cause of:

- Sick building syndrome.
- Condensation – water condensing on walls and windows.
- Unpleasant mould growth in bathrooms and damp places which can damage decorations.
- High levels of airborne pollutants e.g., dust mites, formaldehyde .
- Lingering smells from cooking, smoking and other household activities.
- Poor health from damp conditions.

2.3.2.1 Sick Building Syndrome

Sick Building Syndrome (SBS) is a phenomenon recognised by the World Health Authority. It is where the occupants of some buildings appear to suffer ill health more often than might reasonably be expected. Building Related Illness (BRI) and Tight Building Syndrome (TBS) are two other terms used to describe Sick Building Syndrome.

Sick Building Syndrome Effects:

The illnesses related to Sick Building Syndrome generate the following types of symptoms:

- Eye, nose and throat infections.
- Dryness of throat, nose and skin.
- Breathing difficulties and chest tightness.
- Headaches, nausea and dizziness.
- Mental fatigue.
- Skin rashes.
- Aching muscles and flu-like symptoms.

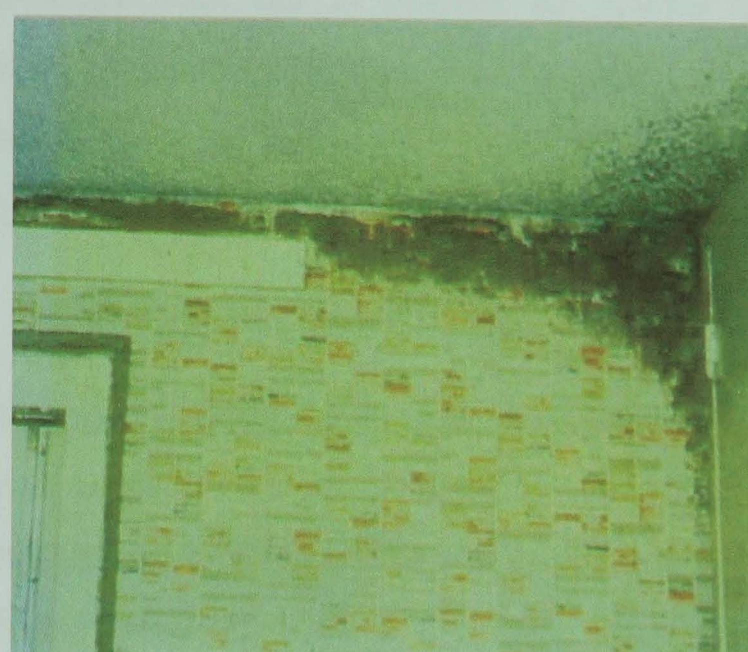
Most complaints of Sick Building Syndrome occur in large offices and as this thesis is concentrating on dwellings, the subject has only briefly been covered.

2.3.2.2 Condensation and mould growth

An average family produces 15 litres (26 pints) of water vapour each day, partly via perspiration, partly through breathing, but also from cooking, bathing, washing and drying clothes [London Electricity]. This moisture is unable to escape from a well-insulated airtight house.

Photograph 2.1: Condensation On Windows [London Electricity]

The build up of condensation on the inside of windows is the most obvious indicator of moisture related problems. More serious moisture problems are the growth of mould, damp patches on walls and the presence of wood rot on doors and window frames. Currently 35% of the UK housing stock is affected by condensation and 17% by mould [Woolliscroft 1997].



Photograph 2.2: Mould Growth On Walls [London Electricity]

Mould growth occurs in the home in areas that are excessively moist. Growth of mould tends to be greater in the summer than it is in the winter. This is due to higher temperatures and levels of humidity. Not only can mould growth result in dampness and deterioration of a building's fabric, a range of psychological conditions and physiological illnesses can be suffered by a building's occupants. The Medical Research Council's Epidemiology unit at the Royal Edinburgh Hospital found that children in damp and mouldy homes had significantly more wheezing, sore throats, persistent headaches, fevers and runny noses than children in dry homes [Platt *et al* 1989]. Damp surfaces which promote the growth of, and long term exposure to, mould can lead to hypersensitivity with severe reactions including breathing difficulties [Environment Committee 1991].

As stated by Oreszczyn (1992), the main requirements for mould to grow are:

- Mould spores which are always present in the air – there are several hundred per cubic metre in the outside air even in winter.
- Oxygen.
- A temperature between 0°C and 40°C – individual species of moulds have an optimum temperature but in practice all moulds will tolerate wall temperatures encountered in buildings, even freezing.
- Some nutrients – most moulds can thrive on the nutrients present in dust and other deposits, even in well cleaned and maintained houses.
- Moisture.

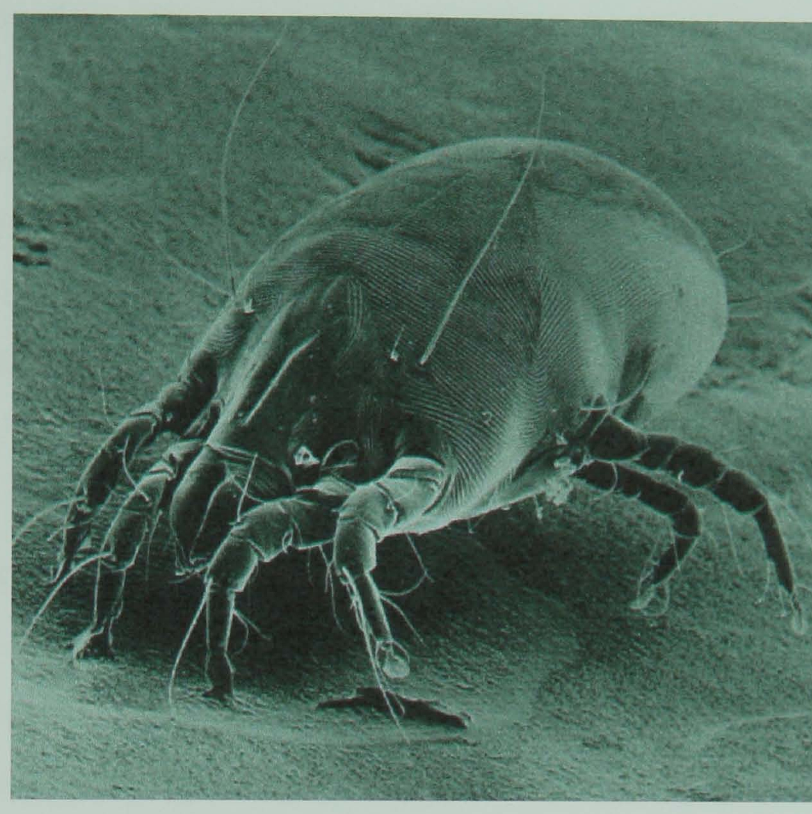
To avoid condensation and mould growth, it is widely agreed that indoor levels of relative humidity should be kept below 70%. One method of achieving this is to use an MVHR system to exhaust moist, stale inside air and replace it with fresh outside air.

2.3.2.3 Dust mites

House dust mites (*Dermatophagoides pteronyssinus*) are arachnids which thrive in areas of high humidity and warmth. They are found in the furnishings of most homes and are particularly abundant in carpets, beds, cushions and blankets. Dust mites are about 0.35mm long and feed on decomposing organic matter such as skin scales from humans and animals.

The most common allergen causing asthma in the UK is the house dust mite. However, the problem is not from the mites themselves, but from their faeces. A protein in the faeces causes an extreme allergic response in sufferers. Because of this, dust mite faeces present in dust can be regarded as one of the most potent allergens present in homes today.

More than two million people in the UK are asthmatic and numbers are showing a steady increase. In 1989, the National Health Service spent approximately £217 million on drugs for asthma, which is about 8% of the total NHS budget. In the UK 670,000 people could be suffering from asthma because they are allergic to dust mite faeces [ECD].



Photograph 2.3: The House Dust Mite **[London Electricity]**

Dust mites live in an environment where there is no liquid water available. The moisture balance is therefore critical to their survival, as they get their water by absorbing water vapour through their skin. If the relative humidity is too low then they cannot absorb enough water, which inhibits their ability to reproduce and they will eventually die. Laboratory studies show that the optimum conditions for mites are 25°C and 80% relative humidity [McIntyre 1992]. A World Health Organisation working party has proposed that humidity below a moisture content of 7g/kg will inhibit the growth of mites [Platts-Mills 1989]. Work by Korsgaard (1983) showed that the number of dust mites decreases considerably if the relative humidity is below 45% at room temperature.

The widespread occurrence of dust mites and its potential for causing serious effects on health have prompted research to establish ways of removing it from homes. Studies in Denmark have shown that the health of people with asthma improved considerably when they moved to well ventilated buildings with mechanical ventilation systems [Korsgaard]. MVHR systems were used to lower relative humidity and, as a result, reduce levels of house dust mites.

UK studies undertaken by EA Technology in cooperation with the Building Research Establishment [McIntyre 1992, 1996] concur with the Danish research. These studies concluded that the presence of dust mites and levels of relative humidity in houses using MVHR systems, were below levels found in houses where there were no MVHR systems used.

2.3.2.4 Gases

The following section contains a brief review of several gases that are regarded as the main gaseous indoor air contaminants within dwellings.

Nitrogen Dioxide:

Nitrogen dioxide (NO_2) is a common atmospheric pollutant. It is a highly toxic, reddish brown gas with a very strong odour and is produced when there is combustion of fossil fuels in air. The process of combustion converts some atmospheric nitrogen into NO_2 and some into nitric oxide (NO). Most of the NO then reacts with oxygen to form more NO_2. The main source of NO_2 in homes is from the burning of heating and cooking fuels. It is also produced by tobacco smoking.

When exposed to nitrogen dioxide people can experience extreme irritation to the skin, eyes and mucous membranes. The health effects are dependent upon the level of exposure and The World Health Organisation recommends that NO_2 concentrations should not exceed 150 $\mu g/m^3$ (80 ppb) over 24 hours [WHO 1987]. This is based on the lowest concentration known to affect asthmatics. Although unconfirmed, long term exposure may also increase the risk of respiratory disease, particularly in children.

The Building Research Establishment report [Berry *et al* 1996] on Indoor Air Quality in Homes refers to a pilot study of NO_2 concentrations in homes undertaken in the Northwest of England. The study confirmed earlier findings that the main factor affecting NO_2 levels in homes is gas cooking. The highest kitchen measurement, in a gas cooking home, was found to be 158.8 $\mu g/m^3$ (85 ppb) [Raw and Coward 1991].

The Building Research Establishment concludes from its own study of NO_2 concentrations in 174 homes, that gas cooking was the dominant indoor source of NO_2. The study also found that 1% of the sample homes exceeded the WHO guideline. This figure equates to over a quarter of a million homes if the same percentage was found in the UK as a whole [Berry *et al* 1996].

The levels of NO_2 could be controlled by using an MVHR system. Extracted air from the kitchen, a potential area for high concentrations of NO_2, would be replaced by fresh air from outside.

Volatile Organic Compounds (VOCs):

Volatile Organic Compounds (VOCs) are highly evaporative chemicals that volatilise into the atmosphere at room temperature. At present, over 400 separate VOCs have been identified in the home, of which, over 250 can be found in carpets [Residential Energy Efficiency Database 1999]. The most common and best known VOC is Formaldehyde and, because of this, it will be covered separately.

VOCs are emitted as gases from items such as building materials, plywood, particle board, adhesives, furniture, carpets, curtains, cleaning agents, aerosol sprays, cigarette smoke and paints. A review by Gustafsson (1992) of scientific literature, found 24 cases of building materials acting as major sources of VOCs in indoor air which resulted in complaints by occupants.

The effects on people's health from the presence of VOCs in indoor air have been experienced widely throughout the developed world. The most common health effects include nausea, dizziness, headaches, drowsiness, respiratory complaints, sinus congestion and irritation to the skin, nose and eyes. All of these symptoms have been experienced by people exposed to VOCs in indoor environments. Exposure to high concentrations is more serious and can affect the central nervous system and cause kidney or liver damage. Thankfully, exposure in such concentrations is unlikely in the home.

Exposure to VOCs can be from a mixture of many individual gases that interact with each other. The collective term for this mixture is Total Volatile Organic Compound (TVOC). The National Health and Medical Research Council of Australia has recommended 500 $\mu g/m^3$ as the level of concern for TVOCs with no single compound contributing to more than 50% of the total [Dingle 1993].

The Building Research Establishment's study of indoor air quality in homes [Berry *et al* 1996] measured TVOC concentrations in 174 homes in Avon over a twelve month period. For all houses in the study, the mean bedroom concentration was 415 $\mu g/m^3$. The concentrations in other rooms were found to be similar and the indoor concentration was ten times that measured outdoors. Twenty five percent of the homes studied, had an annual mean TVOC concentration which exceeded the guideline value proposed by the National Health and Medical Research Council of Australia. Homes which were newly decorated or had new furnishings were found to have higher TVOC concentrations.

Adequate ventilation should be provided in tightly sealed homes to avoid any health problems. Once again, this could be achieved by using an MVHR system.

Formaldehyde:

At normal ambient temperatures, formaldehyde is a colourless gas with a pungent odour. The gas occurs naturally and is also industrially produced. It is used in many manufacturing processes associated with household items.

Formaldehyde is released into the indoor air of homes from all the sources summarised in the section on VOCs. In the UK, urea formaldehyde foam was used extensively as retrofit cavity wall insulation in the early 1980s. This led to elevated concentrations of formaldehyde in these homes.

Formaldehyde gas is released into the air unless the materials it is used in are properly sealed. The air temperature and relative humidity influence the rate at which the gas is released into the air. Temperature increases of 5°C to 6°C can double the gas's concentration, while an increase in the relative humidity from 30% to 70% can cause a 40% rise in the formaldehyde concentration. The concentration can increase to as much as five times its original level if both temperature and relative humidity are raised [Residential Energy Efficiency Database 1999].

The health effects from exposure to formaldehyde include: Eye, nose and throat irritation, headaches, skin rashes and nausea. The International Agency for Research on Cancer (IARC 1982) has shown formaldehyde to cause cancer in animals and class it as a probable human carcinogen.

The odour detection threshold for formaldehyde is approximately $0.1 mg/m^3$ with eye and throat irritation starting at about $0.5 mg/m^3$. The World Health Organization have suggested that 'in order to avoid complaints of sensitive people about indoor air in non-industrial buildings, the formaldehyde concentration should be below $0.1 \mu g/L$ as a 30 minute average, and this is recommended as an air quality guideline value' [WHO 1987].

A study in the USA found that children had an increased likelihood of suffering from asthma or chronic bronchitis when exposed to 60-120 ppb formaldehyde at home, particularly if also exposed to tobacco smoke [Kryzanowski 1990].

The study of indoor air quality in homes conducted by the Building Research Establishment [Berry *et al* 1996] measured formaldehyde levels in 180 UK homes over a three-day period each month for 12 consecutive months. The study found that in 12 homes at least one three-day mean concentration exceeded the WHO recommended air quality guideline value. The study also discovered a strong relationship between mean formaldehyde concentration and the age of dwellings. The mean formaldehyde level was shown to decrease as the age of the dwellings increased. This could be attributed to higher concentrations of formaldehyde present in newer building materials and furnishings used in homes built more recently. It could also be due to the lower air change rates of newer dwellings subject to tighter building regulations. In cases where the minimum exposure levels are exceeded, the use of MVHR systems would provide adequate ventilation to reduce formaldehyde concentrations by providing fresh air and reducing relative humidity.

Radon:

Radon is a radioactive gas which occurs naturally as a product of the radioactive decay of uranium in the Earth's crust. It has no colour, taste or odour and is found in all soils in various concentrations. It is, however, commonly associated with areas where granite exists as the underlying rock structure.

Radon gas enters a house through cracks and joints in the foundations and walls of the ground floor. The decay products of radon can attach themselves to suspended particles in the air. People are then at risk of inhaling these particles which can be deposited on the respiratory tract. The subsequent decay of the inhaled particles leads to an increased risk of developing lung cancer. Radon is now believed to be the second leading cause of lung cancer, after cigarette smoke [Residential Energy Efficiency Database 1999].

The concentrations of radon found in homes varies depending on the levels present in the surrounding ground. Levels of the gas in the ground are usually low and, therefore, contamination is not a problem for many home owners.

It is essential that homes are properly sealed in areas where radon pollution is a problem. Adequate sealing goes some way to avoiding the problem of gas ingress into the home. Further reductions of the gas could be achieved by using a balanced MVHR system to increase the ventilation rate. An exhaust only unbalanced ventilation system should not be used, as it will create a negative pressure in the home, thereby drawing more air and radon into the building. Balanced and unbalanced MVHR systems are explained in more detail in Chapter 3.

2.3.2.5 Reduction in pollutants using an MVHR system

The UK manufacturer Baxi markets an MVHR system based on a static plate heat exchanger (see Chapter 7.1.7). Independent clinical trials were carried out on the Baxi Clean Air System demonstrating the effectiveness of an MVHR system for reducing indoor air pollutants (see Figure 2.3).

The British Allergy Foundation has awarded Baxi their prestigious seal of approval and recommends the system to asthma and allergy sufferers. This is one example of the widespread acknowledgement MVHR systems receive for their control of pollutants affecting the health of occupants within a home.

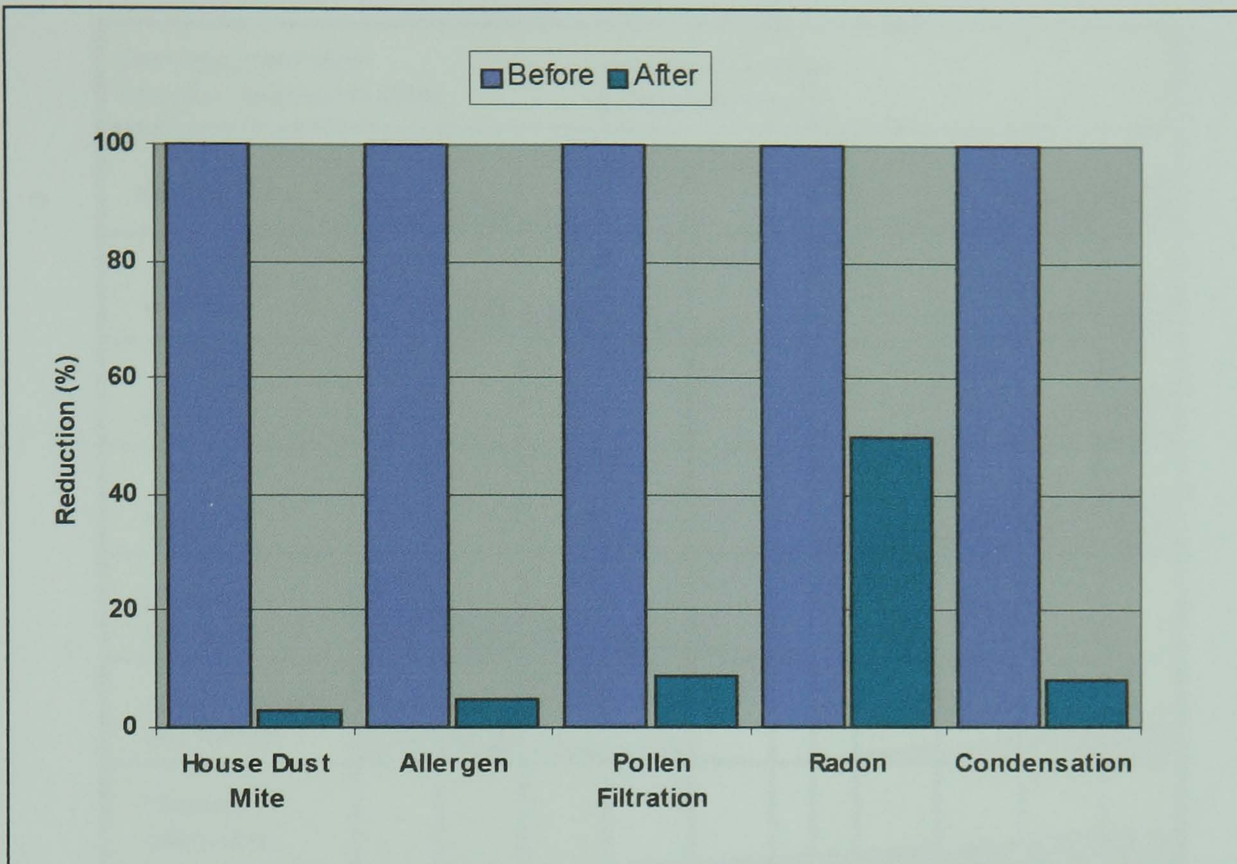


Figure 2.3: Reduction In Pollutants With Baxi Clean Air MVHR System [Baxi Air Management]

2.3.3 Control of relative humidity

Relative humidity is the ratio of the actual partial pressure of the water vapour to the partial pressure of the water vapour when the air is saturated at the same temperature. Relative humidity is often expressed as a percentage. The ideal relative humidity for a home is between 30% and 55%. Problems can begin to occur when the relative humidity is too much above or below these levels. The previous section on indoor air pollution commented on how levels of relative humidity can affect the concentrations of different airborne contaminants. Figure 2.4 illustrates the optimum level of relative humidity in a home required to achieve acceptable indoor air quality.

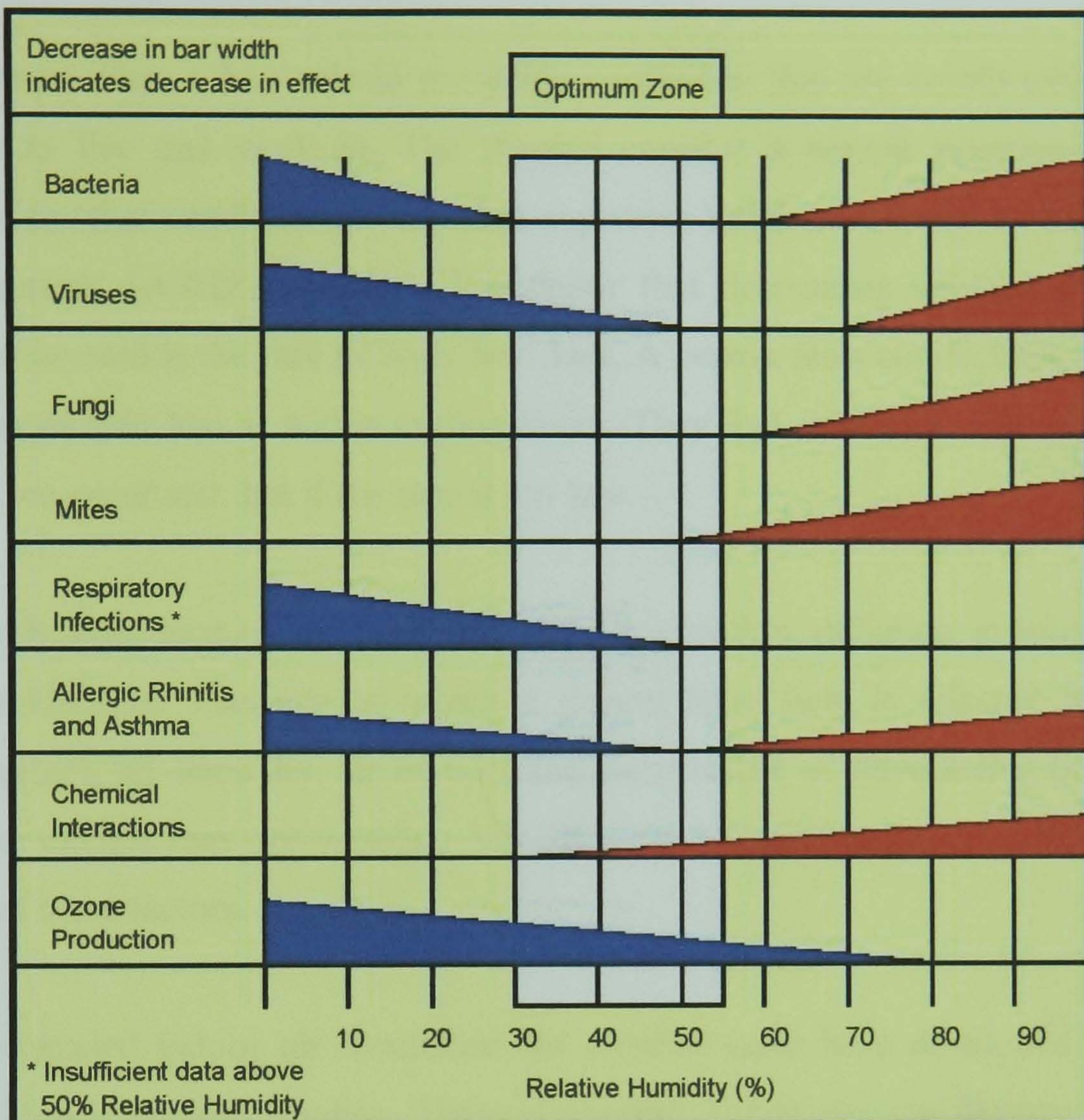


Figure 2.4: Relative Humidity And Air Quality

[Residential Energy Efficiency Database 1999]

Low levels of relative humidity in a dwelling can cause sore throats, sinus congestion and skin disorders for the home's occupants. If the relative humidity is too high, mould growth can occur and there can be an increase in the off-gassing of VOCs within the home, both of which can adversely affect human health.

To control the level of relative humidity in the home it is essential that there is adequate ventilation. At the very least there should be exhaust fans in the bathroom and kitchen and preferably a whole house MVHR system.

2.3.4 Thermal comfort

Ventilation plays a key role in providing conditions that are comfortable for people to live and work in. The thermal comfort a person experiences is defined as 'that condition of mind that expresses satisfaction with the thermal environment' [ASHRAE 1992]. The factor that determines whether people feel hot or cold is the rate of body heat loss. A person feels comfortable when the rate of heat loss is within certain limits. They feel cold if the rate of heat loss is too great and hot if the rate is too low.

The body loses heat to the surroundings by convection, radiation, evaporation and conduction. The rate at which a person loses heat is affected by air temperature, air humidity, air motion, the temperature of surrounding objects and the clothes they are wearing. MVHR systems give control over the first three of these factors.

Recommended indoor air conditions for comfort have been developed from many human comfort studies. Thermal comfort conditions are illustrated by thermal comfort charts. The comfort chart shown in Figure 2.5 illustrates the comfort zones where 80% of sedentary or slightly active persons find the environment thermally acceptable. The chart takes into account changes in clothing for seasonal weather variations by specifying separate comfort zones for winter and summer.

A clothing scale is used to represent how much clothing a person is wearing. The clothing values range from 0clo to 4clo, where 1clo represents 0.155m^2k/W of insulation. The summer and winter clo-values used in the chart are 0.5clo and 0.9clo (0.078 and 0.14 m^2k/W) respectively. Light trousers and a shirt or a light dress and blouse are examples of clothing with a clo-value of 0.5. Examples of clothing with a clo-value of 0.9 are a business suit or a dress with a jumper.

In the middle of a shaded zone a person wearing the appropriate summer or winter clothing would have negligible thermal sensation. Near the boundary of the warmer zone a person would feel warmer and towards the boundary of the cooler zone a person would feel cooler.

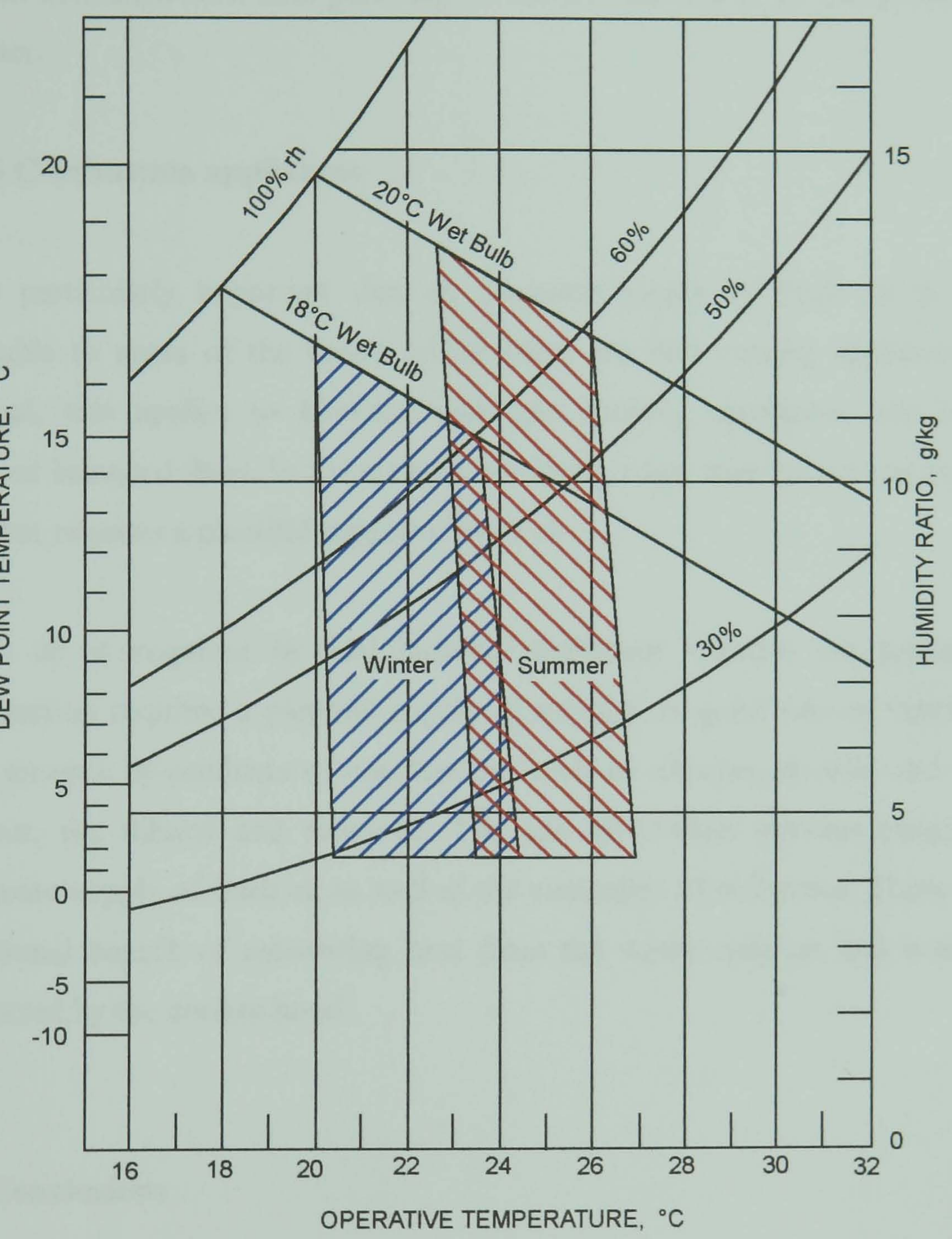


Figure 2.5: ASHRAE Summer and Winter Comfort Zones

(Acceptable ranges of operative temperature and humidity for people in typical summer and winter clothing during primarily sedentary activity).

[ASHRAE 1997]

It was stated in the previous section that, for the best air quality, the relative humidity in the home should be kept between 30% and 55%. This optimum zone also falls within the thermal comfort zones illustrated in Figure 2.5. This further strengthens the requirement for controlled relative humidity through adequate ventilation. To maintain comfort temperatures throughout the year homes require heating and, in some cases, cooling. The use of an MVHR system combined with heat pumping would provide this in an energy efficient manner.

2.3.5 Combustion appliances

It is particularly important that an adequate supply of fresh air is made available to areas of the house where there are fuel burning appliances. In general, this applies to kitchens with gas cooking appliances and boilers without balanced flues. In some cases, the living room may have a gas or open fire that requires a plentiful supply of fresh air.

Fresh air is required by fuel burning appliances because the process of combustion requires a plentiful supply of oxygen. A good rate of ventilation also ensures by-products of combustion, such as nitrogen dioxide and water vapour, are diluted and removed. The use of MVHR systems ensures an adequate supply of fresh air as well as the extraction of pollutants. There is the additional benefit of recovering heat from the warm stale air and warm air extracted by the cooker hood.

2.4 Conclusions

By highlighting the importance of energy conservation, indoor air quality and ventilation requirements, this chapter has explained the need for providing adequate ventilation in an energy efficient manner. Whole house MVHR systems provide fresh air throughout the house, whilst removing unwanted stale air. Heat is recovered from the exhaust air to save energy. The occupants of houses with MVHR systems no longer have to open windows to provide ventilation, so the problems of heat loss, noise and security are eliminated. Systems incorporating a heat pump can supply air at a temperature and humidity that gives thermal comfort conditions to the home's occupants.

3. DOMESTIC MVHR SYSTEMS

Mechanical ventilation heat recovery systems have been used in dwellings on the Continent and in North America for over 20 years. Many studies were undertaken in the 1980s by research bodies from the UK and around the world. Papers published on the subject proclaimed the benefits of using whole house MVHR systems. The systems were shown to save energy and increase indoor air quality.

The use of these systems is widespread in Scandinavia and parts of North America. In some countries, such as Sweden, the installation of MVHR systems is covered by law. By 1980 in Sweden, there were approximately 20,000 MVHR systems installed, and in that year nearly all the new houses built had a ventilation system installed [Svensson, 1982].

Although the benefits of MVHR systems have been shown, the uptake of systems in the UK has been slow. The slow uptake is due to several reasons. The majority of houses in the UK are less airtight than houses on the Continent. Enough fresh air enters most dwellings in the UK by infiltration and, therefore, the requirement for mechanical ventilation systems is less. It should also be noted that MVHR systems should be installed in airtight houses in order to work efficiently. It is generally held that the British are more conservative in their attitude towards innovation than people on the Continent and in North American [Hendley, 1998]. High capital costs may also be a barrier to sales.

Although sales of domestic MVHR systems in the UK are still not large, there has been a sharp increase in sales over recent years. The market value has increased from £1.3 million in 1995 to £2.1 million in 1999, an increase of 62% over the last five years [BSRIA 2000]. Figure 3.1 illustrates the rise in value of the UK MVHR system market. The recent rise in sales could be attributed to the more stringent Building Regulations requiring houses to be built more airtight and with higher levels of insulation. Fresh air supplied to new houses requires a different means of delivery other than infiltration.

There is also greater public awareness of the health problems associated with poor indoor air quality. MVHR systems are being marketed as an energy efficient way of providing clean fresh air to the home.

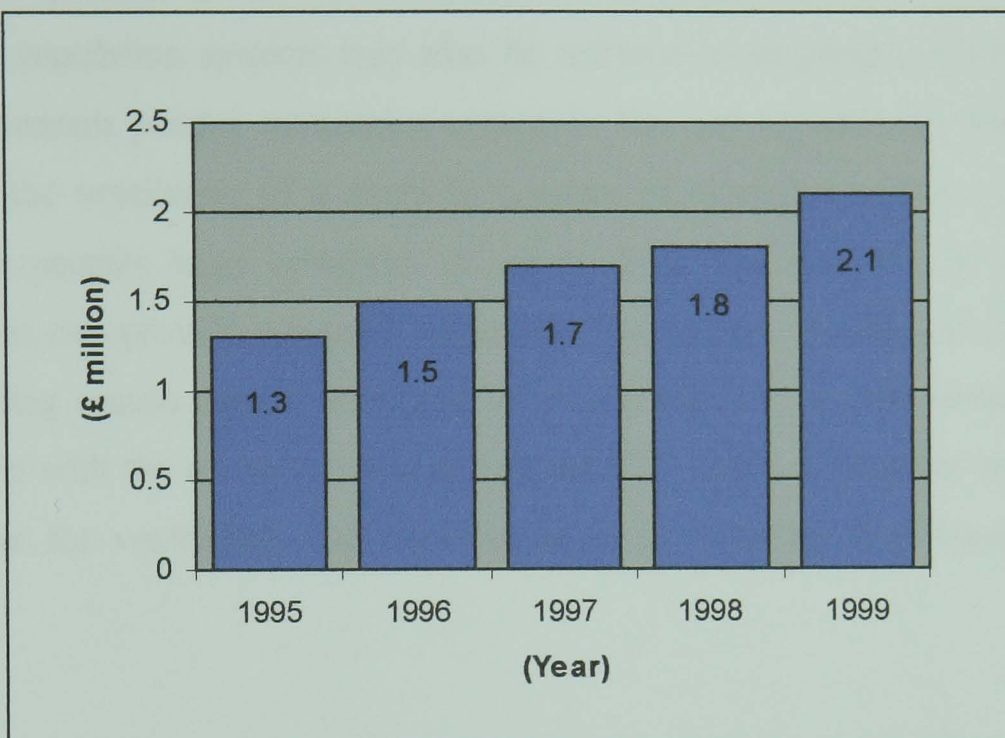


Figure 3.1: UK Historical Trend Of Whole House MVHR Systems, By Value (£ million, 1995-1999)

The following sections review ventilation strategies, legislation and airtightness requirements for dwellings. A summary of ventilation heat recovery equipment is covered as well as the theory used to determine the system efficiency and coefficient of performance.

3.1 Ventilation strategies

Dwellings can be ventilated using the following ventilation strategies:

- Windows only.
- Passive stack.
- Mechanical extract.
- Whole house mechanical ventilation.
- Whole house mechanical ventilation with heat recovery (MVHR).
- Whole house mechanical ventilation heat recovery heat pumps.

3.1.1 Windows only

The majority of houses in the UK are naturally ventilated. Natural ventilation refers to systems that do not use mechanical equipment in order to operate. Natural ventilation systems may also be referred to as passive systems. The most common passive ventilation system is the use of windows. Occupants control the ventilation of a room by opening or closing a window. An open window permits large quantities of air to flow into and out of a house. Windows can provide adequate air supply for summer cooling. However, in the heating season energy is wasted by allowing heat to escape through open windows with the outgoing stale air. Figures 3.2 and 3.3 illustrate the use of windows for ventilation and draw attention to resulting heat losses in the winter.

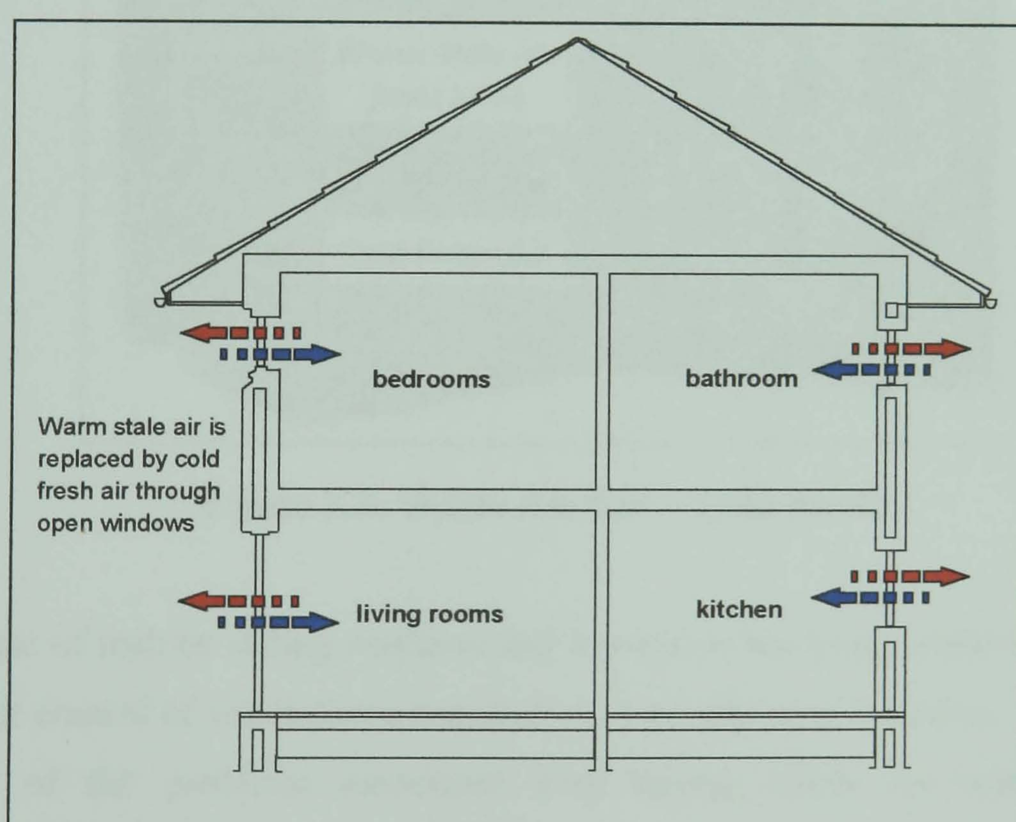


Figure 3.2: Ventilation By Windows Only

Advantages:

- Cheap to install.
- Provides some ventilation.

Disadvantages:

- The rate of ventilation cannot be controlled.
- The distribution of air is likely to be uneven.
- Heat is lost through open windows in cold weather.
- Cold air that enters the house must be heated.
- Cold drafts caused by open windows can cause extreme discomfort .
- Open windows compromise the security of a house.
- Outdoor pollutants are able to enter houses freely through open windows.
- Noise pollution.

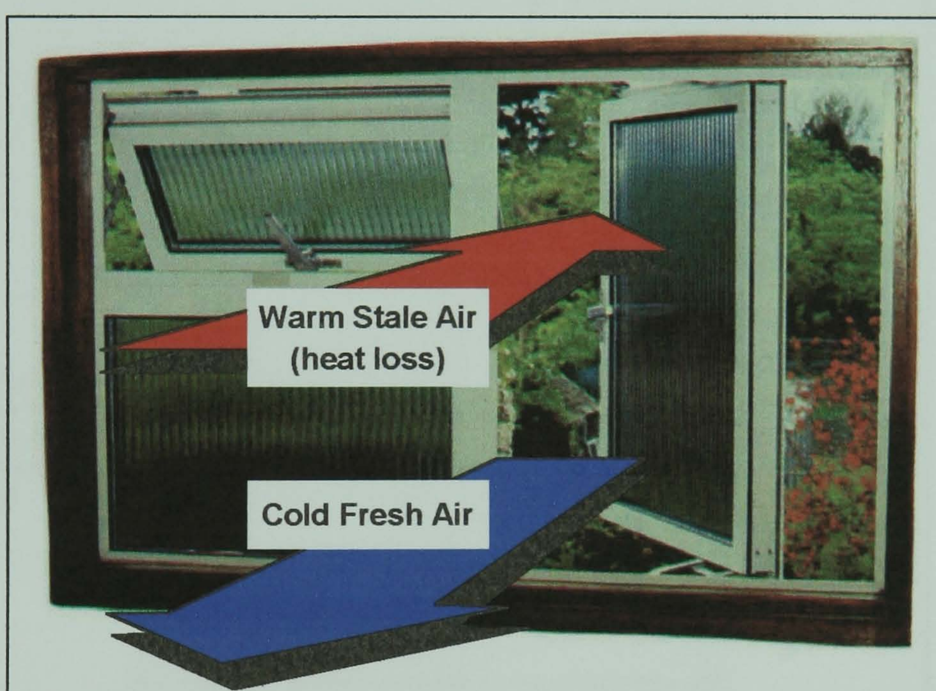


Figure 3.3: Warm Air Out – Cold Air In

The use of sash or sliding windows and louvres or top hung windows provide greater control of ventilation compared to large side hung windows. However, many of the problems associated with relying solely on windows for ventilation remain.

3.1.2 Passive stack

Passive stack ventilation is a system of vertical, or near vertical, ducts which run from wet rooms, such as the kitchen and bathroom, to vents on the roof. The ducts extract moist air by utilising the stack effect [Stephen, 1989]. The stack effect is caused by differences in inside and outside dwelling temperature. As long as the temperature inside the dwelling remains higher than the outside temperature, the system will provide continuous ventilation. The ducts should offer minimum resistance to airflow. There should be no more than two bends per duct and each bend should not exceed 45° [Stephen, 1994]. Duct diameters are typically between 80 and 125mm [DoE, March 1996]. Fresh air enters the dwelling through trickle vents and open windows. A typical configuration is shown in Figure 3.4.

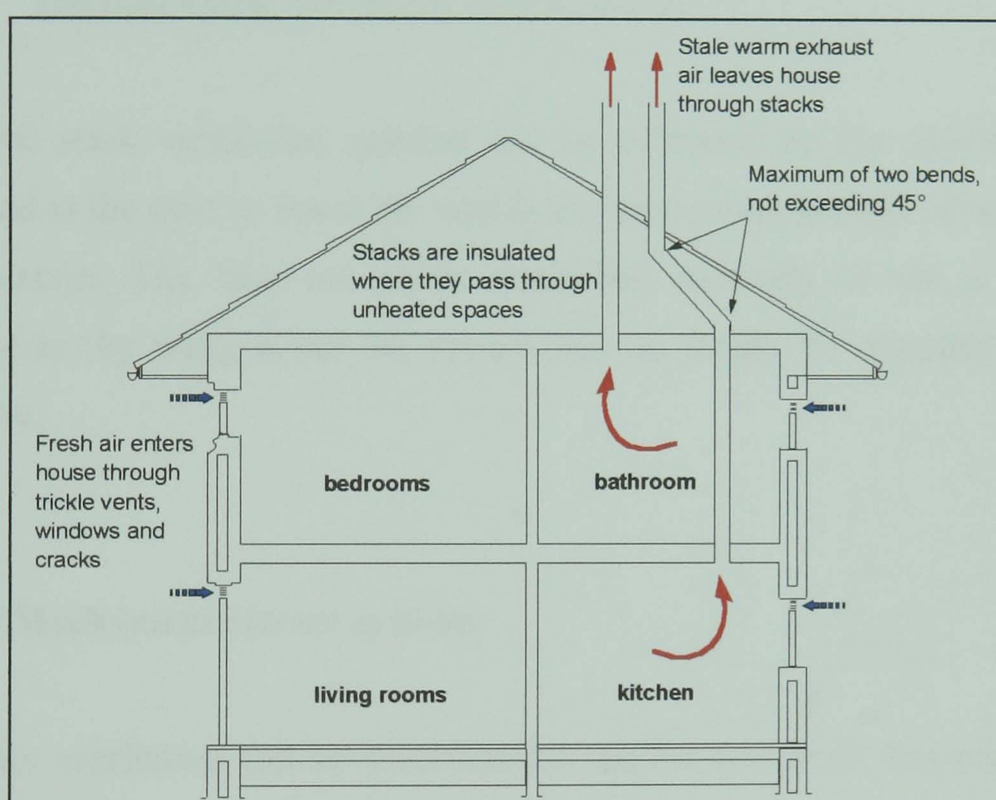


Figure 3.4: Passive Stack Ventilation

Passive stack systems are most suited for use where a consistent winter time driving force can be developed, such as in moderate to medium cold climates [Liddament 1996].

Advantages:

- Relatively inexpensive installation costs.
- Cheap running costs.
- Continuous passive ventilation.

- Continuous removal of moisture and contaminants from areas of high concentration.
- The systems are quiet and unobtrusive.

Disadvantages:

- Occasional back-draughting when the pressure generated in the stack cannot overcome the static pressure of cold outside air sitting above it.
- Heat is lost from vented rooms.
- Cold air that enters the house must be heated.
- The positioning of suitable vertical ducts.

Passive stack ventilation systems can be enhanced by the addition of a fan located in the duct to boost the ventilation rate during periods of high moisture production. The fans are either controlled manually or via a humidistat. However, by using a fan the system can no longer be regarded as entirely passive.

3.1.3 Mechanical extract systems

Extract ventilation can be a centralised ducted system or less complex local extract systems. Centralised ducted systems consist of a central fan that is connected to a number of grilles via a network of ducting. These systems tend to be used in small houses or apartments in cold climates [Liddament 1996]. Using these systems can prevent condensation on the buildings fabric. However, centralised extract systems are not common in the UK. The main form of extract ventilation in dwellings is local extract ventilation, and it is these systems which are covered here. This form of mechanical ventilation is widely used throughout the UK and most modern dwellings will have local extract fans installed.

Local extract systems are low capacity fans used in rooms with high moisture production, such as the bathroom and kitchen. They are usually propeller type fans mounted in a window or cooker hood and have typical capacities of between 25-50 l/s. They are located such that extraction occurs very near to the point of production. Their operation is normally intermittent and is often controlled by a timer switch, humidistat or they are linked with the light switch. As air is exhausted, replacement air is drawn in from other rooms which, in turn, is replaced by air drawn in through open windows, cracks and other openings. Figure 3.5 illustrates a local extract mechanical ventilation system.

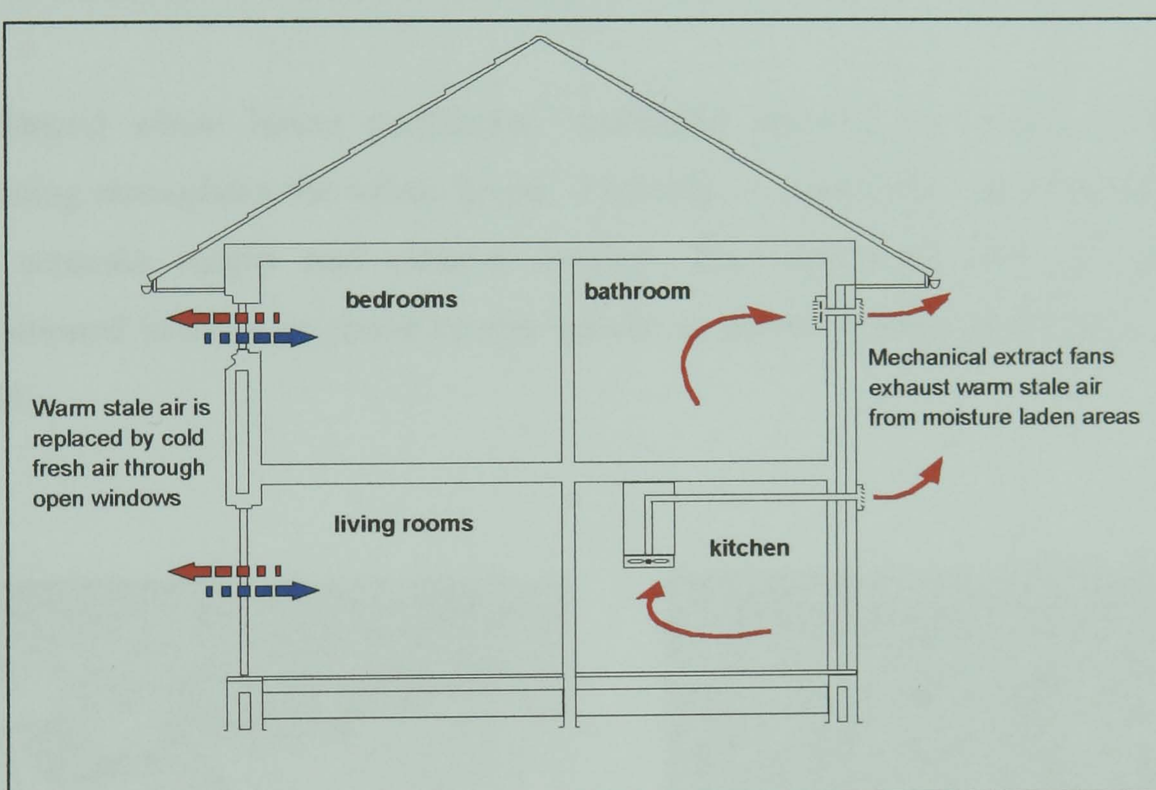


Figure 3.5: Local Extract Mechanical Ventilation

Advantages:

- Directly exhausts moisture and pollutants from source.
- Relatively inexpensive and easy to install.
- Good prevention of condensation.

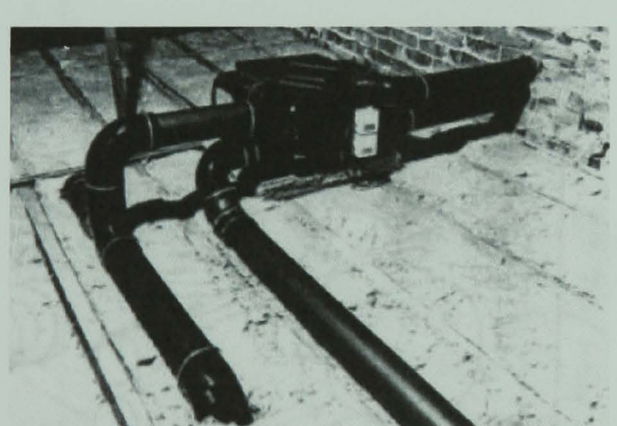
Disadvantages:

- Poor distribution of fresh air.
- No provision for direct make-up air.

- May require regular cleaning and maintenance.
- Can cause draughts.
- Negative pressures produced within the home can cause problems, especially in areas with radon gas present. The negative internal pressure would draw the gas into the house.
- Heat is lost with exhausted air
- Systems can be noisy.

3.1.4 Balanced whole house mechanical ventilation with heat recovery

Balanced whole house mechanical ventilation systems are ducted systems running throughout the whole house. Typically, a centralised unit is connected to separate supply and exhaust ducting. The centralised units are often positioned in attics or above cooker hoods, as shown in Photographs 3.1a and 3.1b.

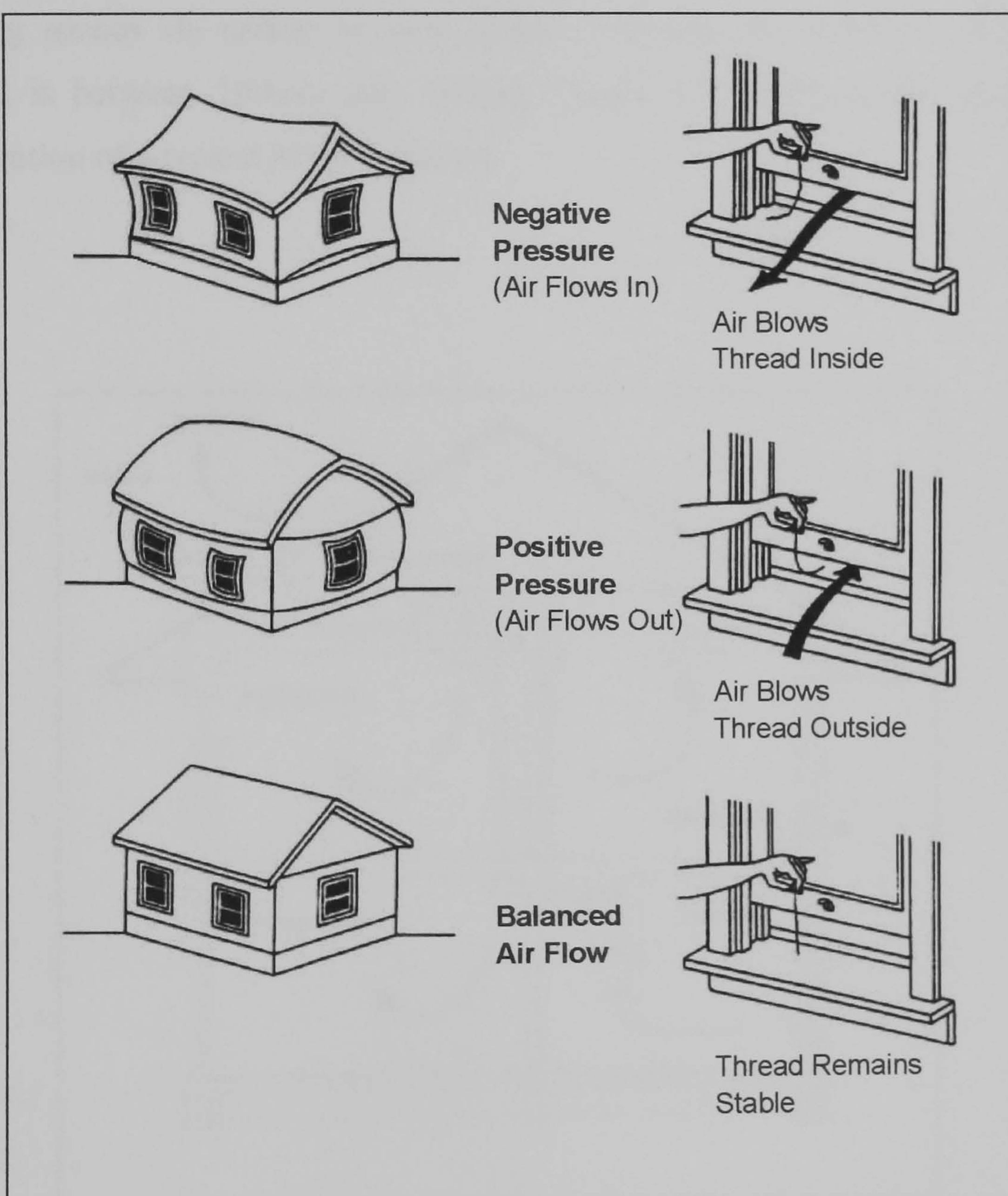


(a) Attic installation

(b) Cooker hood installation

Photographs 3.1a and 3.1b: MVHR System Installations

Systems are designed to exhaust warm stale air from polluted rooms with high moisture content, such as the kitchen and bathroom. At the same time, an equal quantity of fresh air is supplied to occupied rooms, such as living rooms and bedrooms. The system is balanced because the supply and exhaust air flow rates are the same. This prevents any pressure difference occurring between the inside and outside of the house. Figure 3.6 illustrates balanced and unbalanced systems. Systems can be balanced by using dampers. On a calm day, a thread held to an partially open window will show whether a system is balanced or not by its movement.



negative pressure: extract rate > supply rate

positive pressure: supply rate > extract rate

balanced system: supply rate = extract rate

Figure 3.6: Balanced and Unbalanced MVHR Systems [Ncat, 1984]

The majority of balanced whole house mechanical ventilation systems, incorporate two centrifugal fans and an air-to-air heat exchanger to recover heat from the exhaust air. The extract air from the dwelling is ducted through a heat exchanger and heat is recovered from the air. This air is then exhausted to the outside via a wall grille. Through another wall grille, fresh air is drawn from outside and is then passed through the same heat exchanger where it picks up heat from the extract air. This warmed fresh air is then supplied to living spaces via ceiling or wall grilles. Typically, the diameter of ducting used is between 100mm and 150mm. Figure 3.7 illustrates the layout and operation of a typical MVHR system.

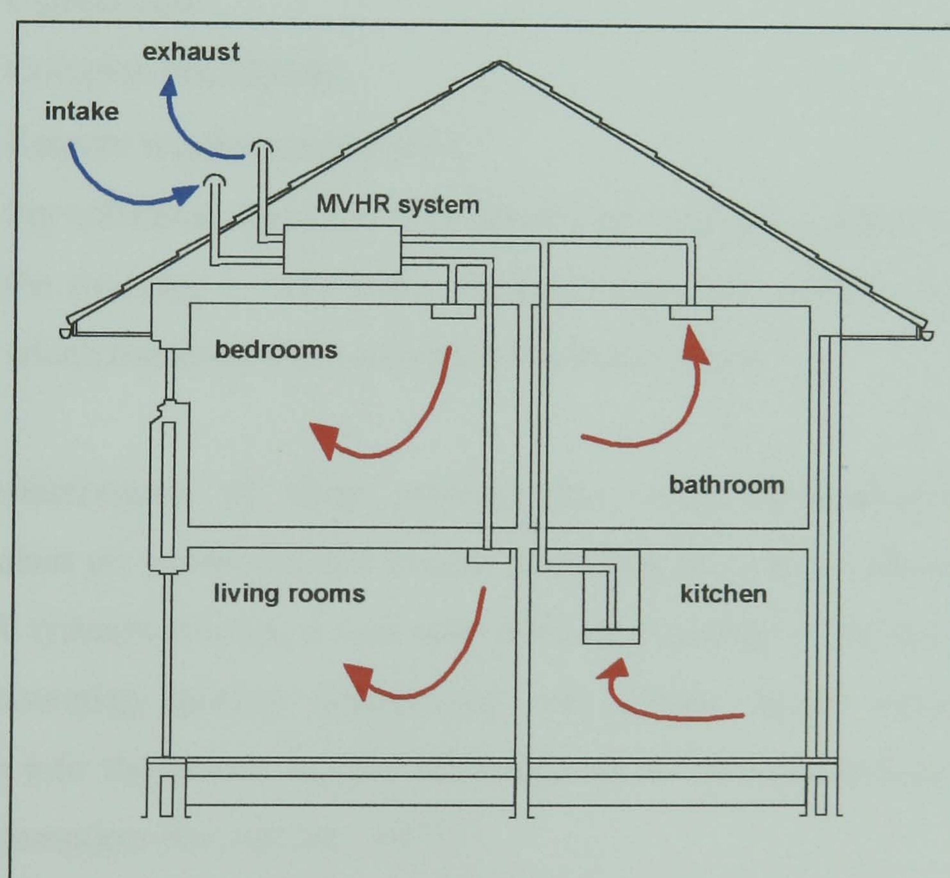


Figure 3.7: Balanced MVHR System

The systems often incorporate a boost function which can be used to increase the rate of ventilation during periods of high moisture production, such as washing and bathing. A large range of MVHR systems are manufactured, varying in size and performance. Section 3.4 covers a number of different heat exchanger types used in these systems

Advantages:

- Warmed supply air.
- No draughts.
- Exhaust heat is recovered.
- Reduction of pollutants and condensation.
- Control of humidity.
- Good control and distribution of air.
- Filtration of supply air is possible.
- Windows can remain closed giving greater security and reduction of external noise and dust pollution.

Disadvantages:

- Capital cost
- Complex installation
- Require regular maintenance
- For effective operation they should be installed in airtight dwellings. If the dwelling is very leaky the infiltration rate will lead to a heat loss which can exceed the savings from heat recovery.

The effectiveness of these systems has been demonstrated in studies undertaken on superinsulated houses in Milton Keynes. In those houses with MVHR systems present, it was noticeable that misting of mirrors after bathing and showering quickly disappeared, and tumble dryers could be vented directly into the house, to take advantage of the heat generated, with no sign of condensation [Ruyssevelt, 1987].

Another study, under the 'Best Practice Program' for the UK Energy Efficiency Office, demonstrated the benefits of installing MVHR systems in refurbished high rise dwellings [DoE 1995]. Previously, the dwellings had suffered from problems associated with damp. However, once the MVHR systems were installed these problems were eliminated. The reaction of the tenants to the MVHR systems was also found to be very favourable.

3.1.5 Whole house mechanical ventilation heat recovery heat pumps

Heat pumps can be used in whole house MVHR systems to recover heat and provide warm air heating. These systems often incorporate a plate heat exchanger in addition to the heat pump. In these systems, the heat pump is used to extract even more heat from the exhaust air after it has passed through the heat exchanger. This additional heat is used to further warm the supply air.

In heat pump systems, the supply air can be warm enough to heat an entire house. A well insulated home with an MVHR heat pump system, may only need a small supplementary heating system for extreme cold periods. Figure 3.8 illustrates how an MVHR heat pump system is used in a house.

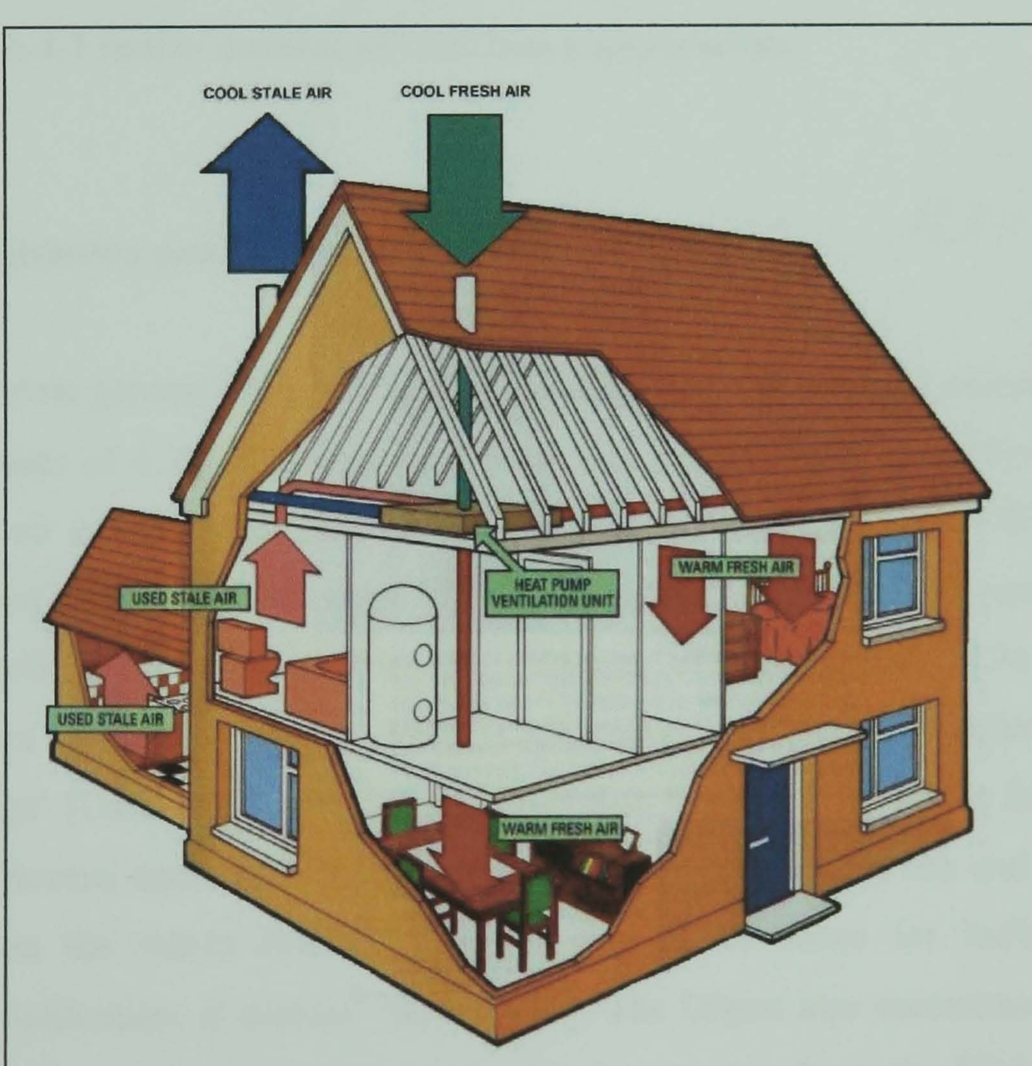


Figure 3.8: MVHR Heat Pump System [London Electricity]

By transferring or 'pumping' the heat from the warm extract air to the cold intake air, a heat pump can produce a lot of heat for very little fuel. Typically, heat pumps use a third of the electricity used by direct-acting panel heaters to give the same amount of heat [ECD]. Heat pump systems offer all the benefits associated with MVHR systems. In addition they provide warmth evenly distributed throughout the home and can reduce domestic energy consumption. The reduction in energy consumption benefits the environment by reducing carbon dioxide emissions.

A study was carried out on an MVHR heat pump system installed in a test house at the Electricity Council Research Centre. The results of the study concluded that the system was effective at reducing energy consumption. The system was shown to provide well distributed heating at a unit cost below that of off-peak electricity [McIntyre, Sept 1986, Nov 1986, 1989].

A full explanation of heat pump technology is covered in Chapter 5, and section 3.4.5 further explains MVHR heat pump systems.

3.2 Legislation covering MVHR systems

To maintain general indoor air quality the ASHRAE Standard 62 recommends a minimum of 0.35 air changes per hour (ach), i.e., in one hour 35% of the home's air is exhausted and replaced with outdoor air [ASHRAE 1989]. The UK Building Regulations cover MVHR systems used in dwellings separately. The Building Regulation Part F1, 'Means of Ventilation', 1.9d refers designers to BRE Digest 398, 'Continuous mechanical ventilation in dwellings' [DoE 1991]. The Digest states that 'the total extract air flow rate during normal operation should be equivalent to between 0.5ach and 0.7ach based on the whole dwelling volume, less an allowance for background natural infiltration, if desired' [BRE 1994]. The Digest also recommends that systems have a facility to boost the air extract rate from the kitchen and bathroom. It is suggested that the extract air flow rate is increased by 50% in a single room or by 25% for the whole system. It is also recommended that the supply and extract air flows should be nominally balanced, except when boost is operating.

3.3 Airtightness

The majority of existing housing has high levels of air infiltration ranging from an average of 0.7 ach to 1.5 ach [DoE 1996]. However, new build and refurbished properties can be built to high airtightness specifications. It is these dwellings that MVHR systems should be used in for cost effective operation. Figure 3.9 illustrates areas of a house through which air leakage can occur. For good airtightness these areas should be sealed adequately to avoid infiltration.

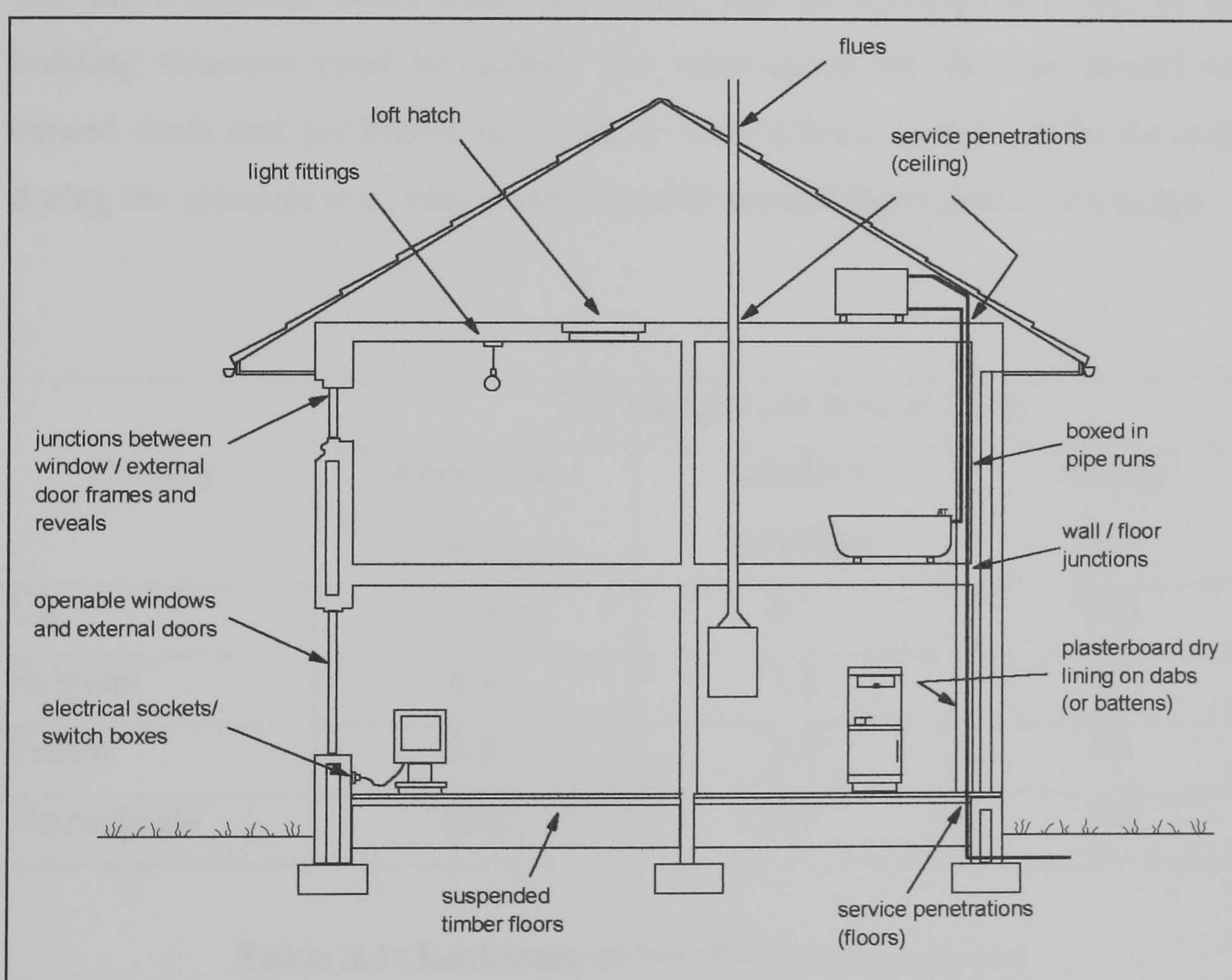


Figure 3.9: Air Leakage Routes

The current guidance on the design, installation and operation of MVHR systems states that dwellings should be as airtight as practicable for economic operation of an MVHR system. Currently, the practical limit is a mean background air infiltration rate of no more than about 0.2 ach [BRE, 1994].

An air pressure test can be used to measure airtightness. The fan pressurisation method is an accepted and convenient method of pressure testing buildings to determine airtightness. The principle is to pressurise (or depressurise) a building while measuring the air flow out (or in). The equipment used comprises of a calibrated variable speed fan, a meter to measure air flow through it and a manometer to measure the pressure difference between the inside and outside of the building. The fan is fitted in a panel sealed to an open external door. A pressure difference of 50Pa is maintained between the inside and outside of the building and the air leakage rate (ach) is measured. Before any measurements can be taken, all windows and other external doors must be closed, and all openings not part of the building structure must be sealed. The wind speed for the test should not exceed 4m/s and preferably be no more than 2.5m/s. Leaks can be detected during the pressure test using a smoke puffer around likely points of leakage.

Country	Air changes per hour at 50Pa		
	Arithmetic mean	Standard deviation	Sample size
UK	13.6	5.7	385
Belgium	8.2	7.2	57
France	3.6	2.0	66
Netherlands	10.1	6.7	303

Table 3.1: Leakiness of Dwellings [Woolliscroft, 1997]

Table 3.1 contains results of air leakage tests carried out on homes in the UK and other countries. The results show UK homes to be leakier than the other European countries covered. Infiltration of 2 ach at 50Pa is an acceptable maximum airtightness requirement for economic operation MVHR systems.

3.4 MVHR system types

The following section summarises the main types of different MVHR systems currently used.

3.4.1 Plate heat exchangers

The most common form of whole house MVHR systems incorporate a plate heat exchanger. Plate heat exchangers are static devices that allow heat transfer from one ducted air stream to another. The plate heat exchanger consists of a series of thin plates (metal, plastic or paper) separated by small gaps. The two air flows pass through adjacent gaps, separated only by one plate, through which heat transfer takes place by conduction. The heat exchanger is normally configured to give-cross flow operation, i.e. the supply and exhaust air flows are at 90° to one another. Figure 3.10 illustrates the operation of a cross-flow plate heat exchanger.

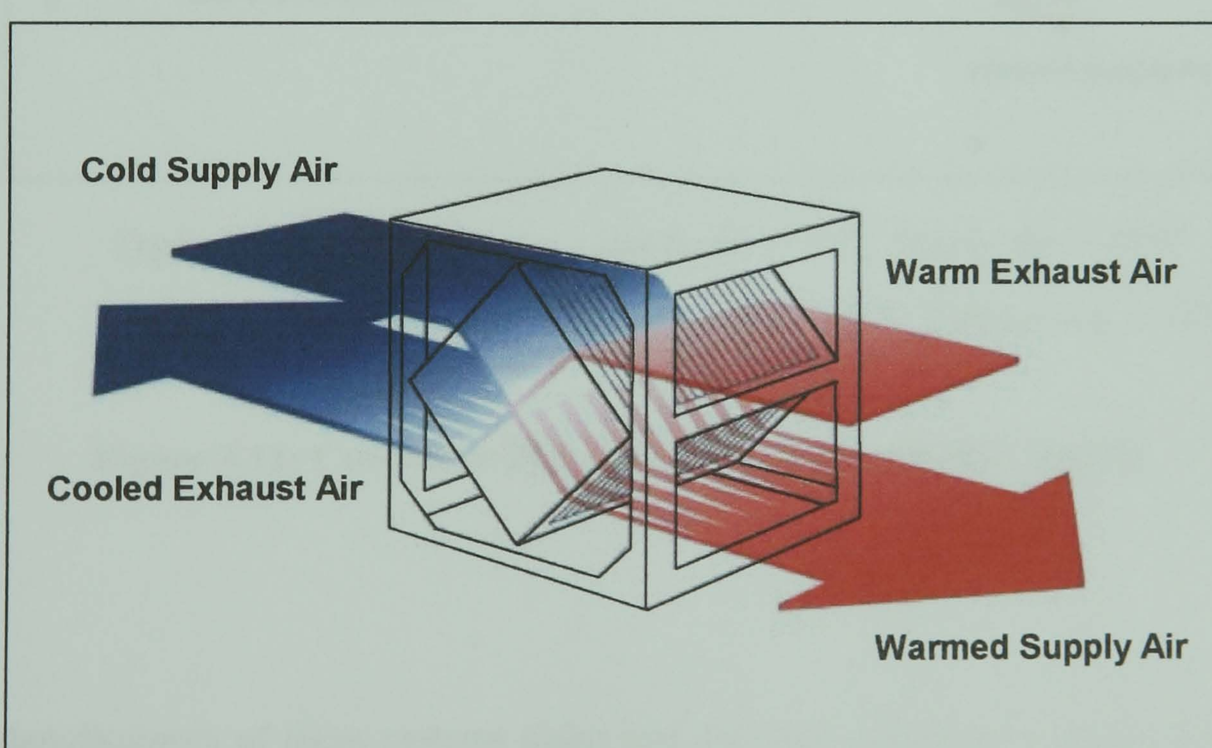
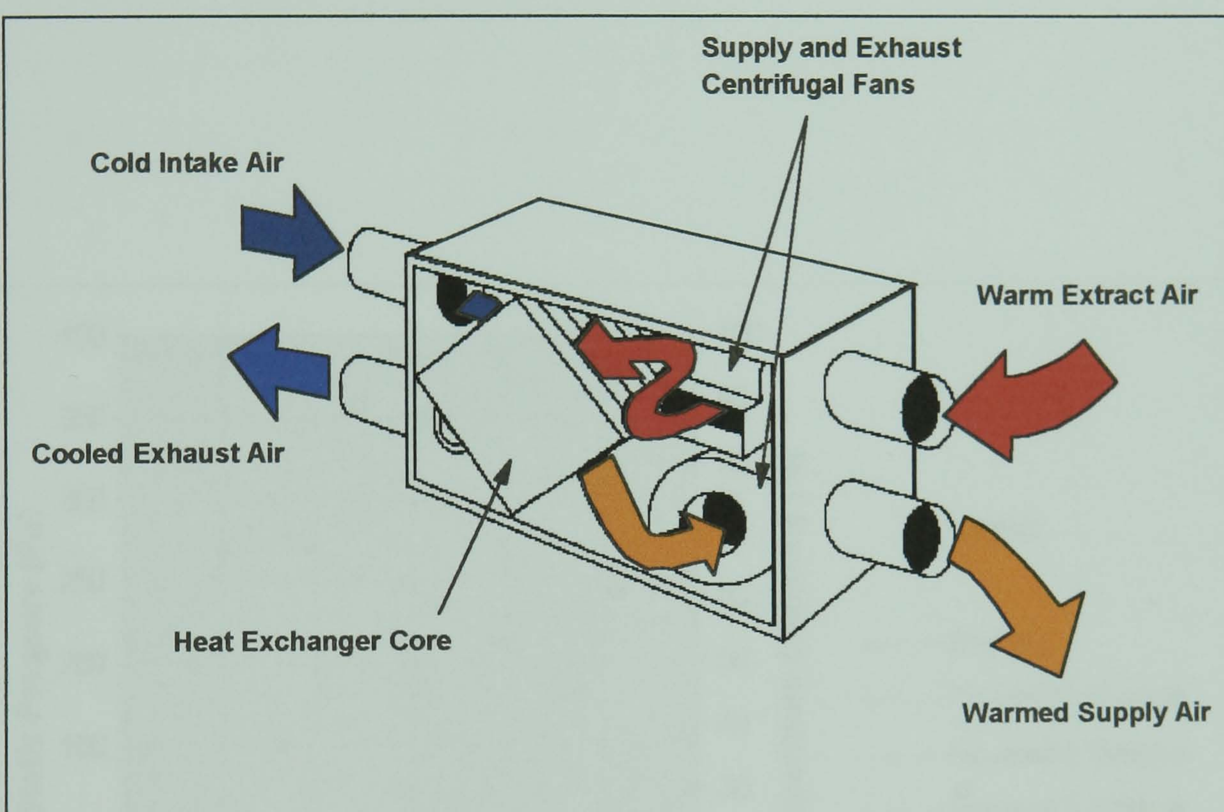


Figure 3.10: Operation of Cross-Flow Plate Heat Exchanger

Typically, units are packaged in an enclosed box with two centrifugal fans and, in some cases, air filters are also present to remove particulates. To prevent overheating a dwelling in warm weather, a by pass damper can be fitted for summer time operation. Figure 3.11 illustrates a typical cross-flow plate heat exchanger MVHR system. Also shown are typical temperatures for the four air streams during operation.



Typical air temperatures: *Intake Air = 6 °C, Supply Air = 15 °C*

Extract Air = 20 °C, Exhaust Air = 11 °C

Figure 3.11: Cross-flow Plate Heat Exchanger MVHR System

Manufacturers of these systems claim heat recovery efficiencies ranging from 40% to 80%. Figure 3.12 illustrates performance curves, published in sales literature, for an MVHR system manufactured by the Norwegian manufacturer, Villovent.

The graphs illustrate that increases in flow rate result in reductions in static pressure and efficiency. It should also be noted that the system is not balanced. The extract rate is greater than the supply rate. Because of this the heat exchanger effectiveness will be less than the heat exchanger efficiency (see section 3.6 for an explanation of efficiency and effectiveness). Laboratory tests on a commercially available plate heat exchanger system have been carried out and the results are given in Chapter 7.

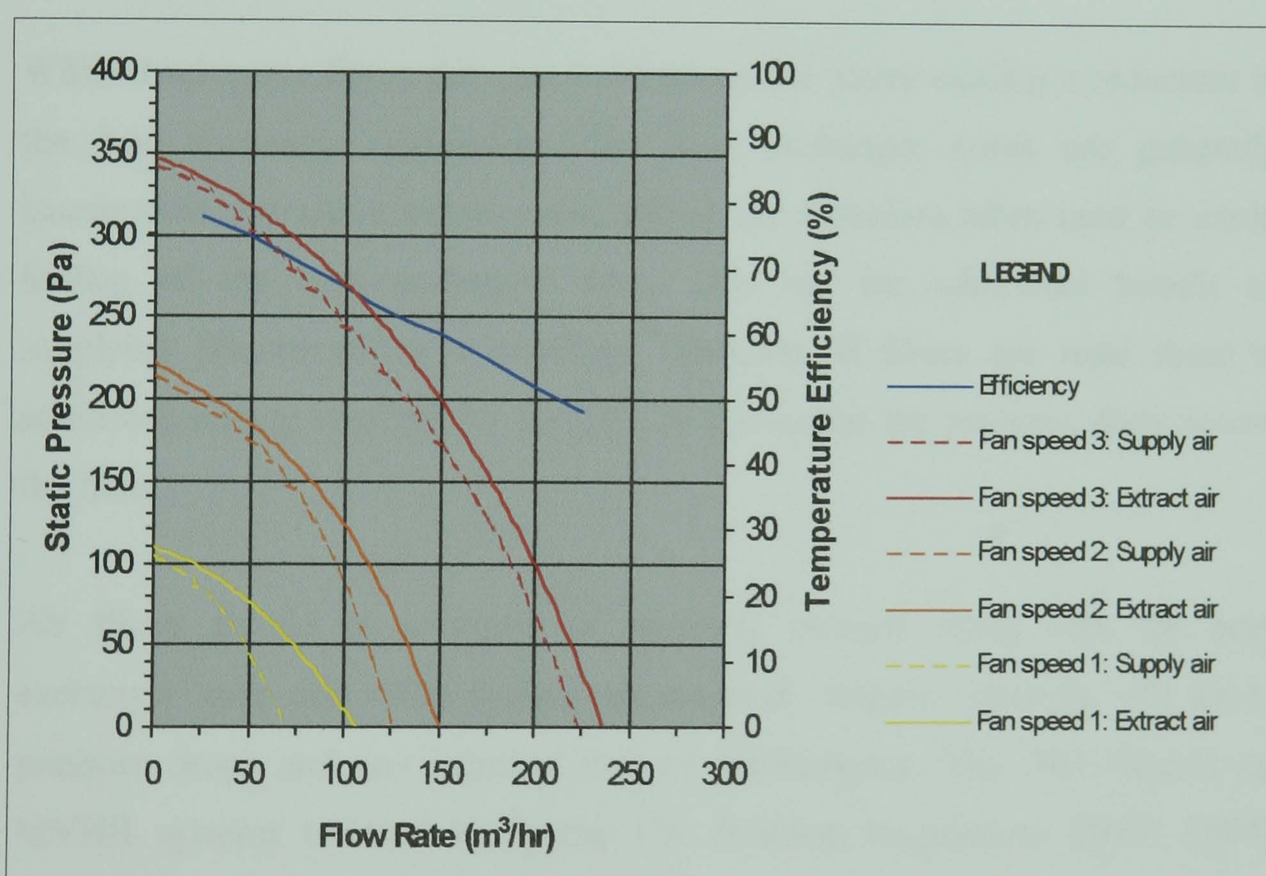


Figure 3.12: Performance Curves For The Villovent VVX-200 MVHR Plate Heat Exchanger System [Villovent]

Because plate heat exchanger systems tend to be modular, the appropriate size or number of modules can be selected for the appropriate flow rates. In addition, the number and spacing of plates are chosen to suit particular applications. Heat transfer coefficients can be enhanced by promoting turbulence when using plates that are corrugated or that have a surface roughness. This, however, results in increased resistance to air flow across the plates and, as a result, fans will consume more energy. Heat exchanger cores of this type are also more prone to fouling. To avoid these problems, some manufacturers chose to design units with laminar air flow through the heat exchanger cores. However, this can inhibit heat transfer due to the increase of boundary layer thickness. Designers therefore have to achieve the right balance between running costs and heat exchanger efficiency.

Without adequate filters dirt can build up on the plates causing a reduction in the heat exchanger efficiency. The heat exchanger cores are generally inaccessible and awkward to clean. Filters are therefore often used to avoid fouling of the heat exchanger core. This has the additional benefit of supplying filtered air to a dwelling. However, if filters are used there is additional energy required by the fans to overcome the pressure drop across the filter.

Air filters should be changed or regularly cleaned along with the heat exchanger core and other system components. Regular cleaning will avoid pressure drops and any incurred loss of performance. The BRE Digest on MVHR systems referred to by the UK Building Regulations [BRE 1994] states that: 'Air filters and air supply and extract grilles will probably need to be cleaned at least two or three times a year; the heat exchanger in MVHR systems annually. Fan impellers can be inspected, and cleaned if necessary, when the heat exchanger is cleaned. The inside of ductwork rarely needs cleaning.'

MVHR systems are usually positioned out of the way in inaccessible places such as the attic. A dwelling's occupants may be reluctant or liable to forget about cleaning a system that is out of view and taken for granted as functioning properly. Because of this, it is not possible to fully ensure that they will be properly maintained. It is therefore necessary for systems to be as maintenance free as possible.

Advantages:

- They are reasonably simple devices with no moving parts.
- Modular construction and ability to alter plate design allows for different specifications for a wide range of applications.
- If properly constructed there is very little chance of cross-contamination of air streams.

Disadvantages:

- Fan energy required to overcome pressure drop across the heat exchanger core.
- Sensible heat recovery only, unless paper core heat exchangers are used which permit the transfer of moisture from the exhaust to the supply air. Or if condensation occurs on the warmer side of the plates. (See section 3.6 for definitions of sensible and latent heat recovery).
- Overheating may be a problem in the summer if no bypass is provided.
- They can only be used for applications where the supply and exhaust ducts can be brought together.
- Any condensation on the exhaust side could be prone to freezing when used in colder climates.
- Without adequate filters dirt builds up on plates causing a reduction in heat exchanger performance and a drop in air flow rates. Regular cleaning of the heat exchanger core is therefore a requirement.
- If filters are used there will be an increase in fan power consumption. Any filters will also need cleaning or changing regularly.

3.4.2 Thermal wheels

The thermal wheel, also known as a heat wheel, rotary regenerator or rotary heat exchanger, is a porous wheel through which the exhaust and supply air streams flow. The wheel is divided into a number of segments packed with a coarsely knitted metal mesh, or some other material. As the wheel rotates it is alternately heated and cooled by the air streams. The wheels generally operate by rotating at between 10 and 20 revolutions per minute [Irving, 1994]. The heat picked up from the warm exhaust air is discharged into the cooler supply air. If latent heat of transfer is also desired the wheel material is treated with a hygroscopic substance [Zarling, 1982]. Where it is essential to restrict cross contamination, a purge section is included. The supply passing through the section of the wheel element closest to the exhaust air zone is purged by re-directing it to the exhaust air. However, the use of a purge section reduces the heat recovery efficiency. Figure 3.13 illustrates the operation of a thermal wheel.

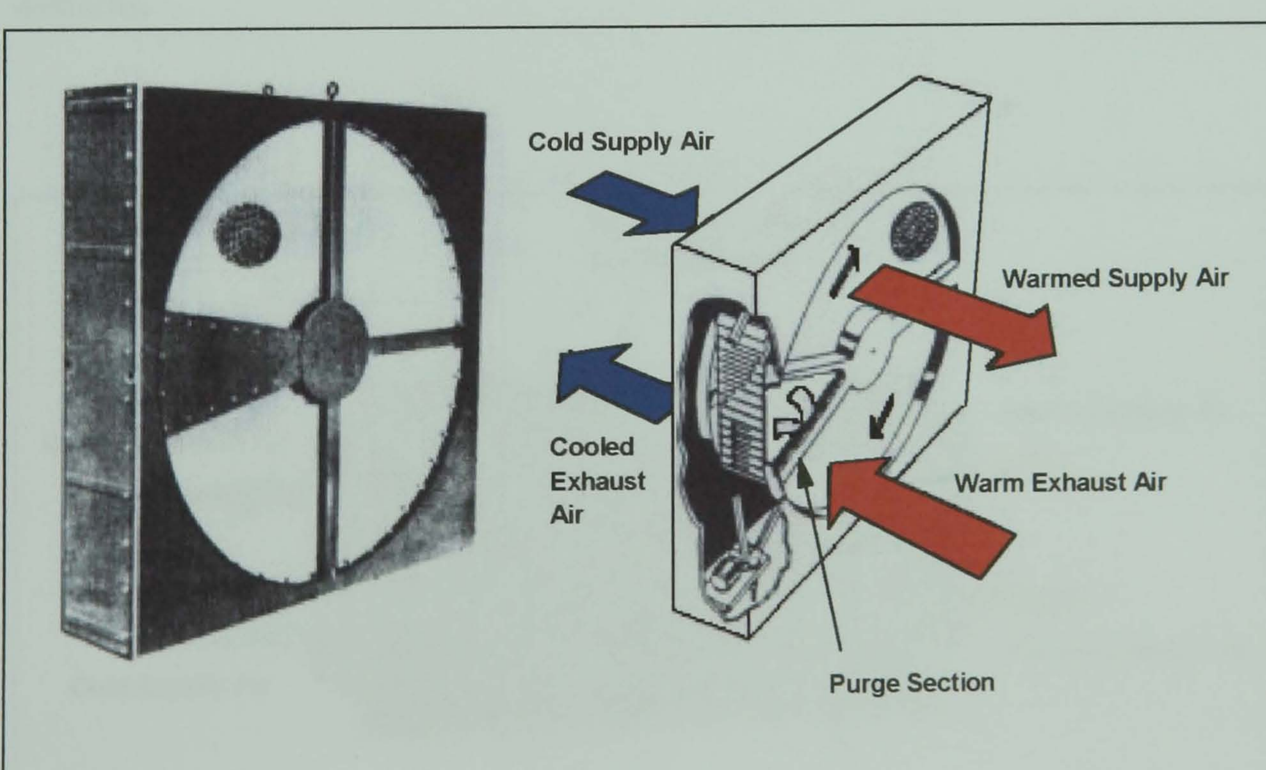


Figure 3.13: Operation of Thermal Wheel [CIBSE, 1995]

The density and formation of the matrix can be altered to suit different applications. For example, where there is dirty air the matrix density will be decreased to reduce deposition. Higher matrix densities result in higher heat exchanger efficiencies and greater fan energy requirement to overcome the pressure drop across the matrix.

Because the wheel matrix is difficult to clean, air filters are often used to prevent fouling. As with plate heat exchanger systems, regular maintenance of these filters is required. Again, due to the hidden location of these systems, regular maintenance cannot always be relied upon.

Thermal wheel heat recovery systems are mainly used in large commercial or public buildings where they form an integral part of the heating ventilation and air conditioning (HVAC) system. However, some domestic systems are manufactured which generally consist of a thermal wheel, supply and exhaust fans and filters packaged into a single unit. A typical domestic thermal wheel MVHR system is shown in Figure 3.14. Thermal wheel diameters range from around 300mm in small domestic units, to several metres in large commercial systems.

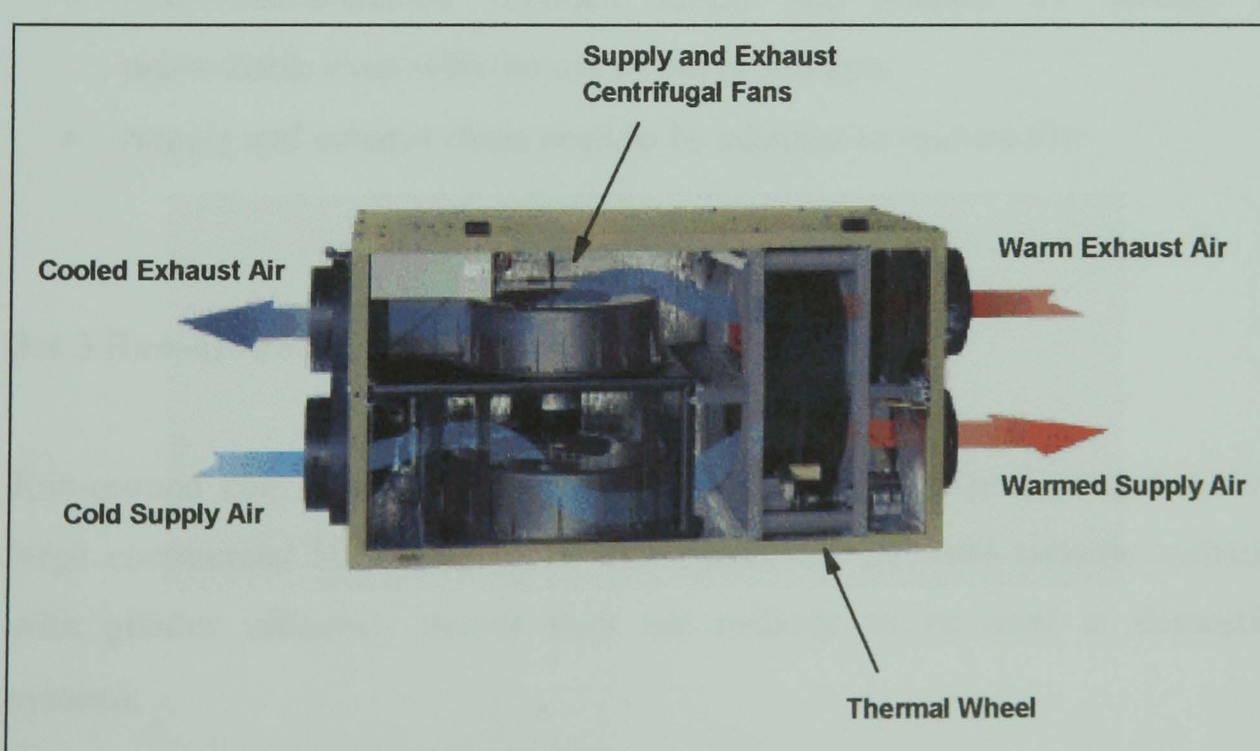


Figure 3.14: Thermal Wheel MVHR System [Oikos, 1999]

Thermal wheel MVHR systems typically have heat recovery efficiencies ranging between 60% and 85%. They have the highest efficiency of all devices currently in use.

Advantages:

- High heat transfer efficiency.
- They can transfer latent heat as well as sensible heat depending upon the material used for the thermal wheel.
- The thermal wheel design, matrix material and density can be varied to suit a wide range of applications.
- Static pressure drops across the systems tend to be low resulting in low fan energy requirement.
- The efficiencies of the systems can be varied by using a variable speed drive.

Disadvantages:

- Small amount of additional energy required to rotate the wheel.
- The matrix of a thermal wheel is difficult to clean and so it is important to regularly maintain the air filters.
- Cross-contamination between supply and exhaust air streams is unavoidable even with the use of purge sections.
- Supply and exhaust ducts need to be adjacent to one another.

3.4.3 Run-around coils

Run-around coil systems are generally used in industrial process systems or large commercial HVAC systems. The availability of more suitable systems with greater efficiency means they are unlikely to be used in domestic systems.

Run-around coils comprise of two fin-type heat exchangers connected in series with a pump. One heat exchanger is positioned in the supply duct and the other in the exhaust duct. A heat transfer fluid (normally a water/glycol solution) is pumped around the system to transfer heat from the exhaust air to the supply air. The fluid picks up heat when passing through the coil positioned in the exhaust duct. This heat is then given up to the air in the supply duct. The process is continuous. Figure 3.15 illustrates the operation of a run-around coil system and Photograph 3.2 shows an example of a system.

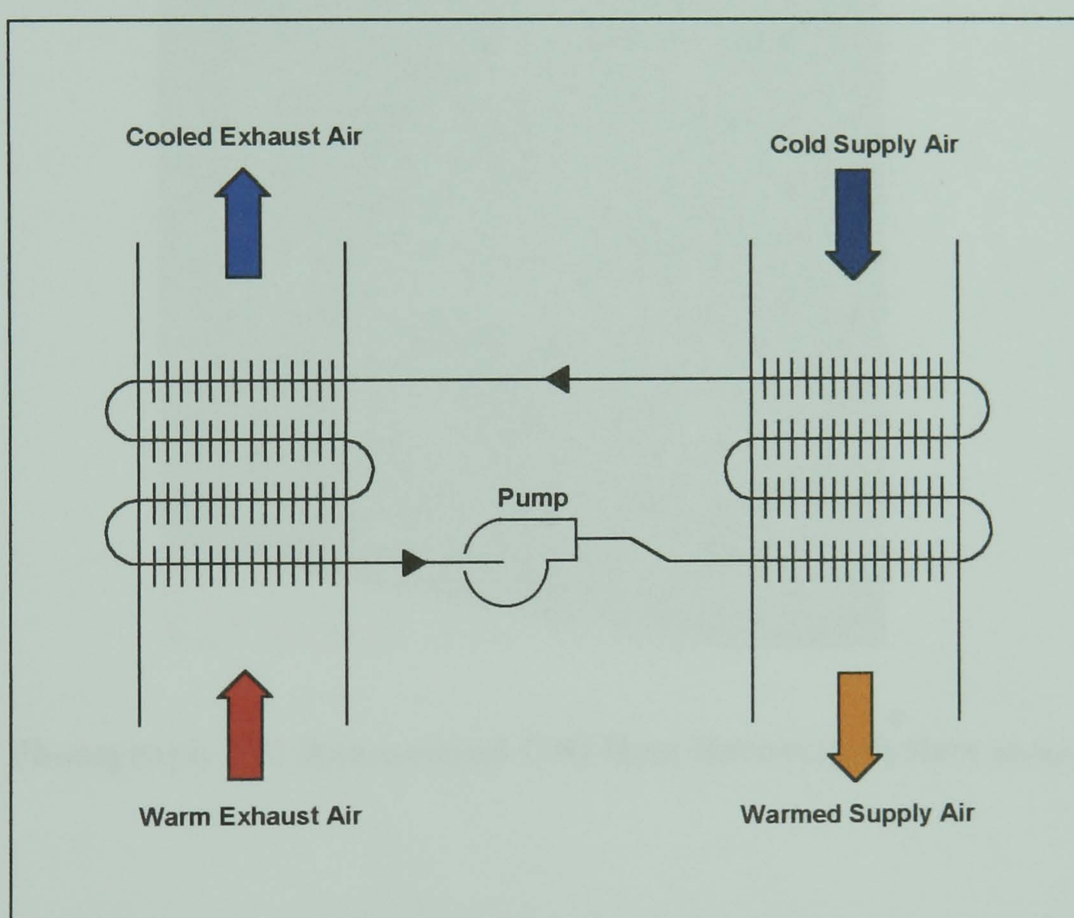
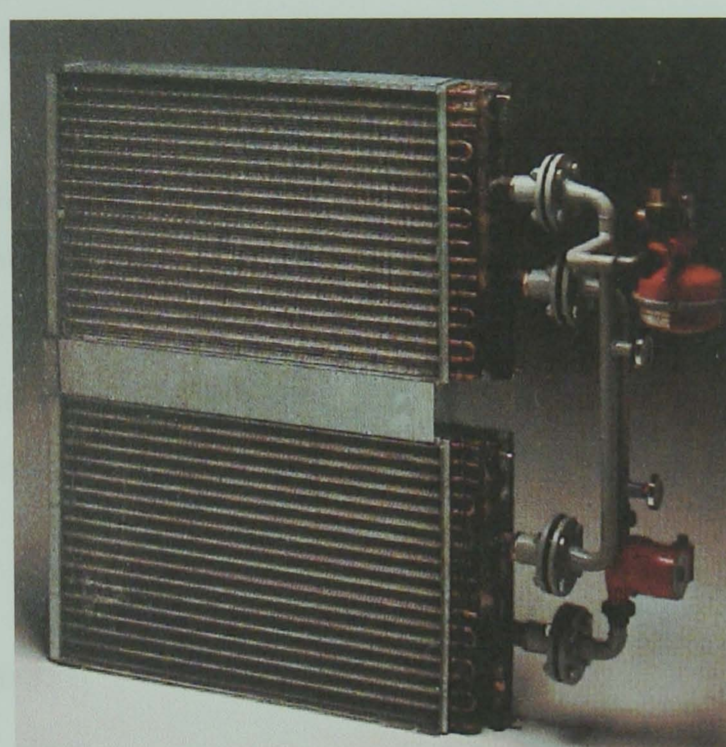


Figure 3.15: Operation of Run-around Coil Heat Recovery System

The number of coils used can be varied depending on the rate of heat transfer required. However, the energy consumed by fans will increase to overcome the increased pressure drop associated with increases in the number of coils.

Glycol solution is often used as the working fluid to prevent freezing. The concentration of the glycol should only be strong enough to prevent freezing during the coldest temperatures the system will experience. Too much glycol will increase the pressure drop around the loop and hence the energy required by the pump. Glycol also reduces the heat transfer efficiency. Typically, a glycol mix of 20% will prevent freezing down to –10 °C but it increases the loop pressure drop by 15% and reduces the heat transfer efficiency by 10-20% [Liddament, 1996].



Photograph 3.2: Run-around Coil Heat Recovery System [Bahco]

Typical heat recovery efficiencies for run-around coil systems range between 40% and 60% [Pita, 1998].

Advantages:

- Run-around coil systems are particularly suited to applications where the supply and exhaust ducts are not adjacent to one another.
- There is no risk of cross-contamination as the supply and exhaust ducts are totally separated. This makes them suitable for applications where any recirculation is prohibited due to contaminants in the exhaust air.

- The number of coils and fin spacing can be designed to suit individual applications.
- A single system can service a number of supply and exhaust ducts.

Disadvantages:

- Energy used by the circulation pump.
- Maintenance of the circulation pump.
- Can only transfer sensible heat, except when condensation occurs on the coil in the exhaust air stream.
- Have relatively low heat exchanger efficiencies.
- Working fluids prone to freezing. This is prevented by the use of anti-freeze which has the negative effect of reducing heat exchanger efficiency and increasing the pump energy required.
- To avoid build up of dirt on the coils filters are recommended. Filters require maintaining and will increase the energy consumed by fans.

3.4.4 Heat pipes

Heat pipes are simple devices that consist of a sealed tube containing a fluid with a high enthalpy of evaporation. When the fluid is heated at one end of the sealed tube it evaporates, flows through the centre of the tube, condenses at the other end of the tube and then passes back along an annular wick by capillary action to resume the cycle. The continuous cycle transfers heat from one end of the pipe to the other. Several finned heat pipes can be used together to form a heat pipe heat exchanger. This is installed with one half in the exhaust air duct and the other half in the adjacent supply air duct. The supply and exhaust air streams are normally in counterflow, separated by a baffle, normal to the heat pipes. Typical heat recovery efficiencies for these systems range between 45% and 65% [Irving, 1994]. Figure 3.16 illustrates a typical heat pipe heat exchanger.

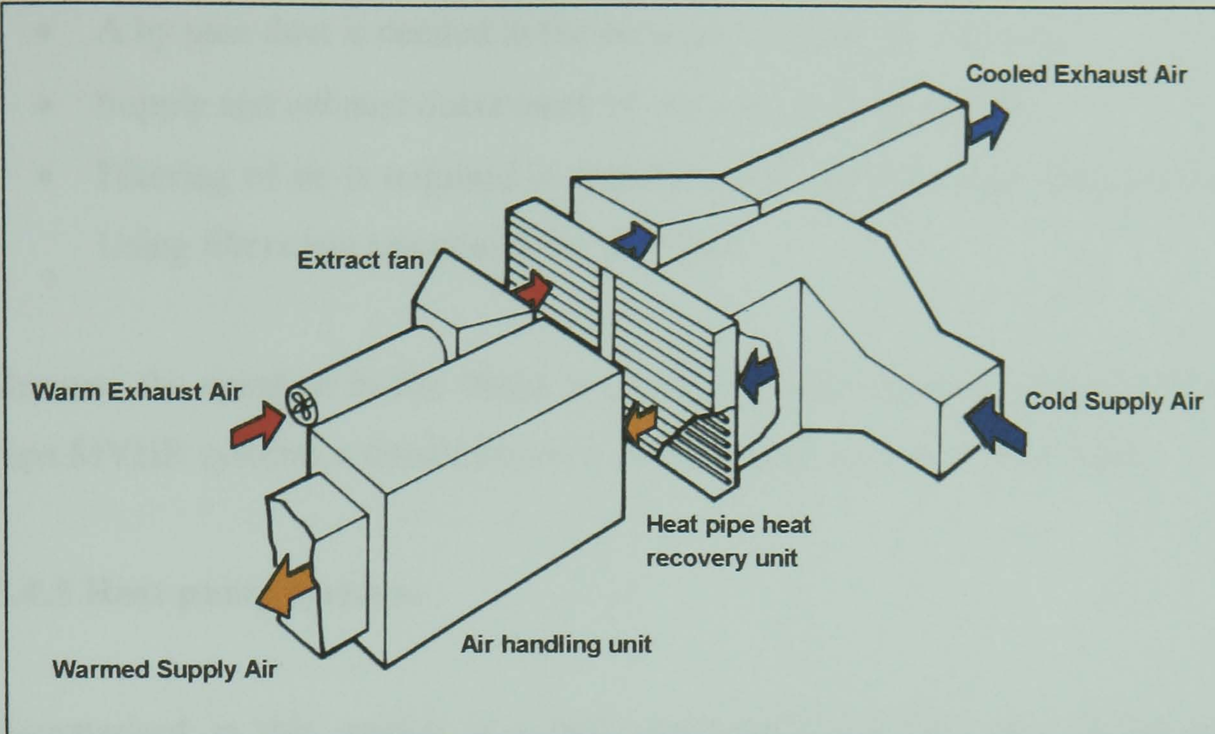


Figure 3.16: Typical Heat Pipe Heat Exchanger

Heat pipe heat exchangers are very compact compared to other types of heat exchanger. In a comparison of heat exchanger types Klaschka (1979) lists the compactness value for heat pipe heat exchangers as 7200W/m^3 °C. The values for plate heat exchangers and thermal wheels were given as 4140 W/m^3 °C and 5400 W/m^3 °C respectively. Heat pipes and the fins can be constructed from different materials to suit a wide range of applications. Also, the quantity and type of refrigerant can be altered depending on the application.

Advantages:

- Heat pipes are robust, compact and have no moving parts.
- The designs can be varied to suit a particular application.
- Relatively high rate of heat transfer.
- No possibility of cross-contamination of air streams.
- Reliable systems. Even if one individual heat pipe were to fail the heat exchanger would still be operational.

Disadvantages:

- The units tend to have high capital cost relative to other systems.
- They will only recover sensible heat unless condensation occurs on the heat pipe surfaces in the warm air duct and is removed as liquid.

- A by pass duct is needed in the summer to avoid overheating.
- Supply and exhaust ducts must be adjacent to one another.
- Filtering of air is required to avoid loss of heat exchanger performance. Using filters has associated fan penalties.

Because the research in this thesis is concerned with the development of heat pipe MVHR systems a detailed review of heat pipes is given in Chapter 4.

3.4.5 Heat pump systems

Summarised in this section is a basic explanation of how heat pump heat recovery systems operate. A full review and explanation of heat pumps is given in Chapter 5.

In the simplest form, an MVHR heat pump system has a finned evaporator coil positioned in the exhaust air duct to extract heat, and a finned condenser coil situated in the supply air duct to heat the air. In these systems, heat pumping is usually achieved through a vapour compression cycle (see Chapter 5 for heat pump cycles). Figure 3.17 shows a schematic of a heat pump heat recovery system.

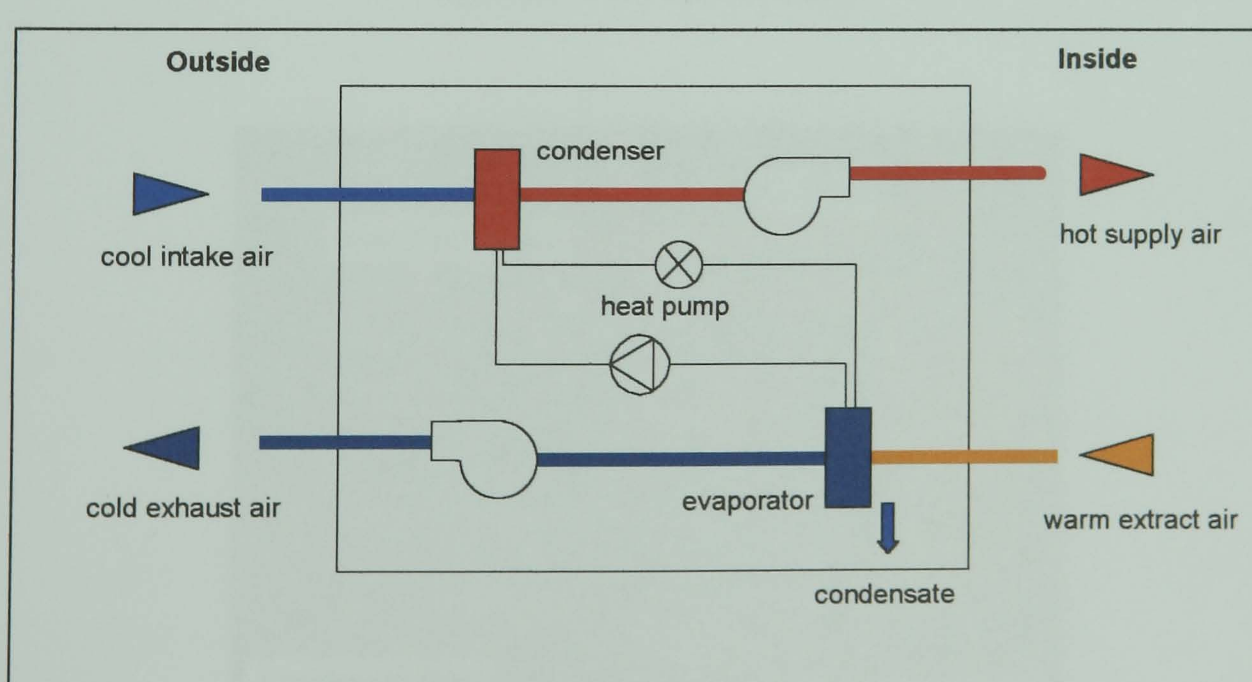


Figure 3.17: Heat Pump Heat Recovery

Heat pumps can be incorporated into plate heat exchanger heat recovery systems to boost the overall heat recovery performance. In these combined systems, the heat pump evaporator is placed in the exhaust duct downstream of the plate heat exchanger to extract even more heat. Further heating of the supply air is provided by the condenser, which is positioned in the supply duct downstream of the plate heat exchanger. Figure 3.18 illustrates this type of combined system and Photograph 3.3 shows a typical example of an MVHR heat pump system installed in the loft of a house.

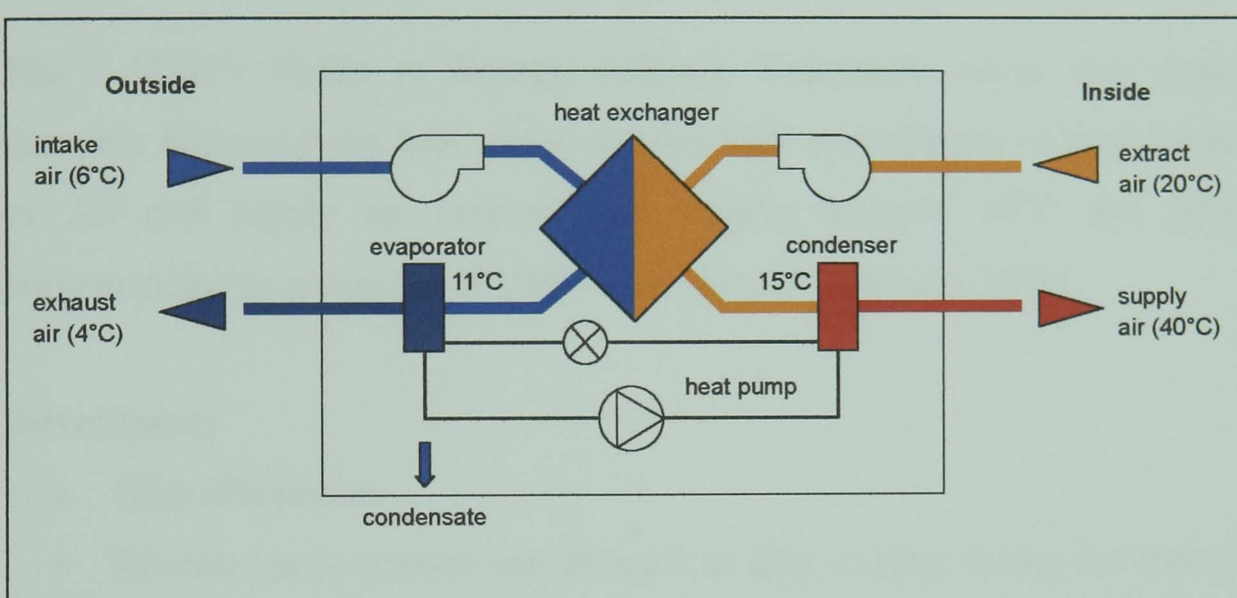
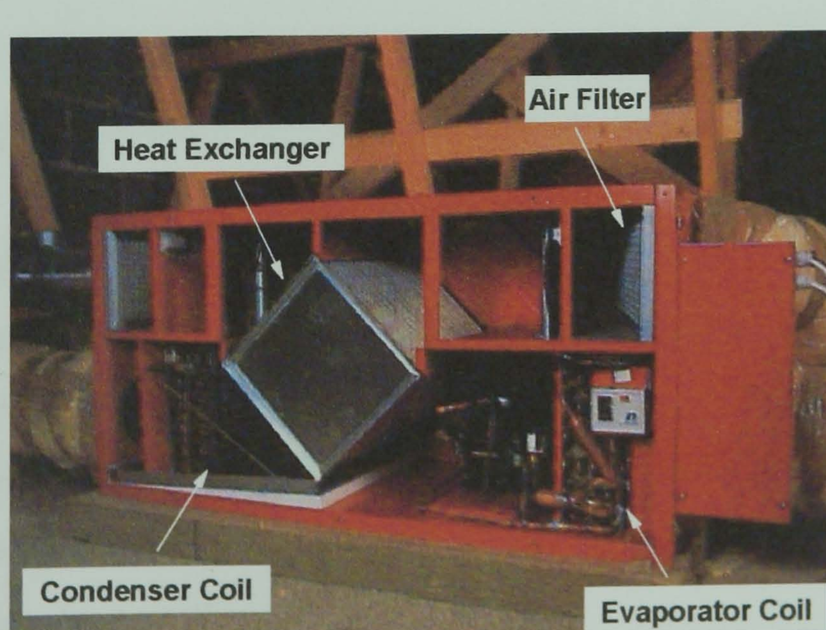


Figure 3.18: Plate Heat Exchanger Heat Recovery System with Heat Pump



Photograph 3.3: Domestic MVHR System Installation

The evaporators in heat pump systems often work at temperatures below the dew point of the exhaust air. Because of this, the water in the air condenses which may cause frosting on the evaporator coil. Too much frost will significantly reduce the performance of the system and therefore defrost cycles are used to remove the frost. In a defrost cycle the frost is melted using heating elements or by reversing the heat pump cycle, i.e. the evaporator becomes the condenser and vice versa. Defrosting uses more energy and interferes with the system's normal operation. Frosting of the evaporator coil is therefore best avoided or kept to a minimum.

The AIVC's Guide to Energy Efficient Ventilation states that typical domestic heat recovery heat pump systems have a coefficient of performance of 3.0 and supply air temperatures ranging between 30°C and 50°C, corresponding to an output of 1000Wh or more [Liddament, 1996].

Advantages:

- High efficiencies.
- Reverse cycle systems can be used to give cooling during the summer months.
- Heat pump systems are very good at extracting latent heat from the exhaust air as they often take the temperature of this air below the dew point.
- Warm air is evenly distributed throughout the home.
- Energy consumption and carbon dioxide emissions are greatly reduced.

Disadvantages:

- Capital cost (however, this will be paid back over a period through the cost of energy saved).
- Higher levels of maintenance due to more moving parts than most MVHR systems.
- Additional costs of defrost cycle.
- Filtering of air is required to avoid loss of heat exchanger performance.
- Using filters has associated fan penalties.

3.5 Review of Domestic MVHR System Available in the UK

The following table summarises the features and performance of a range of domestic MVHR systems on sale in the UK (in April 1999). The table covers those systems designed for medium to large sized dwellings, i.e. flow rates of 200 – 300 m^3/hr.

Type of System	Manufacturer* / Model Ref.	Flow Rate (m^3/hr)	Power (watts)	Thermal Efficiency (%)	Size (mm)	Weight (kg)	Cost** (£)
Cross-flow Plate Heat Exchanger	Baxi Air Management WH 300	up to 300	190	up to 70	550 595 280	13	340
	Johnson & Starley HR 100	up to 200	170	60-70	585 585 250	21	519
	Mitsubishi Electric LGH, 25RS2-E	up to 250	121	up to 70	780 735 275	23	715
	Villovent VVX200	up to 240	150	47-80	600 600 280	22	495
	Vent – Axia HR250	up to 220	185	up to 70	560 610 290	11.5	400
	ABB (UK) Akor system	up to 250	-	up to 70	-	-	500
Thermal Wheel	VENMAR AVS DUO, 1.9	193-300	250	64-77	420 435 891	31	1075

*See Appendix A for Manufacturers details.

**Costs are based on UK April 1999 prices for supply of the basic units. Installation costs and charges for control systems, ducting, grilles etc. are additional to these figures.

Table 3.2: MVHR System Specifications

The range and average specification of the residential MVHR systems shown in Table 3.3 are given in the following table

	Flow Rate (m^3/hr)	Power (watts)	Thermal Efficiency (%)	Size (cm^3)	Weight (kg)	Cost (£)
Range	0-300	121-250	47-80	85556-162786	11.5 - 31	340 – 1075
Average Value	Average max 250	178	average max 72	116249	20.25	578

Table 3.3: Range & Average Specification of Residential MVHR Systems

Cross-flow plate heat exchanger systems, as described in section 3.4.1, are at present the most common MVHR systems being sold for residential applications in the UK. The UK market at present is entirely dominated by this type of MVHR system. Other types of systems tend to be used for larger and more specialised applications. Very few thermal wheel systems are available for residential use. The thermal wheel system shown in Table 3.1 is by far the most expensive system shown. It is also the heaviest, largest and most power consuming of all the systems reviewed.

All the systems on the market claim heat recovery efficiencies up to 70% and some systems claim to be even better. The systems tend to be fairly compact and light enough to be handled by one or two persons.

For any new device to be acceptable and competitive in the UK market, it must fit within the parameters shown in Table 3.2, unless it has other merits that current devices do not exhibit.

3.6 Theory

The following sections cover the theory relating to the performance and operation of heat recovery systems. Heat pump theory is covered separately in Chapter 5.

3.6.1 Sensible and latent heat recovery

The recovery of waste heat from dry air is sensible heat recovery and the recovery of heat released by the condensation of vapour (usually water vapour) is latent heat recovery.

Sensible heat recovery devices do not transfer moisture. With these devices no moisture can be transferred, but latent heat can be transferred if the warmer air stream is cooled below its dew point and condensation occurs. Total energy recovery devices transfer both moisture and heat between the air streams.

3.6.2 Calculation of device performance

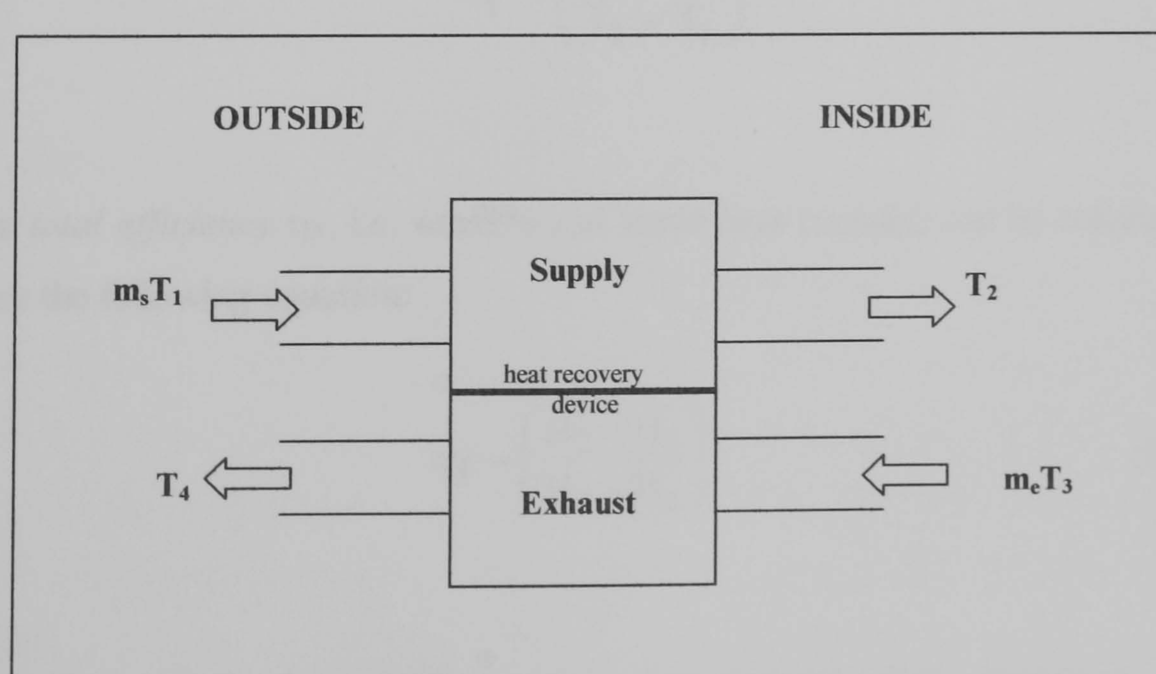


Figure 3.19: Heat Exchanger Terminology

Notation:

T = dry-bulb temperature (K)
H = Specific enthalpy (kJ/kg)

At positions: 1 = Intake air
2 = Supply air
3 = Extract air
4 = Exhaust air

m_s = Supply mass flow rate (kg/s)
m_e = Exhaust mass flow rate (kg/s)
C = Specific heat capacity of air at constant pressure (J/kgK)

Temperature Efficiency, η_t:

For sensible heat recovery, with the supply and exhaust flow rates equal, the performance of heat exchangers is normally quoted as the *temperature efficiency,* η_t. This can be calculated from the following equation:

$$\eta_t = \left(\frac{T_2 - T_1}{T_3 - T_1} \right) \tag{3.1}$$

The *total efficiency* η_T, i.e. sensible and latent heat transfer, can be calculated from the following equation:

$$\eta_T = \left(\frac{H_2 - H_1}{H_3 - H_1} \right) \tag{3.2}$$

The efficiency of a heat recovery system is a function of the flow rate. The efficiency will rise as the flow rate through the heat exchanger reduces.

Manufacturers of MVHR systems almost always quote the performance of their systems in terms of the temperature efficiency. Temperature efficiencies are usually multiplied by 100 and expressed as a percentage efficiency.

Energy Effectiveness, ε:

The mass flows through the supply and exhaust sides of a heat exchanger are not necessarily equal. Because of this, the transfer of energy between the exhaust and supply air is not always reflected by the change in temperatures.

When the two flow rates are not the same the device should then be rated in terms of its energy effectiveness (ε). The energy effectiveness of a heat exchanger is defined as the amount of heat transferred to the incoming air stream divided by the maximum heat transfer possible [Irving, 1994]. The energy effectiveness can be calculated from the following equation:

$$\varepsilon = \frac{m_s(T_2 - T_1)}{m_e(T_3 - T_1)} \quad \textbf{(3.3)}$$

The specific heat of air at constant pressure is assumed to be constant over the range of temperatures involved.

Coefficient of Performance, COP:

The overall coefficient of performance (COP) of a MVHR system is defined as the heat input to the supply air, divided by the total power input (p_i) to the MVHR system. The power input refers to any form of power supplied to the MVHR system in order for it to operate, e.g., electrical power consumed by fans, thermal wheel motors, run-around coil system pumps etc [McIntyre, 1986].

$$\mathrm{COP} = \frac{m_s C(T_2 - T_1)}{p_i} \quad \textbf{(3.4)}$$

3.7 Conclusions

The UK market for MVHR systems is small, especially when compared to the Scandinavian and North American markets. However, the last five years have seen the UK market significantly grow. This growth could be attributed to MVHR systems being used in renovated and new properties which are built more airtight in accordance with more stringent building regulations.

Many studies have shown the benefits of using MVHR systems in terms of providing fresh pre-warmed air, as well as removing moisture and pollutants from within homes. Whilst there are many benefits for installing a system, it is necessary that the houses they are installed within are as airtight as possible, for efficient and economic operation.

Currently cross-flow plate heat exchanger systems dominate the UK market for balanced whole house MVHR systems. Manufacturers of these systems claim heat recovery efficiencies of up to 70% and over.

Current MVHR systems require filters to prevent the build up of dust on the heat exchangers. If filters are not used, systems should be cleaned regularly to prevent a drop off in heat exchanger efficiency. When filters are used they should be changed regularly to prevent reductions in flow rates due to clogged filters.

Heat pump systems have been shown to provide low cost heating with all the benefits of a whole house MVHR system.

4. HEAT PIPES

The following chapter reviews heat pipe technologies. Particular reference is made to revolving heat pipes, as they are used in the systems developed during the research covered by this thesis.

4.1 Introduction

Heat pipes are self contained sealed devices of very high thermal conductance. They can have an equivalent thermal conductivity of several hundred times that of copper. The high thermal conductivity of a heat pipe is achieved by utilising the constant temperature processes of evaporation and condensation, which allow the transfer of heat with very little drop in temperature. Of all the different heat transfer mechanisms, the heat pipe is one of the most efficient.

Historically, heat pipes have been around for many years. The Perkins tube, which is regarded as the predecessor to the heat pipe, was developed by the Perkins family between the middle of the nineteenth century and the beginning of the twentieth century. The Perkins tube design that is closest to the present heat pipe was patented by Jacob Perkins in 1836 [Faghri, 1995]. Most of the Perkins tube designs were thermosyphon type designs, relying on gravity to assist the phase change cycle. The operation of thermosyphons is covered in section 4.2.

In 1929 a waste heat recovery system was proposed by F. W. Gay in US Patent 1725906. It was intended for use as a gas/gas heat pipe heat exchanger. Figure 4.1 illustrates the patent drawing. Several finned thermosyphons were arranged with the evaporator sections vertically below the condensers. A plate seal arrangement was used to separate the two air streams. The working fluids proposed for the system include water, methanol and mercury, depending upon the likely exhaust gas temperatures [Dunn & Reay, 1994].

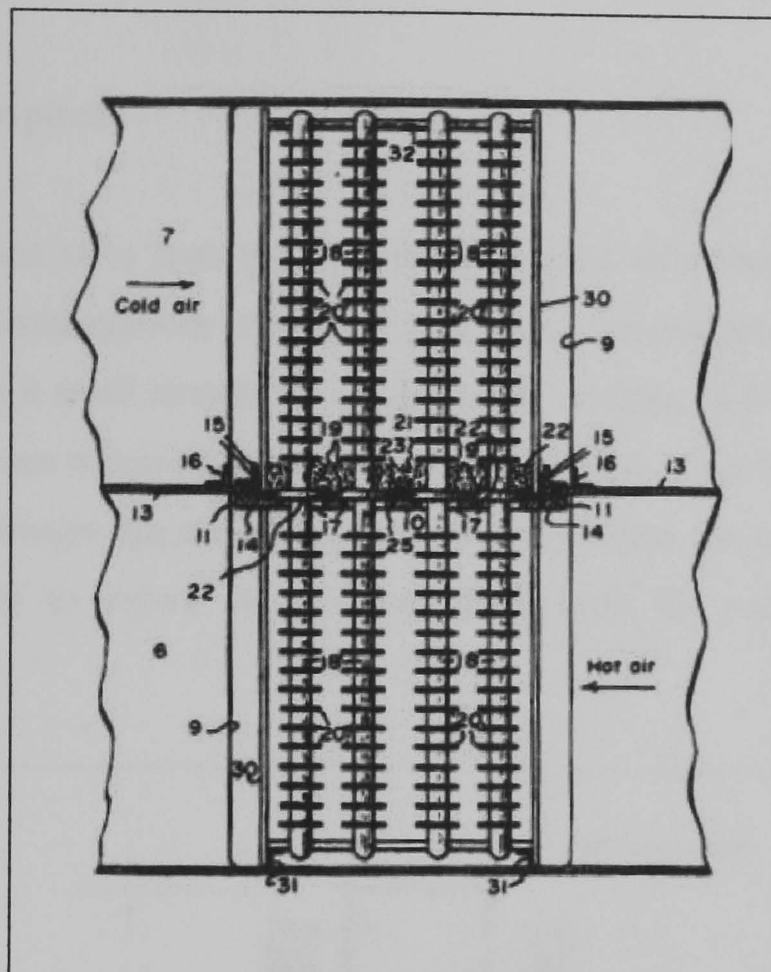


Figure 4.1: Thermosyphon Heat Exchanger

Proposed By F. W. Gay **[Dunn & Reay, 1994]**

The design of a heat pipe that is not reliant on gravity for its operation was originally patented in the early 1940s by R.S. Gaugler, of the General Motors Corporation, Ohio, USA. The device had a continuous evaporation and condensation cycle which used capillary action to return the fluid from the condenser to the evaporator. However, the research by Gaugler was not developed beyond the patent stage and it was not until the 1960s that the heat pipe principle was resurrected.

During the 1960s a lot of research work was carried out on heat pipes. The majority of this work was the development of heat pipes for use in atomic energy and space applications. It is only more recently that heat pipes have been applied to the HVAC industry. Full accounts of the historical development of heat pipes can be found in the texts by Faghri (1995) and Dunn & Reay (1994).

4.2 Principle of operation

4.2.1 Thermosyphons

As the operation of a heat pipe is similar to that of a thermosyphon, the operation of thermosyphons is covered here first. Thermosyphons consist of a pipe containing a small amount of a vaporisable working fluid (such as water or methanol) from which all the air has been evacuated. They have no wicking structure and always operate with the evaporator below the condenser. They rely on gravity to return the working fluid from the condenser to the evaporator.

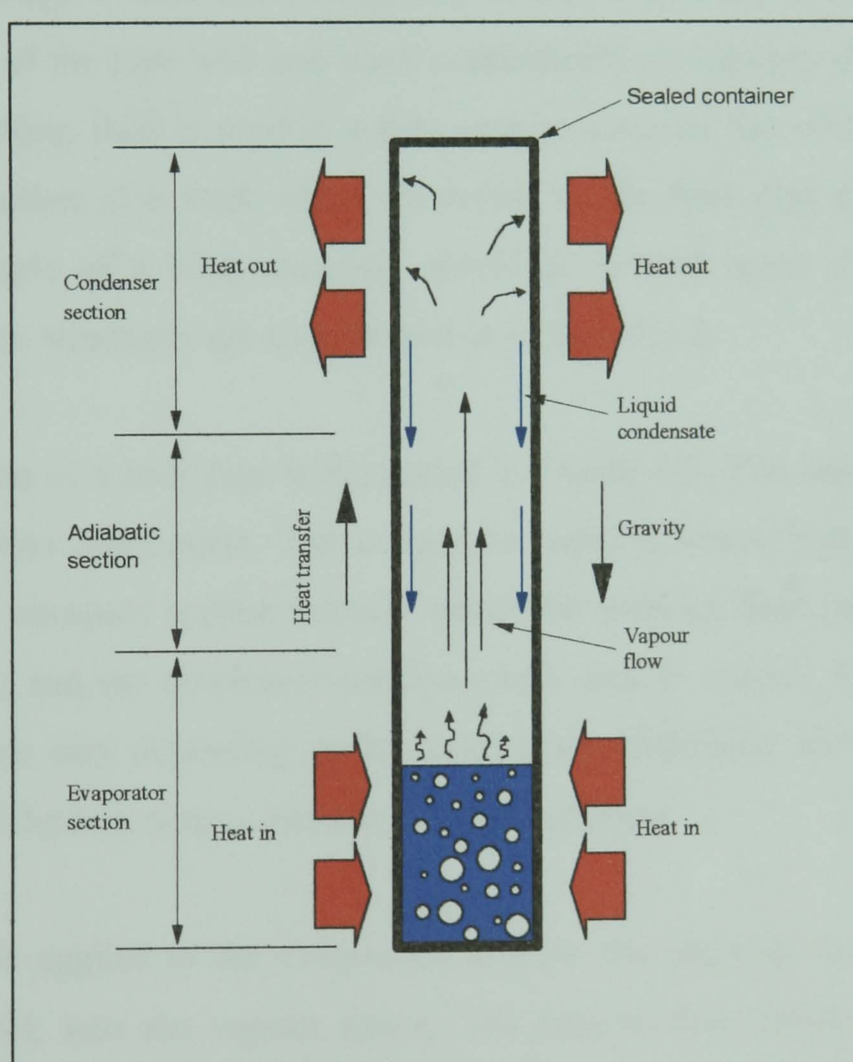


Figure 4.2: Thermosyphon - Gravity Assisted Wickless Heat Pipe

Figure 4.2 illustrates the extremely simple and effective operation of a thermosyphon. When heat is applied to the lower end (evaporator) of the thermosyphon the working fluid is vaporised. The vapour travels up the centre of the tube towards the cold end (condenser) where it condenses, giving out its heat. The condensate then returns to the evaporator by gravity. The process is continuous and allows large quantities of heat to be transferred along the length of the pipe. The main limitation of the thermosyphon is that the evaporator must be located below the condenser so that gravitational forces can return the condensate.

4.2.2 Capillary driven heat pipes

Heat pipes are not restricted by their orientation. Rather than using gravitational forces to return the condensate, in heat pipes the liquid is returned through a wick due to capillary forces. Wicks are positioned on the inner radius of the pipe wall and work continuously as capillary driven pumps. Enough working fluid is used in a heat pipe to saturate the wick with liquid. The construction of a wick varies according to the heat pipe application. A typical example of a wick structure would be several layers of fine gauze. Types of wick structures are summarised in section 4.3.2.

The operation of a heat pipe is illustrated in Figure 4.3. The heat pipe can be divided up into three parts: The evaporator section where heat is input, the adiabatic or transport section through which the working fluid passes (no heat loss or gain) and the condenser section where heat is output. The lengths of these sections vary depending on the application. Adiabatic sections may not be present if the heat source and heat sink are adjacent.

When heat is applied to the evaporator section, the working fluid evaporates from the wick into the vapour space. The vapour then travels towards the condenser due to the higher vapour pressure present in the evaporator. The removal of heat from the condenser end of the heat pipe causes the vapour to condense. When the vapour condenses the working fluid releases its latent heat of vaporisation. The capillary pressure created by the menisci in the wick then pumps the condensate back to the evaporator section [Faghri, 1995]. This process allows the continuous transfer of latent heat of vaporisation from the evaporator to the condenser section.

Heat pipes are near isothermal devices. Within heat pipes the liquid and vapour states are always trying to reach equilibrium. Wherever there is a hot spot, the liquid will boil and wherever there is a slightly colder spot, the vapour will condense [Dinh]. Therefore a heat pipe tends to maintain its total surface area at a uniform temperature (typically ± 2°C).

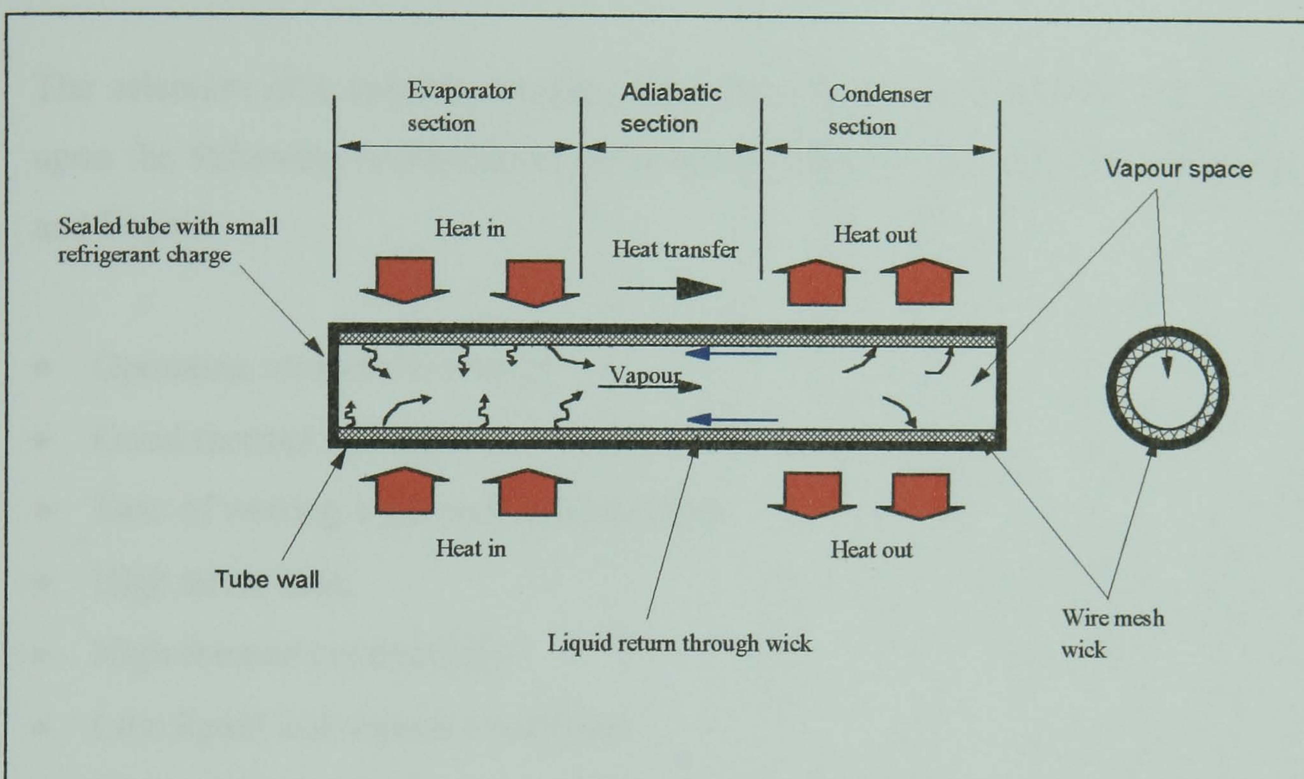


Figure 4.3: Heat Pipe Operation

In addition to gravitation and capillary forces, the use of centrifugal force is another mechanism by which condensate return is achieved. Rotating and revolving heat pipes utilise centrifugal forces for this purpose. These types of heat pipes are covered in section 4.5.

4.3 Heat pipe construction

Heat pipes are suitable for a wide range of applications. Because of their versatility, the conditions within which heat pipes operate varies considerably. The temperature range and working medium are just two of the considerations taken into account in heat pipe design. This section covers the selection criteria for the main components of a heat pipe: working fluids; wicks and capillary structures; and containment vessels. The compatibility of different materials used for each component is also discussed.

4.3.1 Working fluids

The selection of a suitable working fluid for a heat pipe application is based upon the following requirements summarised from the texts by Dunn & Reay and Faghri.

- Operating temperature range.
- Good thermal stability
- Ease of wetting wick and wall materials.
- High latent heat.
- High thermal conductivity.
- Low liquid and vapour viscosities.
- High surface tension
- Cost.

The working fluid specified must be capable of working within the operating temperature of the heat pipe. Table 4.1 lists some typical working fluids with their useful working temperature range. Faghri states: 'As a rule of thumb, the useful range extends from the point where the saturation pressure is greater than 0.1 atm and less than 20 atm. Below 0.1 atm, the vapour pressure limit may be approached. Above 20 atm, the container thickness must increase to the point where the heat pipe becomes limited by the thermal resistance through the container.' It is also necessary to use working fluids with good thermal stability over the working temperature range of the heat pipe. The use of some organic fluids at certain temperatures should be avoided as they may break down into different compounds due to thermal degradation.

Working fluid	Melting point (°C)	Boiling point at atmospheric pressure (°C)	Useful range* (°C)
Helium	-271	-261	-271 - -269
Nitrogen	-210	-196	-203 - -160
Ammonia	-78	-33	-60 - 100
Pentane	-130	28	-20 - 120
Acetone	-95	57	0 - 120
Methanol	-98	64	10 - 130
Fluctec PP2	-50	76	10 - 160
Ethanol	-112	78	0 - 130
Heptane	-90	98	0 - 150
Water	0	100	30 - 200
Toluene	-95	110	50 - 200
Fluctec PP9	-70	160	0 - 225
Thermex	12	257	150 - 350
Mercury	-39	361	250 - 650
Caesium	29	670	450 - 900
Potassium	62	774	500 - 1000
Sodium	98	892	600 - 1200
Lithium	179	1340	1000 - 1800
Silver	960	2212	1800 - 2300

*The useful range is a guide only.

Table 4.1: Heat Pipe Working Fluids [Dunn & Reay, 1994]

For high heat transfer the working fluid should have a high latent heat of vaporisation. This enables large quantities of heat to be transferred with minimum fluid flow and hence low pressure drops within the heat pipe can be maintained. The fluid's thermal conductivity should be as high as possible so as to minimise the temperature gradient through the fluid. A high thermal conductivity will also reduce the possibility of nucleate boiling at the wick/wall interface. The quantity of working fluid used in a heat pipe should be enough to fully saturate the wick and in some cases it is common practice to fill slightly over this amount.

The working fluid should have a high value of surface tension enabling the heat pipe to operate against gravity and generate high capillary driving forces. It is also desirable to use fluids with low vapour and liquid viscosities so as to minimise the resistance to fluid flow.

Different working fluids used in capillary driven heat pipes can be compared by their merit number, M. The merit number takes into account the liquid density, surface tension, latent heat of vaporisation and the viscosity of a working fluid. The merit number should be as high as possible for the most efficient operation over a given temperature range. The merit number can be found using equation 4.1. Table 4.2 gives values of M for some typical working fluids at normal working temperatures.

$$M = \frac{\rho\sigma L}{\mu} \tag{4.1}$$

Where: ρ is the liquid density of the working fluid.

σ is the surface tension.

L is the latent heat of vaporisation.

μ is the viscosity of the liquid working fluid.

Working Fluid	**Mean temperature (°C)**	**M (10^{-6} kW/m^2)**
Ammonia	20	70.24
Acetone	20	44.55
Water	20	177.89
Water	80	390.27
Thermex	400	6.31

Table 4.2: Figure of Merit For Some Working Fluids [Eastop & Croft, 1995]

Water has a very high merit number due to its excellent thermophysical properties, such as high values of latent heat and surface tension. It also has the added benefits of being inexpensive, abundant and safe to use during handling. The physical properties of water make it one of the most widely used working fluids.

Working fluids cannot be chosen purely on their merit number and suitability for a particular temperature range. Other factors, such as cost and the compatibility of the working fluid with the wick and container material, have to be taken into account.

4.3.2 Wicks and capillary structures

The capillary wick structure is used to return the working fluid from the condenser to the evaporator by utilising capillary forces. It should also evenly distribute the working fluid over the inside wall of the evaporator. The geometry of the wick structure must be optimised to achieve maximum heat transfer along the heat pipe. Optimisation involves using a pore size that achieves maximum capillary pressure with low liquid pressure drop along the wick. Small pore sizes generate high capillary pressures and large pore sizes will reduce the liquid pressure drop in the wick. Optimisation has lead to the development of many different types of wicks suited to a wide range of applications.

In most cases the entire inner circumference of the heat pipe is covered by the wick. The wick structure must therefore be thin and have a high value of thermal conductivity in order to prevent radial temperature drops through the wick. The design of the wick is complex and influenced by many factors, all of which must be considered for optimal design. Table 4.3 lists some common wick types with their associated properties.

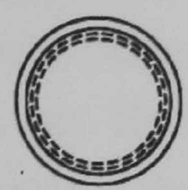
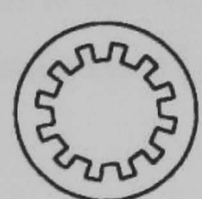
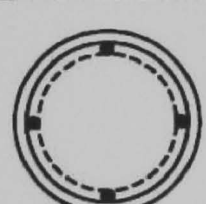
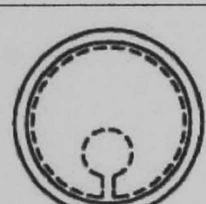
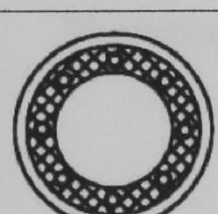
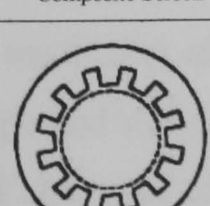
Wick Type	Capillary Pumping	Thermal conductivity	Permeability	Comments
 Wrapped Screen	High	Low	Low – average	Homogeneous wick: Multiple wraps of wire screen mesh
 Sintered Metal	High	Average	Low – average	Homogeneous wick: Packed spherical particles, Felt, metal fibres or powder
 Axial Grooves	Low	High	Average – high	Homogeneous wick: Rectangular, circular, triangular or trapezoidal grooves
 Open Annulus	Low	Low	High	Homogeneous wick: Wire screen mesh spaced from wall
 Open Artery	Low	High	High	Homogeneous wick: Wire screen mesh formed into artery and wall lining
 Integral Artery	High	High	Average – high	Homogeneous wick: Homogeneous material with built in arteries
 Composite Screen	High	Low – average	Average	Composite wick: Two or more layers of homogeneous material. The material next to the wall has the largest pore size
 Screen Covered Grooves	High	High	Average – high	Composite wick: Axial grooves covered with one screen

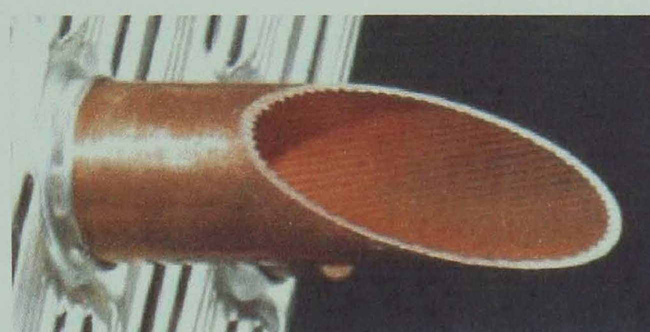
Table 4.3: Typical Wick Designs [Faghri, 1995]

The wicks shown in Table 4.3 fall into two categories, homogeneous and composite wick structures. Homogeneous wicks are made from a single base material, where as composite wicks are manufactured from several base materials. Homogeneous wicks tend to be less complex and cheaper to manufacture than composite structures.

Probably the most common homogeneous wick structures are screen mesh wicks. These consist of a single or several layers of metal cloth mesh sprung against the inner wall of the heat pipe. By varying the size and density of pores in the wick the characteristics of the wick structure can be changed. A typical piece of wire mesh wick is shown in Photograph 4.1.

Photograph 4.1: Phosphor Bronze 150μm Screen Mesh Wick

One problem associated with screen mesh wicks is that there is often poor contact between the wick and the pipe wall and between different layers of the screen mesh. This results in poor thermal conductivity through the wick. However, this is not a problem when axial grooves in the pipe wall are used as the capillary structure. The grooves are formed during extrusion of the tube or by machining along the axial direction of the inside wall. Grooves vary in shape and size and the groove pattern can vary, e.g. straight or spiral. Grooves will also tend to increase the heat transfer coefficient as they act like fins on the inside surface of the pipe. The permeability of the grooves is high but they can only generate low capillary pressures and are therefore not suited for applications working against gravity. Photographs 4.2(a) – 4.2(c) illustrate tubes with internal capillary grooves manufactured by Hitachi.



(a) Spiral grooved pipe:
Hitachi Thermofin – Ex, Hex Tube

(b) Spiral grooved pipe:
Hitachi Thermofin – SP Tube

(c) Longitudinal grooved pipe:
Hitachi Thermofin – SG Tube

Photographs 4.2a-c: Capillary Structures [Hitachi]

Sintered metal wicks are another form of homogeneous wicks. This type of wick comprises of small spherical metal particles in powder form, which are heated until the spheres are sintered to each other and to the inner wall of the tube. Sintered wicks can generate high capillary pressures and due to the connectivity between the wick and tube, these wicks have high thermal conductivity. The disadvantages with these wicks are that they have high liquid pressure drops and are difficult and costly to manufacture.

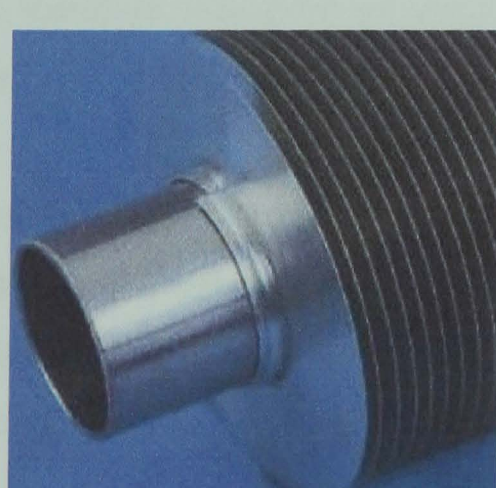
Composite wicks comprise of more than one basic wick structure to form a single combined structure. By combining wick structures it is possible to have a wick that has small pores for generating high capillary pumping pressures and large pores for low liquid pressure drops. A screen mesh wick comprising of two screens of different pore size is the simplest form of a composite wick structure. The screen mesh with the large pore is used against the tube wall as the liquid return path, and the smaller pore sized mesh is positioned adjacent to the vapour space to develop high capillary pressures. Composite wicks can also be formed by using a screen mesh wick in an axially grooved tube. Again, this form of composite wick will have high capillary pumping pressures with low liquid pressure drops.

An important consideration in wick design is that the wick structure should be made from a material that is compatible with the working fluid and containment vessel. It is also often the cost that is the deciding factor in the choice of wick used. It is more desirable to use wicks that are inexpensive and easy to form.

4.3.3 Heat pipe containment vessels

The majority of heat pipe containment vessels are a length of pipe with caps at either end. Other forms of vessels are used for special types such as flat plate or loop heat pipes. All containment vessels should possess the following characteristics. Vessels should be leak proof containers that completely isolate the working fluid from the medium the heat pipe is operating in. The material the vessel is fabricated from should have high thermal conductivity to ensure minimum temperature drop between the wick/working fluid and the heat source or heat sink. For ease of fabrication, it should be made from a material that is easily machinable. The material used should be compatible with the wick and working fluid.

Most of the time, the problem with using heat pipes is how to transfer heat to and from the environment, not with the heat transfer within the heat pipe itself. This is a particular problem when heat pipes are used as air to air heat exchangers due to the inherent difficulty of transferring heat to and from an air stream. The external heat transfer coefficients play a far greater part in limiting the heat transfer performance of a heat pipe. Because of this, extended surfaces (fins) are often used on heat pipes to improve heat transfer. Fins are regarded as a secondary surface. They are an extension of the primary surface which, in the case of heat pipes, is the external surface of the tube. Photographs 4.3a and 4.3b illustrate typical examples of finning used on tubes to increase heat transfer.



(a) Wound aluminium 'L' Fin

(b) Copper wire finning

Photographs 4.3a & 4.3b: External Tube Fins [Tube Fins Ltd]

The total surface area for heat transfer is increased by attaching fins to the heat pipe surface. A large variety of fin geometries exist for different heat transfer applications. The geometry of fins can also be chosen to promote laminar flow with small pressure drops across the fin surface or turbulent flow with higher pressure drops. Laminar flow has less energy penalties in terms of fan or pump energy consumption. However, boundary layers are formed in laminar flow which inhibit heat transfer. Therefore it is often more desirable to have turbulent flows which can offer better heat transfer due to the enhanced mixing arising from velocity fluctuations in a turbulent boundary layer. An example of finning with turbulent characteristics is shown in Photograph 4.3b. The wire finning shown has good heat transfer characteristics because of its ability to promote turbulence in the fluid passing over it.

4.3.4 Compatibility of materials

As already mentioned in the previous sections the compatibility of the materials used in heat pipes is very important. The longevity of a heat pipe cannot be assured if the container, wick and working fluid are not compatible. It is therefore important to specify materials that are compatible with each other.

Corrosion and the generation of non condensable gases through reactions can severely reduce the heat pipe performance or even cause failure. Therefore, the working fluid should not chemically react with the other materials. The wick and container should also be compatible with one another.

Table 4.4 is a combination of the compatibility data given in the texts by Dunn & Reay and Faghri. The compatibility data shown in these texts is from general scientific references on chemicals and materials. It is also based on life time tests of heat pipes fabricated from different material combinations.

Working Fluid	Recommended	Not Recommended
Water	Copper, 347 Stainless Steel, Titanium, Nickel	Aluminium, Inconel, Carbon Steel, Stainless Steel, Silica
Ammonia	Aluminium, Stainless Steel, Carbon Steel, Iron, Nickel	Copper
Methanol	Stainless Steel, Iron, Copper, Brass, Silica, Nickel	Aluminium
Acetone	Aluminium, Stainless Steel, Copper, Brass, Silica	
Heptane	Aluminium	
Thermex (Dowtherm A)	Stainless Steel, Copper, Silica, Nickel	
Lithium	Tungsten, Tantalum, Molybdenum, Niobium	Stainless Steel, Nickel, Inconel, Titanium
Potassium	Stainless Steel, Inconel	Titanium
Sodium	Stainless Steel, Nickel Inconel, Niobium	Titanium
Cesium	Titanium, Niobium	
Mercury	Stainless Steel	Molybdenum, Nickel, Titanium, Inconel, Tantalum, Niobium
Lead	Tungsten, Titanium	Stainless Steel, Nickel, Inconel, Titanium, Niobium
Silver	Tungsten, Tantalum	Rhenium

Table 4.4: Compatibility Data [Dunn & Reay, (1994) and Faghri, (1995)]

4.4 Heat transfer limitations

The operation of heat pipes is subject to a number of constraints. These constraints are physical factors that can limit heat transfer and reduce the heat pipe performance. The design of a heat pipe should account for all the limiting factors. The maximum heat transport limitation of a heat pipe at a given temperature is defined by the lowest limiting constraint. The limiting factors of heat pipe operation are summarised in this section.

4.4.1 Boiling limit

Evaporation within the heat pipe evaporator occurs from the liquid surface. For low values of heat flux the heat is transferred to the liquid surface by conduction and convection. As the heat flux is increased, nucleate boiling occurs at nucleation sites on the surface of the wick. These bubbles increase the heat transfer by transferring latent heat energy to the surface and by increasing convective heat transfer. However, at high heat levels of heat flux, the large number of bubbles that form on the wick can prevent the liquid from wetting the pipe wall, causing hot spots. The boiling limit is the point at which boiling in the evaporator causes the wick to dry out.

4.4.2 Entrainment limit

The liquid and vapour flows in heat pipes travel in opposite directions. At the interface of the two flows, the vapour flow will exert a shear force on the returning liquid. When the relative velocity of the two flows is high, the shear forces can entrain returning liquid into the vapour flow. The evaporator will dry out if the entrainment becomes too large. At this point of operation the entrainment limit has been reached.

4.4.3 Capillary limit

The most commonly encountered limit to heat transfer in low temperature heat pipes is the capillary limit. When the maximum capillary pressure is exceeded by the sum of the liquid and vapour pressure drops, the capillary structure is unable to provide enough liquid to the evaporator. When this occurs, the capillary limit is reached and the capillary structure is unable to provide enough liquid to the evaporator. If attempts are made to increase the heat transfer above the capillary limit the evaporator will dry out.

4.4.4 Other limitations

In addition to the three limitations already covered, heat pipes can be subject to several other constraining limits. The remaining limits are the condenser heat transfer limit, vapour continuum limit, frozen start up limit, sonic limit, and vapour pressure limit. These limits are not covered here as they are usually only associated with specialist heat pipes. For example, the sonic and vapour pressure limits are problems that occur more in liquid-metal heat pipes. Full explanations of these limitations are given in the texts by Faghri (1995) and Dunn & Reay, 1994.

4.5 Rotating and revolving heat pipes

This section covers heat pipes that use centrifugal forces to enhance condensate return and heat transfer.

4.5.1 Introduction

Rotating and revolving heat pipes use the centrifugal force produced by the motion of the pipe to pump the working fluid from the condenser back to the evaporator [Harley, 1995]. Rotating heat pipe characteristics were first presented in a Ph.D. thesis by Morris (1964). More detailed work by Gray (1969) demonstrated that rotating heat pipes are capable of transferring significantly more heat than similar stationary heat pipes.

Rotating and revolving heat pipes differ from each other due to their position relative to their axis of rotation. Rotating heat pipes are defined as those heat pipes which rotate around their own axis, i.e. they are co-axial. Heat pipes which rotate around an axis located some distance from the central axis of the heat pipe, are revolving heat pipes. Figure 4.4 illustrates the difference between rotating and revolving heat pipes.

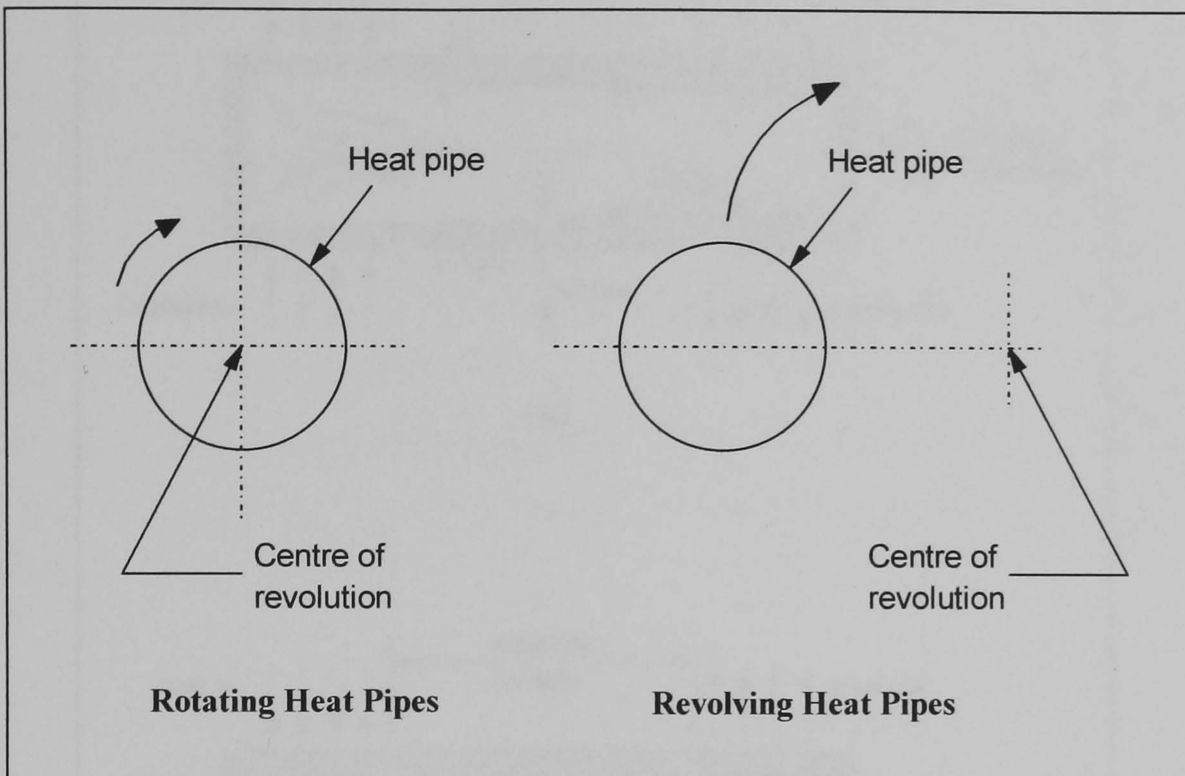


Figure 4.4: Rotating and Revolving Heat Pipes

4.5.2 Rotating heat pipes

Rotating heat pipes often have an internal taper along their axis to drive/centrifuge the condensate back to the evaporator. The pipes are tapered so that the internal diameter is smaller at the condenser than it is at the evaporator. This change in cross section creates a component $\omega^2 r\sin\alpha$ along the wall of the pipe from the centrifugal acceleration $\omega^2 r$, where α is the taper angle. The condensed working fluid will flow along the wall back to the evaporator due to this force [Dunn & Reay, 1994]. This pumping effect is the main driving force for condensate return in tapered rotating heat pipes. Figure 4.5(a) illustrates an internally tapered rotating heat pipe.

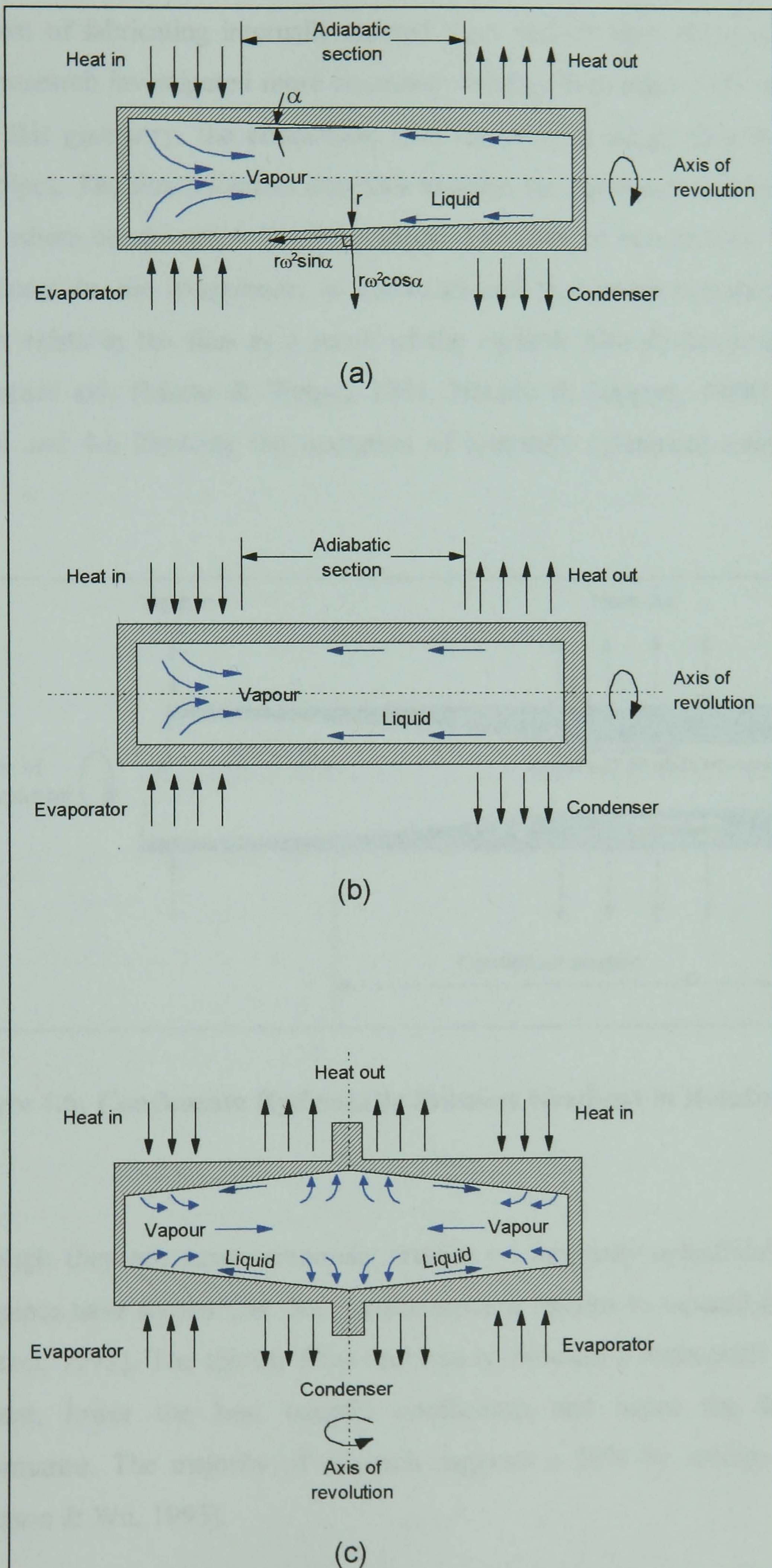


Figure 4.5: Types of Rotating Heat Pipes

(a) Internally tapered **(b) Internally cylindrical** **(c) Radially rotating**

Early research on rotating heat pipes concentrated on tapered pipes. However, the cost of fabricating internally tapered pipes is very high. Because of this, later research investigated more economic rotating heat pipes with no tapers. With this geometry, the condensate film thickness is larger than in tapered heat pipes. The film thickness increases towards the condenser end of the heat pipe, where condensation is taking place. The flow of condensate, from the condenser to the evaporator, is due to an axial hydrostatic pressure gradient which exists in the film as a result of the variable film thickness along the condenser axis [Marto & Weigel, 1981, Nimmo & Leppert, 1968]. Figures 4.5(b) and 4.6 illustrate the operation of internally cylindrical rotating heat pipes.

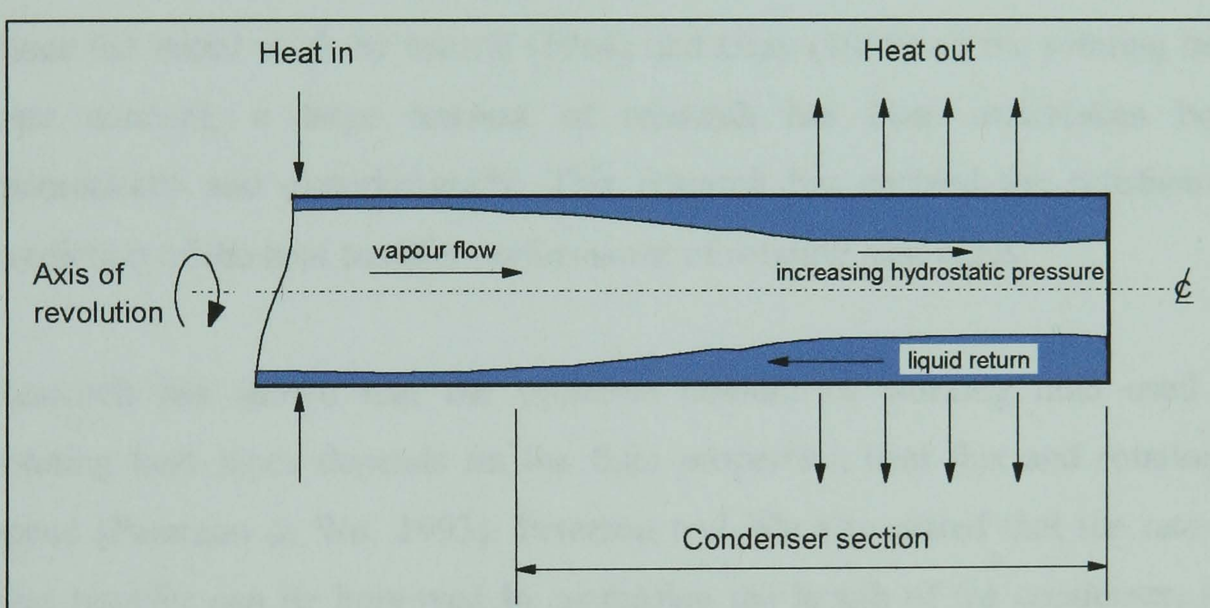


Figure 4.6: Condensate Hydrostatic Pressure Gradient in Rotating Pipes

Although they are more economic, studies on internally cylindrical rotating heat pipes have shown that their performance is inferior to tapered heat pipes [Roetzel, 1975]. The thicker films required to produce a hydrostatic pressure gradient, lower the heat transfer coefficients and hence the heat pipe performance. The majority of research suggests a 30% by volume fill ratio [Peterson & Wu, 1993].

Research has shown that the heat transfer performance of internally cylindrical pipes can be enhanced by using pipes with internal spiral grooves or fins such as those shown in Photographs 4.2a and 4.2b. The spiralled grooves act as a pump to return the condensate to the evaporator. Heat transfer performance with these surfaces was shown to be 2 to 4 times as effective as smooth walled pipes [Marto & Weigel, 1981].

Other forms of rotating heat pipes have condensate return that flows radially, as opposed to the more common axial flow. Figure 4.5(c) illustrates a heat pipe with radial condensate return. These heat pipes are often disc shaped devices. Experiments were carried out using such devices with radial flow for cooling brakes in heavy road vehicles [Maezawa, 1981]. The experiments demonstrated that brake temperatures could be greatly reduced.

Since the initial work by Morris (1964) and Gray (1969) on the rotating heat pipe concept, a large amount of research has been undertaken both theoretically and experimentally. This research has enabled the satisfactory prediction of the heat transfer performance of rotating heat pipes.

Research has shown that the optimum amount of working fluid used in rotating heat pipes depends on the fluid properties, heat flux and rotational speed [Peterson & Wu, 1993]. Peterson and Wu also stated that the rate of heat transfer can be improved by increasing the length of the condenser, the taper angle or the average radius of the heat pipe.

The rotation of heat pipes has been shown to enhance the heat transfer capacity by reducing the condensate film thickness and hence the thermal resistance. The condenser heat transfer coefficient is proportional to the angular velocity to the 2/5ths power [Vasiliev & Khrolenok, 1993]. Therefore, heat transfer increases with increasing rotational speed. The improved condensate return due to rotation also enhances the heat transfer capacity.

For high heat transfer coefficients nucleate boiling is the most favourable mode of operation [Dunn & Reay, 1994]. However, for low heat fluxes and thick liquid films the centrifugal force reduces nucleate boiling in the refrigerant film. This is compensated for by strong natural convection present in the liquid. The latter mode of operation indicates that evaporation takes place from the liquid surface [Faghri, 1995].

The entrainment and boiling limits are suppressed in rotating heat pipes. Any droplets that may be entrained tend to be thrown back out due to the centrifugal forces present. As already stated, nucleate boiling is reduced due to the centrifugal forces present. In addition, rotating heat pipes are unaffected by capillary limits.

Dunn and Reay (1994) reference work by Al-Jumaily (1973) which establishes a figure of merit, M' for rotating heat pipes. This figure is different to the figure of merit established for capillary driven heat pipes. The value takes into account the thermal conductivity of film thickness and neglects surface tension values which are important for capillary action. Water has high figures of merit and is the optimum working fluid in the temperature range 20°C to 100°C.

$$M' = \frac{\rho^2 L k^3}{\mu} \tag{4.2}$$

Where: ρ is the liquid density of the working fluid.
k is the liquid thermal conductivity.
L is the latent heat of vaporisation.
μ is the viscosity of the liquid working fluid.

4.5.2.1 Rotating heat pipe applications

Rotating heat pipes have been applied to a wide range of applications. By virtue of their rotation, they are particularly suitable for heat transfer applications in rotating machinery. Dunn and Reay refer to work carried out by Polasek (1973) and Gray (1969). Polasek undertook experiments on a.c.

motors which incorporated rotating heat pipes, see Figure 4.7. The results of the experiments showed that the power output of these motors could be increased by 15% without any rise in winding temperature.

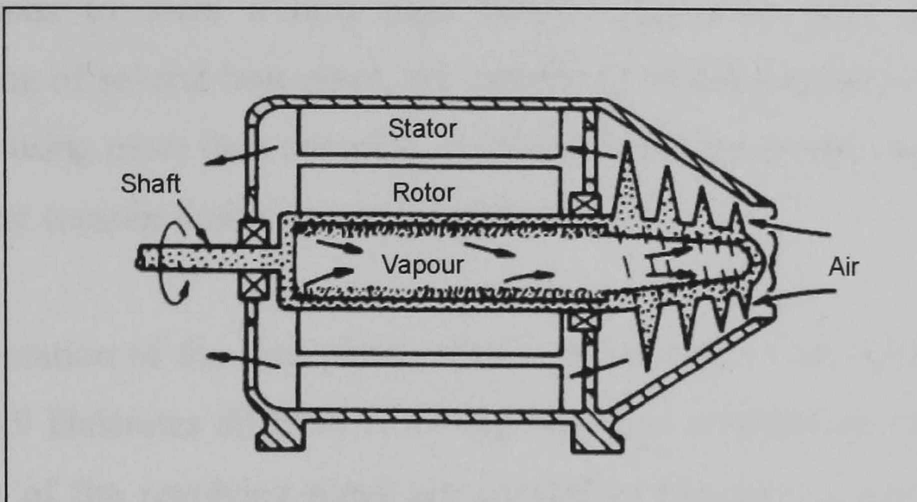


Figure 4.7: Rotating Heat Pipes For Cooling Motor Rotors [Dunn & Reay, 1994]

Gray proposed using rotating heat pipes in a compact air conditioning system. The system proposed was a vapour compression heat pump system, based around a tapered rotating heat pipe. Figure 4.8 illustrates the system.

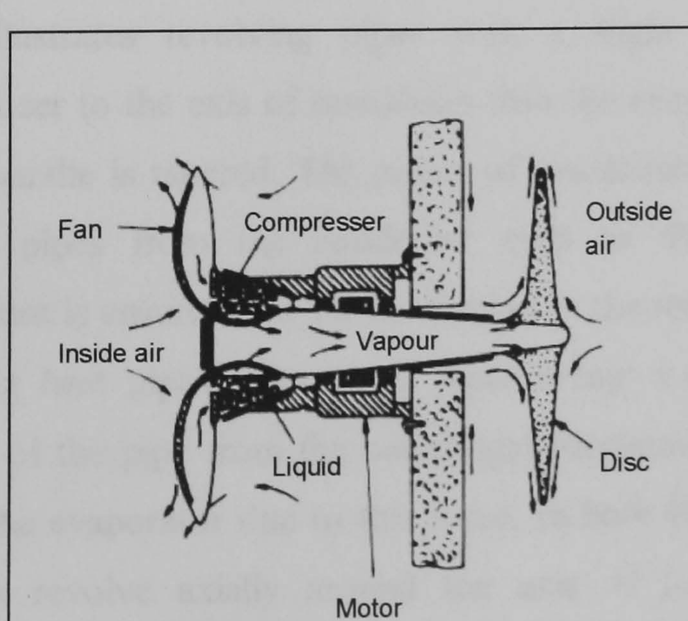


Figure 4.8: A Compact Air Conditioning Unit Based on the Wickless Rotating Heat Pipe [Dunn & Reay, 1994]

4.5.3 Revolving heat pipes

As already stated, revolving heat pipes revolve about an axis which does not coincide with the heat pipes' axes. A number of pipes are situated around a central axis to form a heat pipe bundle. Revolving heat pipe bundles, comprising of several heat pipes, are capable of transferring large quantities of heat. By using more than one pipe, revolving heat pipe systems are not subject to the heat transfer limitations of individual pipes.

The orientation of the heat pipes, relative to the axis of revolution, can vary. Figure 4.9 illustrates different revolving heat pipe orientations. In Figure 4.9a the axes of the revolving pipes are parallel to the axis of revolution. With parallel revolving heat pipes, the return of the condensate is ensured by the force resulting from the action of the centrifugal force on the difference in fluid level between the evaporator and condenser. This works in the same way as condensate return in internally cylindrical rotating heat pipes. Figure 4.10 illustrates the varying film thickness along the heat pipe. In revolving heat pipes, the working fluid film is localised and not spread around the entire inside wall of the pipes. The location of working fluid is subject to gravitational and centrifugal forces, this is explained later.

Figure 4.9b illustrates revolving pipes with a slight tilt, such that the condenser is closer to the axis of revolution than the evaporator. In this case, the heat pipe bundle is tapered. The radius of revolution increases along the length of the pipes from the condenser ends to the evaporator ends. Condensate return is ensured in a similar fashion to the return of condensate in tapered rotating heat pipes. The tilting pipes create a component of force along the wall of the pipe from the centrifugal acceleration. The condensate will return to the evaporator due to this force. In both Figures 4.9a and 4.9b the heat pipes revolve axially around the axis of rotation. Figure 4.9c illustrates a third revolving heat pipe configuration where the heat pipes revolve radially around the axis of rotation. The axes of the heat pipes are perpendicular to the axis of rotation.

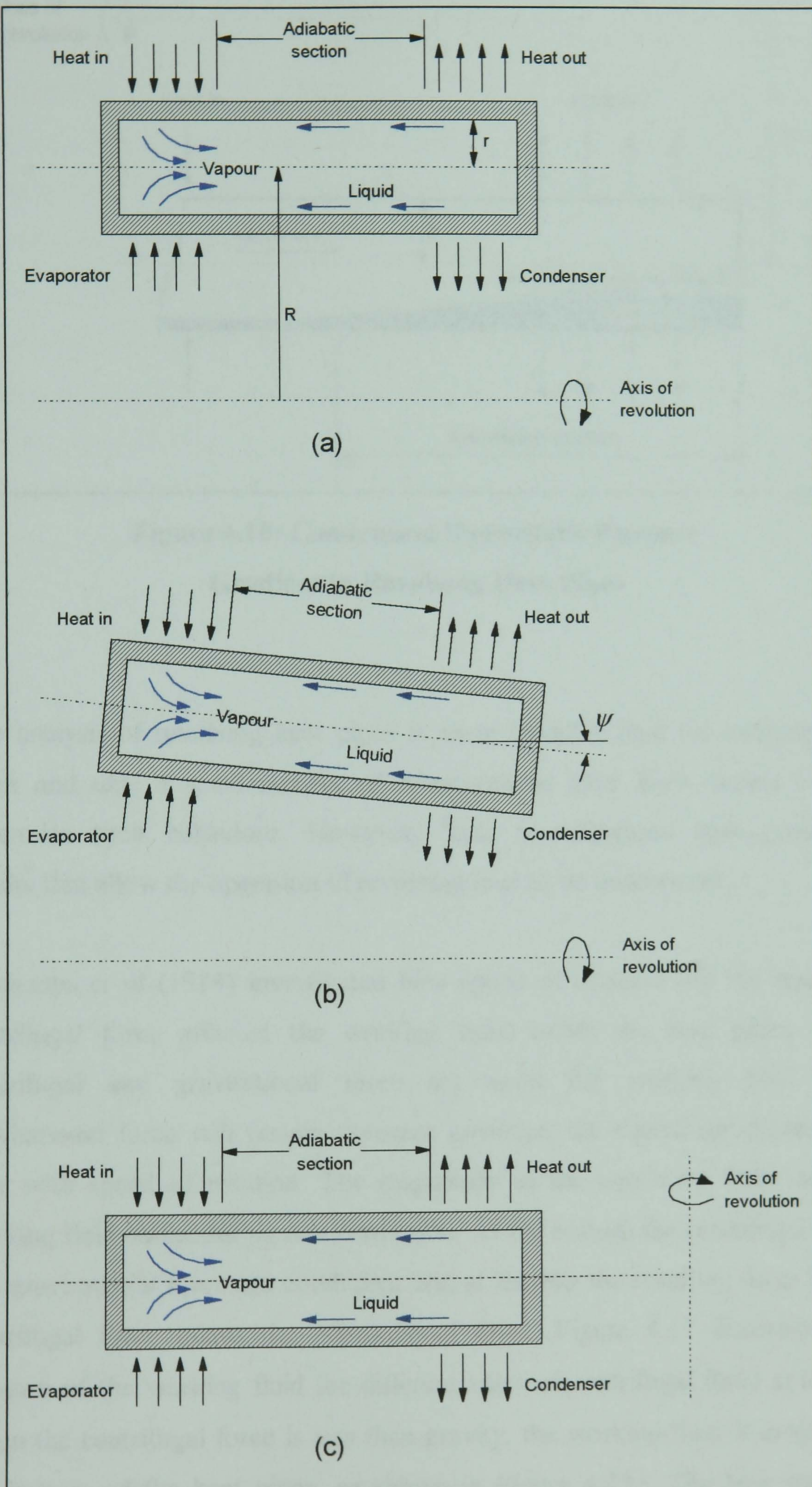


Figure 4.9: Types of Revolving Heat Pipes

(a) Parallel axial (b) Tilted axial (c) Perpendicular radial

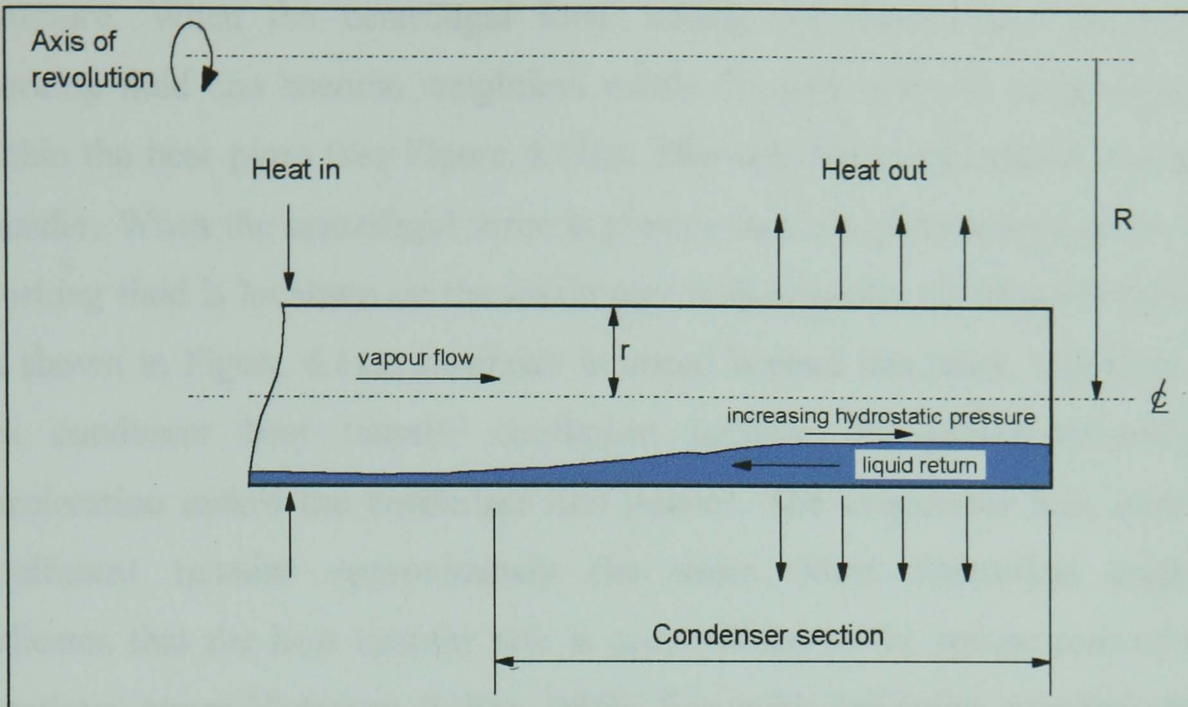


Figure 4.10: Condensate Hydrostatic Pressure Gradient in Revolving Heat Pipes

The analysis of revolving heat pipes is more complex than for rotating heat pipes and only a small number of investigations have been carried out to determine their behaviour. However, these investigations have produced results that allow the operation of revolving heat to be understood.

Bontemps *et al* (1984) investigated how speed of rotation and the resulting centrifugal force affected the working fluid within the heat pipes. Both centrifugal and gravitational force act upon the working fluid. The gravitational force will remain constant however, the centrifugal forces will vary with speed of rotation. The magnitude of the combined force on the working fluid varies during one revolution. At the bottom the centrifugal force and gravitational force are combined, and at the top the resulting force is the centrifugal force minus the gravitational force. Figure 4.11 illustrates the location of the working fluid for different vales of centrifugal force $\omega^2(R+r)$. When the centrifugal force is less than gravity, the working fluid is located in the bottom of the heat pipes, as shown in Figure 4.11c. The heat transfer coefficients tend to be low due to the relatively thick condensate films with high thermal resistance. At these speeds, there are no pumping forces due to rotation. Condensate return would need enhancing with the use of a wick structure. When the centrifugal force equals the gravitational force, the working fluid can become weightless within the pipe and will splash around within the heat pipes (see Figure 4.11b). This splashing will increase the heat transfer. When the centrifugal force is greater than the gravitational force, the working fluid is localised on the inside pipe wall opposite the axis of rotation, as shown in Figure 4.11a. Increases in speed beyond this point, will increase the condenser heat transfer coefficient because the greater centrifugal acceleration makes the condenser film thinner. The evaporator heat transfer coefficient remains approximately the same. Most theoretical analysis indicates that the heat transfer rate is proportional to the square root of the rotational speed [Peterson & Wu, 1993]. For stable operation, revolving heat pipes must operate in the region shown in Figure 4.11a [Bontemps *et al*,1984].

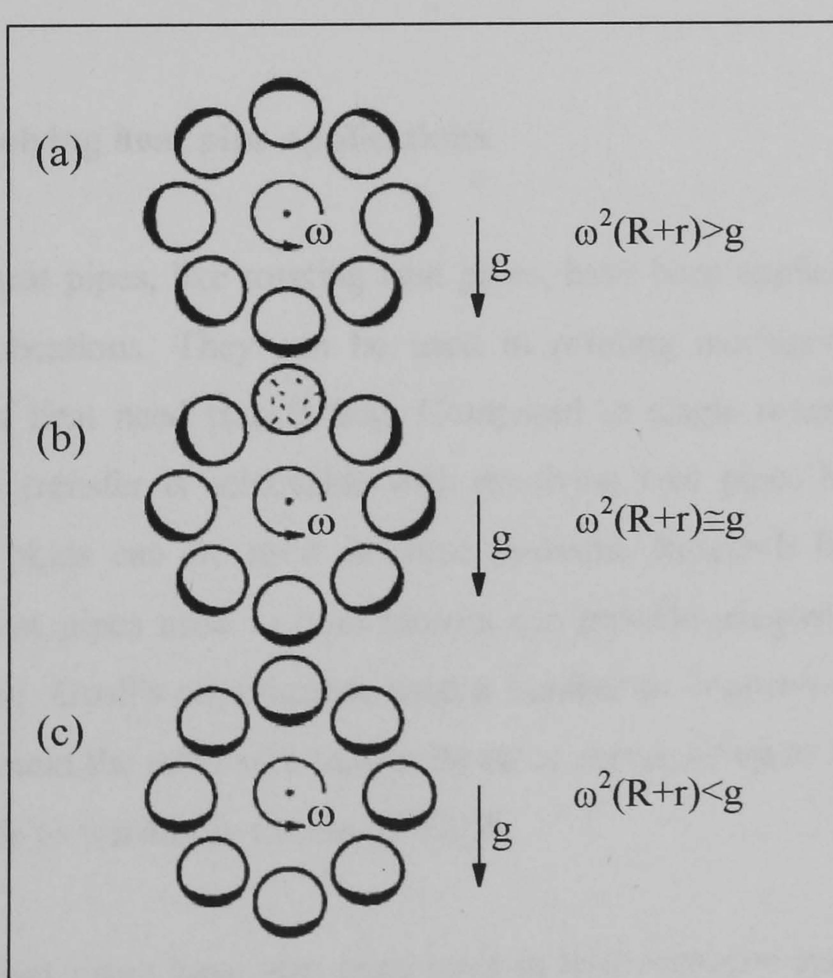


Figure 4.11: The Effect of Centrifugal Force on the Location of Working Fluid

Research by Chen and Tu (1987) and Chen and Lou (1990) showed that tilted revolving heat pipes provide better heat transfer than parallel revolving heat pipes. This is due to the component of force along the tube walls from the centrifugal acceleration.

Studies have also been undertaken to establish the optimum quantity of working fluid used within revolving heat pipes. Fluid loads are quoted as a fill ratio, which is a percentage of the total internal pipe volume. There is some discrepancy between the working fluid fill ratio recommended from different research. Keiyou and Maezawa (1990) recommend an optimum fill ratio of approximately 10%, whilst Curtila and Chataing (1984) state that the fill ratio must be between 25% and 30%. Peterson and Wu (1993) state that the optimum fill ratio should be between 10% and 30% by volume, depending on the power input used.

4.5.3.1 Revolving heat pipe applications

Revolving heat pipes, like rotating heat pipes, have been applied to many heat transfer applications. They can be used in rotating machinery where large quantities of heat need transferring. Compared to single rotating heat pipes, greater heat transfer is achievable with revolving heat pipes because a large number of pipes can be used in these systems. Research has shown that revolving heat pipes used to cool motors can provide adequate heat transfer [Groll, 1978]. Groll's experiments used a number of copper/water heat pipes arranged around the rotor of a motor. At rotor speeds of up to 6000 rpm, each pipe was able to transfer in excess of 500W.

Revolving heat pipes have also been used in heat recovery systems. Work in Japan [Niekawa *et al*, 1981] used revolving heat pipes for gas to gas heat recovery. A heat pipe bundle was formed using several revolving heat pipes located eccentrically around a central shaft. The evaporator and condenser sections of the pipes were positioned in different gas streams at different temperatures. Heat was transferred from the hot stream to the cold stream. A similar configuration was used for gas to vapour heat recovery. Figure 4.12 illustrates the two systems and Photograph 4.4 shows a prototype revolving heat pipe heat exchanger used as a waste heat boiler.

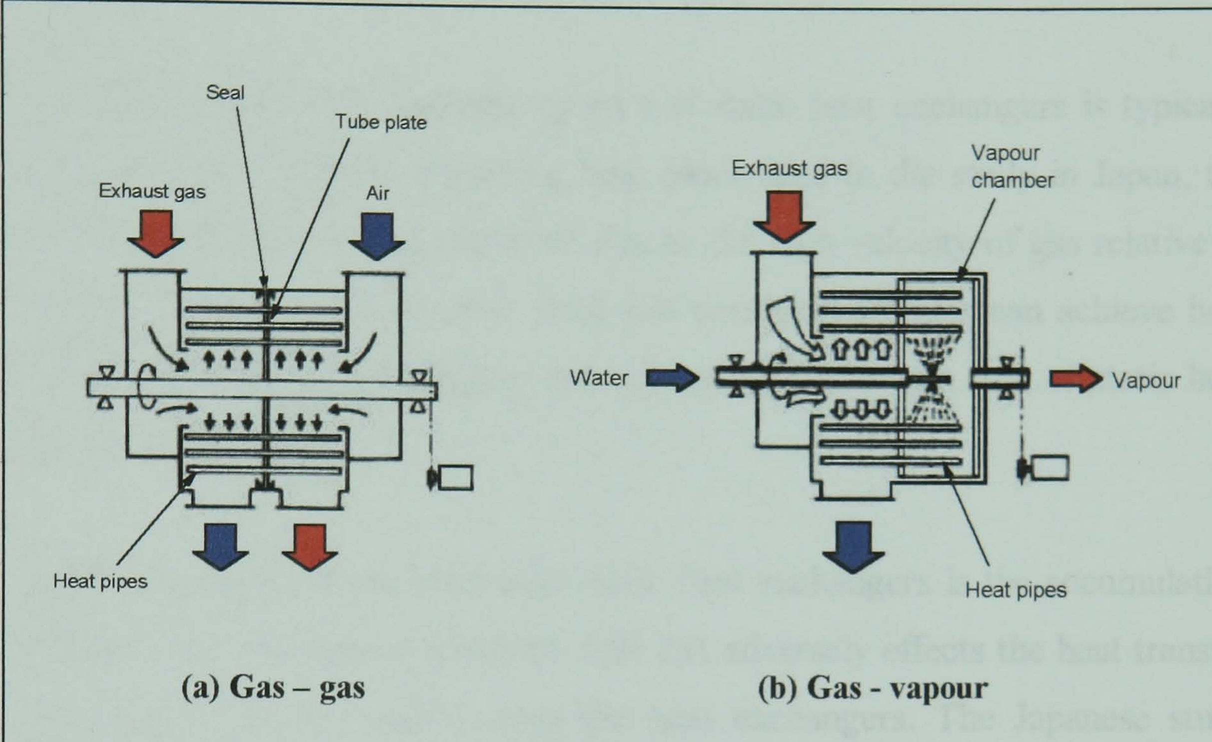
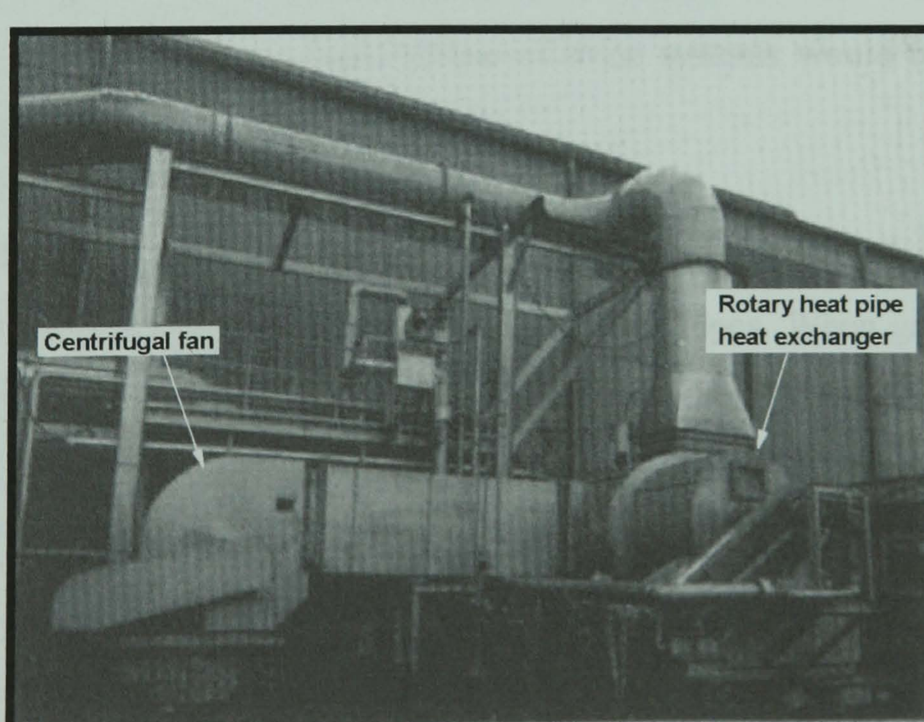


Figure 4.12: Revolving Heat Pipe Heat Exchangers [Niekawa *et al*, 1981]



Photograph 4.4: Revolving Heat Pipe Heat Exchanger used as a Waste Heat Boiler [Dunn & Reay, 1994]

The rotary heat exchanger system investigated by Niekawa *et al,* offered a number of benefits over other systems. The overall heat transfer was shown to be better than stationary heat pipe systems. Centrifugal forces improved the condenser heat transfer coefficients of the heat pipes. The heat transfer rate could be varied by changing the speed of rotation.

The thermal resistance between gases and static heat exchangers is typically high. However, with the revolving heat pipes used in the study in Japan, the external heat transfer was improved due to the high velocity of gas relative to the pipe surface. Niekawa *et al* state that revolving systems can achieve heat transfer more than 120% higher per unit surface area than that of static heat pipe heat exchangers.

A problem often encountered with static heat exchangers is the accumulation of dirt on heat exchanger surfaces. This dirt adversely effects the heat transfer and can reduce flow rates across the heat exchangers. The Japanese study found that dust accumulation and fouling of the rotating heat exchangers was less than for stationary heat exchangers. The induced turbulent nature of the gases passing over the heat pipes prevented the build up of dust. This feature of revolving heat pipe systems makes them suitable for use in dust laden or dirty gases. Cleaning and maintenance of these systems would be lower than stationary systems.

Another feature of revolving heat pipes demonstrated by the Japanese study, was a fan effect produced by the revolving heat pipes. When gases were fed by fans into the centre of the heat pipe bundle and exited from the periphery, the pressure drop across the pipes was reduced. This is an energy saving feature, as the power required by the fans used in the system is less than if stationary heat pipes were used.

A self cleaning rotary heat exchanger using revolving heat pipes (US Patent No. 4,640,344) has all the attributes of the systems investigated by Niekawa *et al*: high rates of heat transfer; low pressure drops; and low rates of fouling [Pravda, 1987]. The system uses a revolving finned heat pipe heat exchanger for processing hot contaminated gas flows emanating from appliances such as laundry driers. Figure 4.13 Illustrates the system.

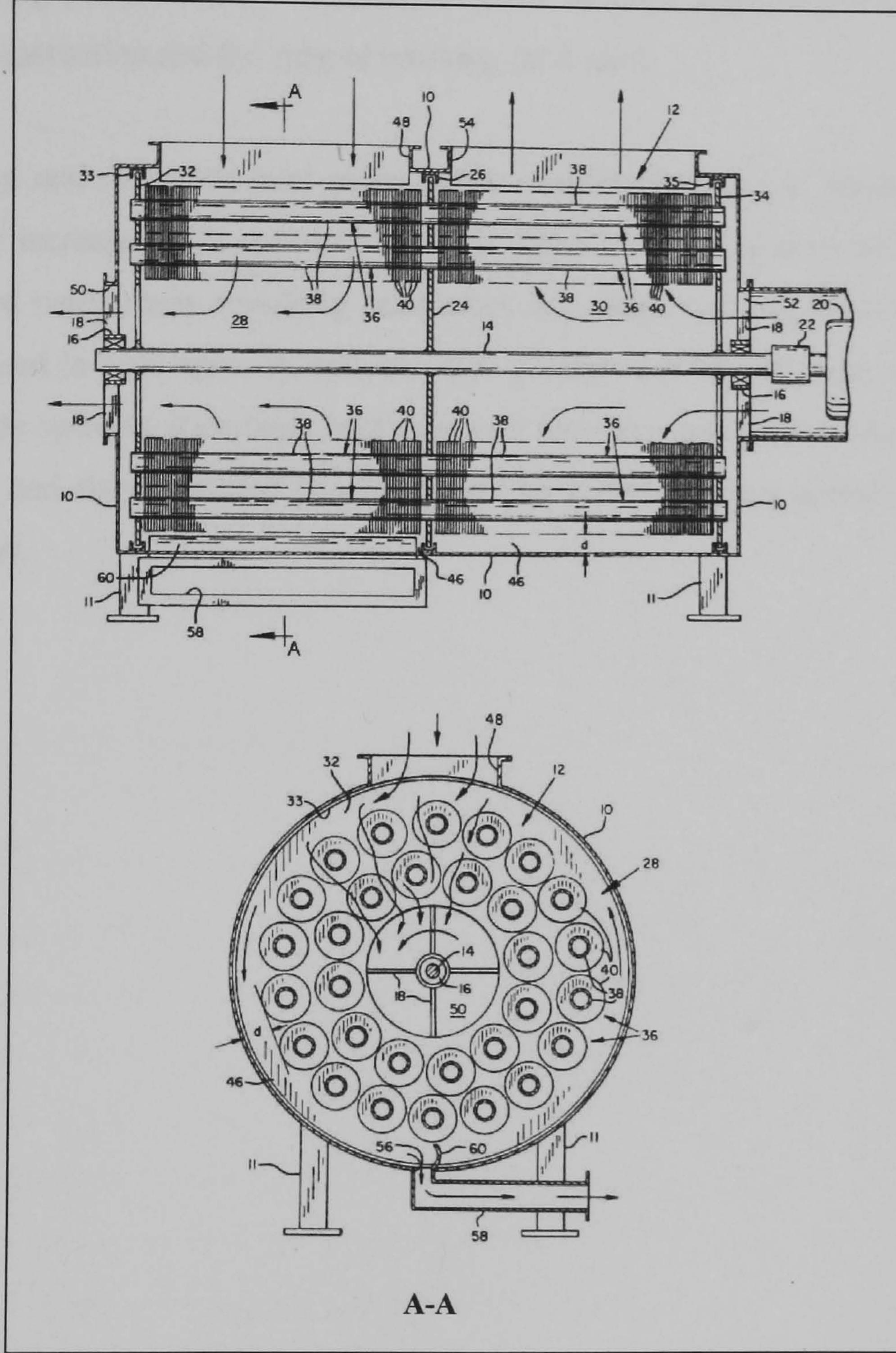


Figure 4.13: Self Cleaning Rotary Heat Exchanger [Pravda, 1987]

The device shown in Figure 4.13 was tested by the inventor with different contaminated air streams passing over the heat pipes. Tests were carried out using aerosols of different sizes including cornmeal flour, white flour and lint contaminated air. In all cases, the device performed well with no significant build up of contaminates on the heat pipes during extensive periods of use .

4.6 Conclusions

Heat pipes have high rates of heat transfer, they are compact, robust and have no moving parts. They can be used for many different applications by altering their construction and the type of working fluid used.

Rotating and revolving heat pipes utilise centrifugal forces to operate. Heat transfer increases with speed of rotation. High levels of heat transfer can be achieved using many revolving heat pipes in a single system. Revolving heat pipes used in heat recovery systems have greater heat transfer than stationary heat pipe systems. Revolving heat pipe heat recovery systems are less prone to fouling and they consume less fan power as pressure drops across them are very low.

5. HEAT PUMPS

Heat pumps are devices that absorb heat from a low temperature heat source and then discharge it to a sink at a higher temperature. The absorbed heat is upgraded by means of work or heat input to the system. The majority of heat pumps can be classified into two main categories, vapour compression systems and absorption systems. Vapour compression systems are driven by a compressor which inputs work into the system. Absorption systems use thermal energy to drive the system. As the work in this thesis is concerned with the development of a vapour compression heat pump system, this chapter only covers vapour compression systems. A full description of different heat pump cycles is given in the texts on heat pumps by Heap (1979) and Reay & MacMichael (1988).

5.1 The vapour compression cycle

The majority of heat pumps work on the principle of the vapour compression cycle. In its simplest form, a vapour compression system consists of an evaporator, condenser, compressor and expansion device. These components are connected to form a closed system, as shown in Figure 5.1. The system contains a charge of volatile liquid, known as the working fluid or refrigerant, which circulates through the components. Figure 5.2 illustrates the vapour compression cycle on a pressure/enthalpy diagram. The refrigerant enters the evaporator as a liquid/vapour mixture at low temperature and pressure (E). In the evaporator heat is transferred from a low temperature heat source to the refrigerant. The heat causes the refrigerant to evaporate (E-A) and it leaves the evaporator as a low pressure vapour (A). The pressure and temperature of the vapour are then raised by the compressor (A-B). The vapour is superheated at point B as it is above its saturation temperature for its pressure (B). The high temperature refrigerant vapour then enters the condenser where it is cooled by the rejection of its latent heat to a high temperature heat sink (C-D). High pressure and temperature liquid refrigerant then exits the condenser and passes through the expansion device, where it is expanded at constant enthalpy to the temperature and pressure of the evaporator thus completing the cycle (D-E).

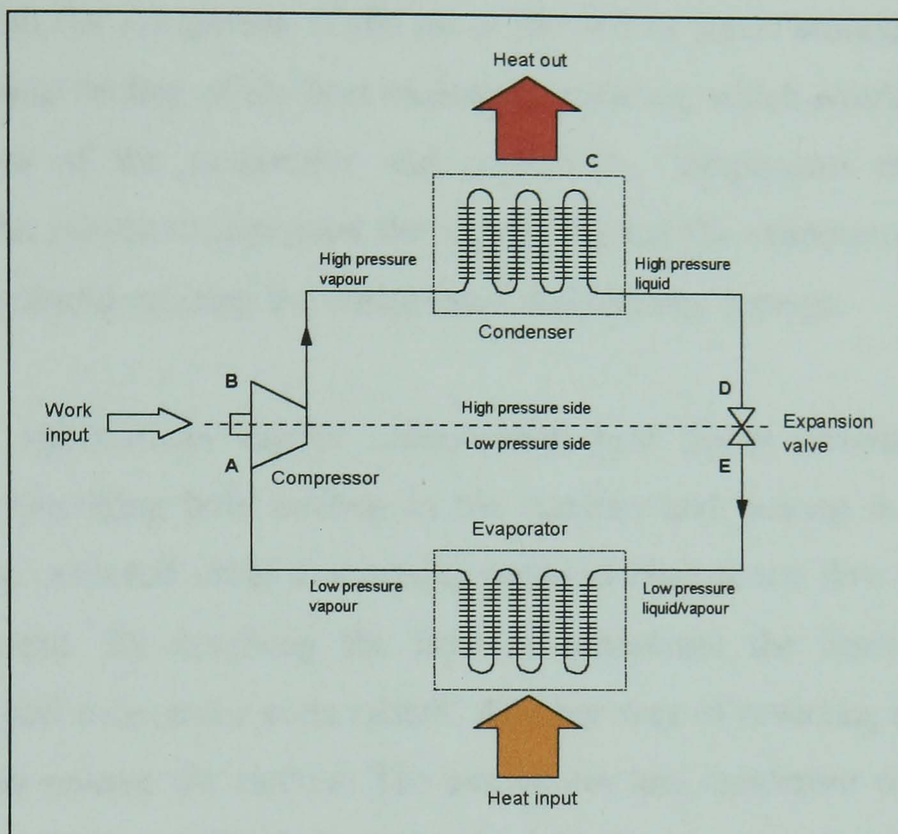


Figure 5.1: Basic Vapour Compression Heat Pump

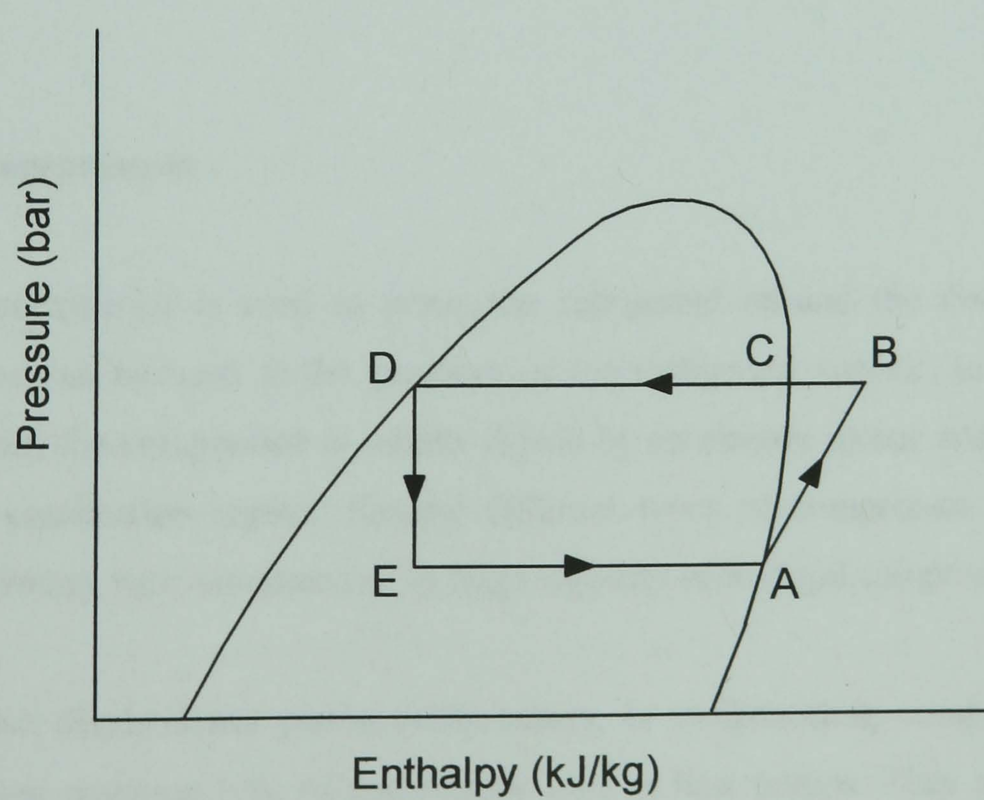


Figure 5.2: Vapour Compression Cycle

In practice, vapour compression cycles may have additional components. If an oil lubricated compressor is used, then an oil separator is included in the system. This is because the oil used to lubricate the compressor is slightly miscible with the refrigerant. If the oil is allowed to travel around the system it would cause fouling of the heat exchanger surfaces, which would reduce the effectiveness of the evaporator and condenser. Components may also be added to the system to superheat the vapour leaving the evaporator. This is to prevent any liquid entering the compressor and causing damage.

In HVAC applications vapour compression heat pump systems are often reversible, providing both cooling in the summer and heating in the winter. This can be achieved using a reversing valve to change the flow direction of the refrigerant. By reversing the flow of refrigerant the functions of the condenser and evaporator coils switch. Another way of reversing a heat pump system is to reverse the airflow. The evaporator and condenser coils function the same in summer and winter. However, the ducts are arranged so that the supply air is passed over the condenser in the winter and the evaporator in the summer. This provides heating and cooling respectively.

5.2 Compressors

The compressor is used to pump the refrigerant around the system and to produce an increase in the pressure of the refrigerant vapour. In heat pump systems, the compressor is usually driven by an electric motor and sometimes by a combustion engine. Several different types of compressor exist, from small rotary vane compressors to large capacity centrifugal compressors.

Positive displacement piston compressors, or reciprocating compressors, are the most common type of compressor used in heat pumps. They are available in a very large range of sizes to accommodate many different applications. Reciprocating compressors generally provide a high pressure difference at moderate flow rates. Figure 5.3 illustrates the operating principle of reciprocating compressors. In practice, compressors with multiple cylinders are used.

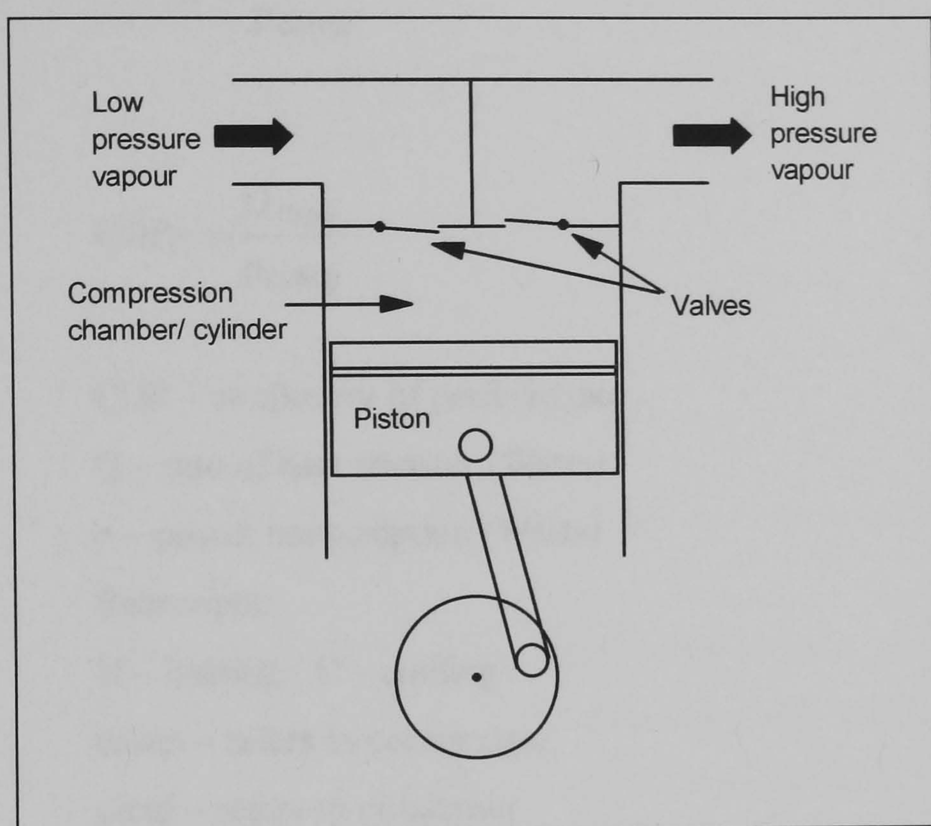


Figure 5.3: Reciprocating Compressor Principle

5.3 Heat pump performance

The basic measure of performance of a heat pump (or refrigerator) is the coefficient of performance (COP). The COP of a system is defined as the useful amount of heating or cooling relative to the rate of energy input to the system. In a vapour compression system the energy input to the system is that supplied to the compressor. The theoretical COP can be found from equation 5.1, where H(A), H(B) and H(D) are enthalpy values at points A, B and D, respectively on Figure 5.2.

$$COP = \frac{H(B) - H(D)}{H(B) - H(A)} \qquad \textbf{(5.1)}$$

In practice, the heating and cooling COPs can be found from the following equations.

$$COP_H = \frac{Q_{cond}}{p_{comp}} \tag{5.2}$$

$$COP_C = \frac{Q_{evap}}{p_{comp}} \tag{5.3}$$

Where: COP – coefficient of performance
Q – rate of heat transfer (Watts)
p – power consumption (Watts)
Subscripts:
H – heating / C – cooling
comp – refers to compressor
cond – refers to condenser
evap – refers to evaporator

Typical COPs for electrically driven heat pumps are between 2.5 and 5.0 [IEA Heat Pump Centre, 1999]. The COP varies with the evaporating and condensing temperatures. The smaller the temperature difference, the better the COP. In packaged heat pump systems, the COP may also take into account auxiliary power inputs, such as fans, which will have the effect of slightly lowering the COP.

5.4 Heat sources

The thermal and economic performance of a heat pump is largely dependent on the characteristics of heat source used. A good heat source should be abundantly available, non-corrosive and have a high and stable temperature. It should also have good thermophysical properties such as high heat capacity. Table 5.1 presents the temperature range for commonly used heat sources.

Heat Source	Temperature Range(°C)
Ambient air	-10-15
Exhaust air	15-25
Ground water	4-10
Lake water	0-10
River water	0-10
Sea water	3-8
Rock	0-5
Ground	0-10
Waste water and effluent	>10

Table 5.1: Commonly Used Heat Sources [IEA Heat Pump Centre, 1999]

Ambient air is freely available and is the most common heat source used for heat pumps. It is, however, often at low temperatures and is prone to large temperature swings which will reduce performance. Ground sources and bodies of water are less susceptible to changes in weather. Their temperatures fluctuate by only a few degrees over a heating season which make them ideal heat sources. Exhaust air also makes an ideal heat source. It tends to be at fairly high temperatures which do not substantially fluctuate. When using exhaust air as a heat source, the heat pump recovers heat from this air to provide water and/or space heating. Continuous operation of the ventilation system is required to obtain heat from the exhaust air.

5.5 Refrigerants

A good refrigerant should have a high value of latent heat over the evaporation and condensing temperature range of the system. High latent heat values give good theoretical COPs with low mass flow rates. Ideally the refrigerant should have a relatively low condensing pressure to reduce the strength requirements of the condenser and seals. Refrigerants with low viscosities are also desirable to reduce pipe work losses. The thermal conductivity of refrigerants should be high, enabling good heat transfer. Refrigerants should be chemically stable, have low toxicity and be compatible with components in the system and compressor oil.

At present many refrigeration systems use existing freon refrigerants. These refrigerants have been in use since the 1930s, when Thomas Midgley Jnr of the Frigidaire division of General Motors developed trichlorofluoromethane and dichlorofluoromethane now known as the chlorofluorocarbons (CFCs) R11 and R12. A variety of CFCs exist and they have been widely used due to their good thermophysical properties. However, in 1974 Rowland and Molina of the University of California claimed that CFCs were damaging the ozone layer. These claims have been supported by subsequent research and it is now widely accepted that the release of CFCs into the atmosphere is destroying the ozone layer [UNEP, 1992].

The ozone layer surrounds the Earth and is situated at high altitude in the stratosphere. It reduces the amount of UV-b radiation reaching ground level. UV-b radiation is damaging to life, causing skin cancer and eye damage in humans. The chlorine present in CFCs causes the depletion of the ozone layer. One chlorine atom can destroy 100,000 ozone molecules [Pita, 1998]. The ability of a substance to deplete the ozone layer is called its ozone depletion potential (ODP). ODPs are quoted relative to the ODP of CFC-11 and CFC-12 which have the highest ODP equal to 1.

Figure 5.4 illustrates the hole over Antarctica in the autumn of 1998. The colours indicate the concentration of ozone (in Dobson units) present in the stratosphere. It is a satellite shot taken by NASA. The hole that year was 10.4 million square miles, at that time the largest on record [NASA, 1999]. Not only do CFCs have large ozone depletion potential, they also have large global warming potential (GWP). The GWP of a chemical represents how much a given mass of the chemical contributes to global warming, over a given time period, compared to the contribution of the same mass of carbon dioxide. Carbon dioxide's GWP is defined as 1.0. The GWPs of R11 and R12 are 4000 and 8500 respectively.

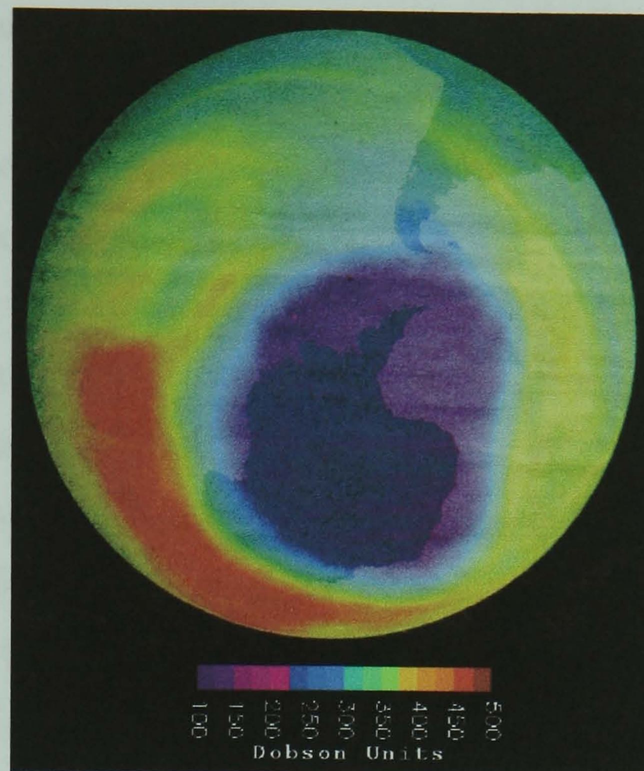


Figure 5.4: Ozone Depletion Over Antarctica

The global release of CFC refrigerants to the environment was estimated to be 1700 tonnes in 1990 [UNEP, 1991]. Since then, the ozone depletion problem has been addressed by controlling the use of CFCs. By the beginning of 1997 the Montreal Protocol, on the phase out of CFCs, had been endorsed by 161 countries.

The phase out of CFCs has led to a search for viable alternative refrigerants. New refrigerants have been formulated based on hydrochlorofluorocarbons (HCFCs) such as R22 and hydrofluorocarbons (HFCs) such as R134a. However, both these groups of refrigerants still have associated environmental impacts. HCFCs have ozone depletion potential and significant global warming potential. Because of this the phase out of HCFCs is due in the near future. HFCs are chlorine free refrigerants with zero ODP. However, HFCs do still contribute to global warming and therefore cannot be considered as being truly environmentally friendly. Not only are HFCs unattractive alternative refrigerants due to their high GWPs, they are also expensive to manufacture and are incompatible with existing compressor lubricants.

A group of refrigerants that have zero ODPs and very low GWPs are naturally occurring refrigerants such as, water, carbon dioxide, ammonia and hydrocarbons. There is growing interest in using these refrigerants as long term replacements for CFCs and HCFCs. Water is an excellent refrigerant with good thermodynamic properties. It has been used successfully in large scale high temperature industrial heat pumps. However, the major disadvantage with water as a refrigerant, is that it has a low volumetric heat capacity. As a result, vapour compression systems using water require large expensive compressors. Carbon dioxide (CO_2) is potentially a good natural refrigerant. It is non-toxic, non-flammable and has good material compatibility. Unfortunately, it is non-condensable at typical condenser temperatures and pressures and the theoretical COP of conventional heat pump cycles with CO_2 is poor. Ammonia is a very good refrigerant but cannot be used in existing systems. It is flammable, toxic and highly corrosive to alloys of copper. Its use is restricted to new refrigeration equipment and it tends to be applied in medium to large refrigeration and cold storage plants. Codes and regulations are in place to deal with the toxicity and flammability of ammonia.

Hydrocarbons are a group of refrigerants with good thermodynamic properties and material compatibility. They are by-products of petrochemical processes, and are therefore inexpensive and widely available. Hydrocarbons are flammable and, until recently, worries over their flammability have restricted their use. However, in 1995 the UK safety standard for refrigeration systems, BS 4434, was revised [British Standards Institute, 1995]. The revisions allowed natural refrigerants, like hydrocarbons, to be used in a wide range of applications. By designing systems to the standard set out in BS 4434, the flammability of hydrocarbons can be easily mitigated through adequate safety measures. The key requirement for domestic systems is that the charge should be less than 1.5kg and all sparking sources adequately sealed. This is easily achievable in small domestic heat pump systems. The mass of hydrocarbons required for a heat pump system is typically less than half that of R12 [Richardson, 1995].

Studies by Richardson (1995), Halozan and Granryd have shown hydrocarbons perform well in refrigeration cycles. Heat pumps and air conditioning units using propane have been shown to use 10-20% less energy than equivalent systems charged with R12, R22 or R134a. It has been stated that the heat transfer coefficients are up to 25% higher in evaporators when using hydrocarbons rather than R22 [Butler & Reay, 1999].

Across Europe, hydrocarbons are now accepted as a viable alternative refrigerant for heat pump and refrigeration systems with small refrigerant charges. There are over 10 million hydrocarbon refrigerators in the world today. The German domestic refrigerator market has converted to 100% hydrocarbon use [Greenpeace, 1999]. In Sweden, approximately 1,500 exhaust air heat pump systems using propane as a working fluid were sold in 1998. Approximately 20% of residential heat pump sales in Germany are systems charged with propane or propylene (300-400 units per year) and all German heat pump manufacturers offer hydrocarbons as working fluids [IEA Heat Pump Centre, 1999].

Hydrocarbon refrigerants are sold in the UK by Calor under the generic name CARE. The properties of CARE refrigerants are given in Table 5.2. Except for CARE 10, these refrigerants can be used as replacement refrigerants without the requirement for system modifications. The volumetric cooling capacity of CARE 10 (isobutane) is smaller than the capacities of CFC12 and HFC134a, which means that compressors with greater swept volumes are required.

Refrigerant	Composition	Chemical formula	Boiling point*	Replacement
CARE 10	R600a (isobutane)	$CH(CH_3)_3$	-11.7	R12, R134a
CARE 30	Blend of R600a (isobutane) and R290 (propane)	$CH(CH_3)_3+CH_3CH_2CH_3$	-31.7	R12, R134a
CARE 40	R290 (propane)	$CH_3CH_2CH_3$	-42.1	R502, R22, R404a, R407C, R507
CARE 50	Blend of R290 (propane) and R170 (ethane)	$CH_3CH_2CH_3+C_2H_6$	-49.1	R502, R22, R404a, R407C, R507

* at standard atmospheric pressure (101.325kPa)

Table 5.2: CARE Refrigerant Range [Calor Gas]

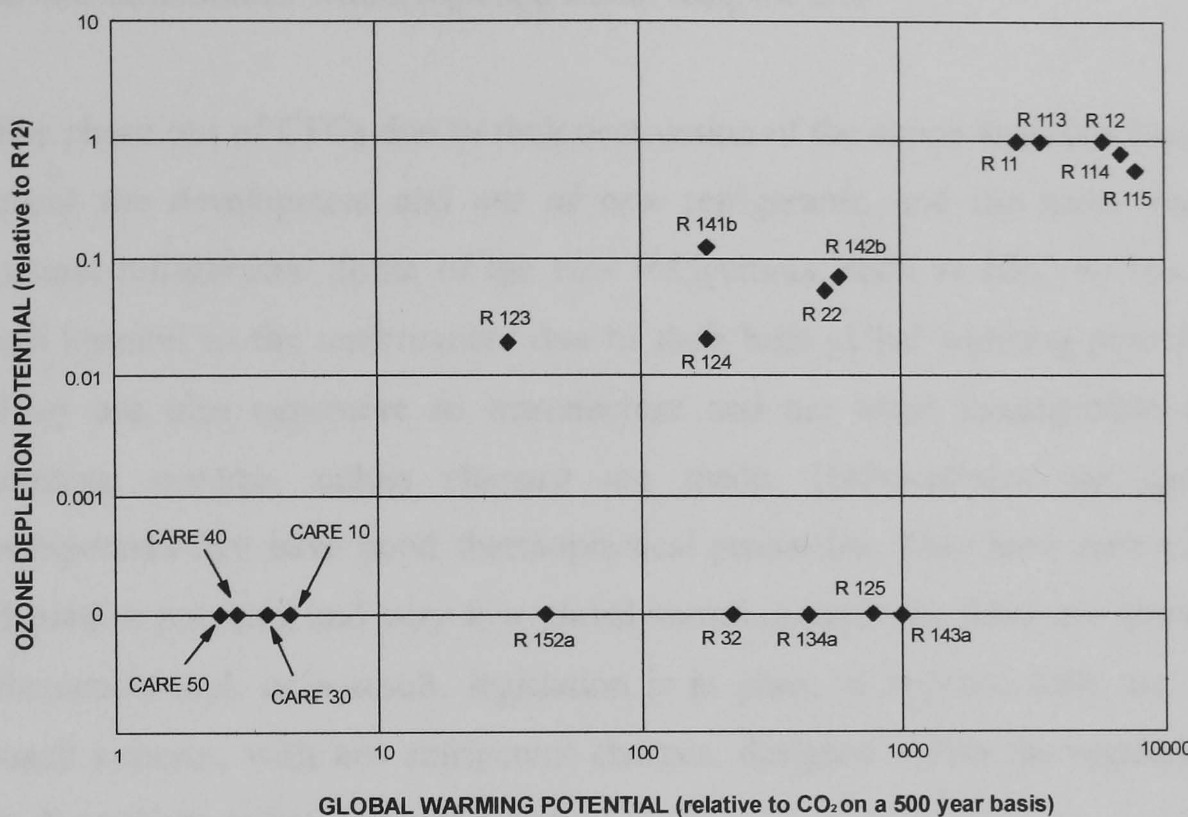


Figure 5.5: Environmental Effect of CARE Products Relative to Other Refrigerants [Calor Gas]

Figure 5.5 compares the global warming and ozone depletion potentials of the CARE range of refrigerants with other refrigerant types. One of the objectives of the research contained within this thesis is to use a natural refrigerant, such as a hydrocarbon, in a prototype ventilation heat recovery heat pump.

5.6 Conclusions

The heat pump is a refrigeration system that can be used for both heating and cooling. Because a heat pump uses electricity to extract heat energy from one place and move it to another, rather than as a direct means of generating heat, efficiencies are very high. A typical system is capable of providing a heating effect equivalent to about three times the quantity of energy needed to drive it. When using exhaust air as a heat source, energy that would otherwise be wasted can be reclaimed. For good heat pump performance the heat source should be abundant with a high and stable temperature.

The phase out of CFCs due to their destruction of the ozone layer has brought about the development and use of new refrigerants, and the wider use of natural refrigerants. Some of the new refrigerants, such as HFC R134a, are still harmful to the environment due to their high global warming potentials. They are also expensive to manufacture and are often incompatible with existing systems, unless changes are made. Hydrocarbons are natural refrigerants that have good thermophysical properties. They have zero ozone depletion potential and very low global warming potential. They are however flammable and, as a result, legislation is in place to regulate their use. For small systems, with low refrigerant charges, designed within the regulations, hydrocarbons make ideal refrigerants.

6. REVOLVING HEAT PIPE MVHR SYSTEM

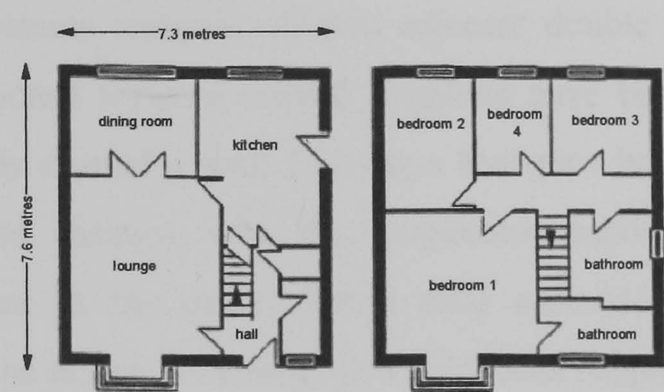
This chapter covers the design and construction of several prototype test rigs developed to provide mechanical ventilation with heat recovery. The MVHR prototype systems developed use revolving heat pipes to both impel air and transfer heat. It is not the aim of this work to explore further the theory of the highly complex mechanisms which operate with revolving heat pipes, but to utilise the established theory for use in a revolving heat pipe MVHR system.

The main aims of the work are to minimise the overall energy consumption of dwellings, minimise maintenance and provide fresh air at an appropriate rate. The work has sought to incorporate several features which have been shown by previous work to enhance performance. The features are the enhanced heat transfer of revolving heat pipes, both externally and internally, together with the fan effect and low fouling characteristics. These features were covered in the previous chapter.

6.1 System requirements

MVHR systems should provide a low velocity, constant flow of fresh air for the home while recapturing 50% to 80% of the exhaust air heat. To maintain general indoor air quality in homes, ASHRAE recommend a minimum of 0.35ach and the UK Building Regulations, under the guidance of the BRE, recommend 0.5ach to 0.7ach less background natural infiltration (see Chapter 3.2). There will always be some natural infiltration in homes since it is highly impractical to achieve a fully airtight dwelling. As previously stated in Chapter 3, the BRE have established that, for economic use of MVHR systems, the current practical limit for background infiltration should be about 0.2 ach [BRE, 1994]. Therefore, to conform to the UK Building Regulations, MVHR systems must provide 0.3 ach to 0.5 ach assuming 0.2 ach background infiltration. The ASHRAE recommendation of 0.35 ach is within this range and is being used for the purpose of this work.

At present UK homes equipped with MVHR systems tend to be large detached houses. In order to size such a system, the volume of a typical detached home should be known. Figure 6.1 shows a standard design with average characteristics for a new build four bedroom detached house. This design was produced jointly by the BRE and the House Builders Federation (HBF) [Gillott, 1994].



Total floor area:	113.0 sqm
Height:	5.0 m
Roof area:	56.5 sqm
Ground floor area:	56.5 sqm
Area of openings - Windows:	16.6 sqm
Doors:	3.8 sqm
Total:	20.4 sqm
Net exposed wall area:	133.6 sqm
Opening area/Total floor area:	18.1%

Figure 6.1: BRE/HBF Standard New Build Detached House

In order to maintain 0.35ach for the BRE/HBF standard new build detached house, a whole house mechanical ventilation system should supply fresh air at the following calculated rate:

$$\begin{aligned} \text{Flow rate} &= \text{air change rate} \times \text{total floor area} \times \text{height} \\ &= 0.35 \times 113 \times 5 \\ &= 197.75 \\ &\approx \mathbf{200m^3/hr} \end{aligned}$$

The MVHR system must therefore be able to provide $200m^3/hr$ continuous ventilation and $250m^3/hr$ (+25%) for periods of boost ventilation. These figures are typical of current systems on the market. The system should also be capable of overcoming system losses due to the ductwork of approximately 70-100Pa.

6.2 Prototype construction

The prototype systems comprise of two adjacent double inlet centrifugal fan casings. The standard forward curved impellers have been replaced by heat pipes fixed radially around a shaft forming a heat pipe bundle. The heat pipes span the two fan casings, with the evaporator sections in one and the condenser sections in the other. Warm stale exhaust air passes over the evaporator sections in one fan casing, and cold fresh supply air passes over the condenser sections in the other casing. To stop air leakage and cross contamination between the two air streams, rotating plate seals are used where the heat pipes pass between the casings. The heat pipe bundle is rotated by a motor via a belt and pulleys. When the heat pipes rotate they impel the air in two separate air streams. Heat transfer between the two air streams takes place using the heat pipes. Double inlet fan casings were used due to their large width relative to impeller diameter. A wide fan casing houses longer heat pipes with greater heat transfer surface. Figure 6.2 illustrates the basic configuration and operation of the revolving heat pipe MVHR system and Figure 6.3 illustrates the system components. Appendix B gives details of manufactures and suppliers used during prototype construction.

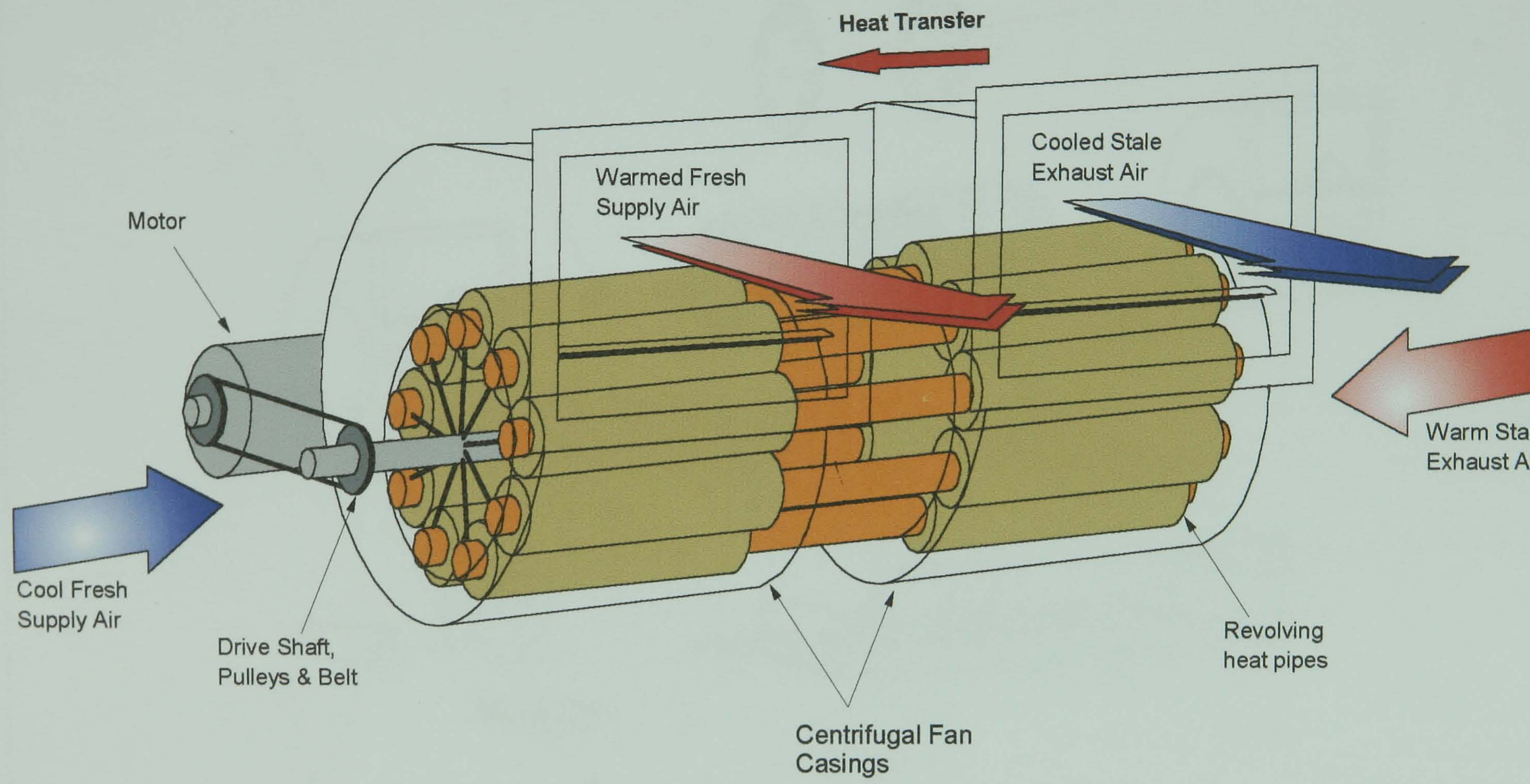


Figure 6.2: Revolving heat pipe MVHR system

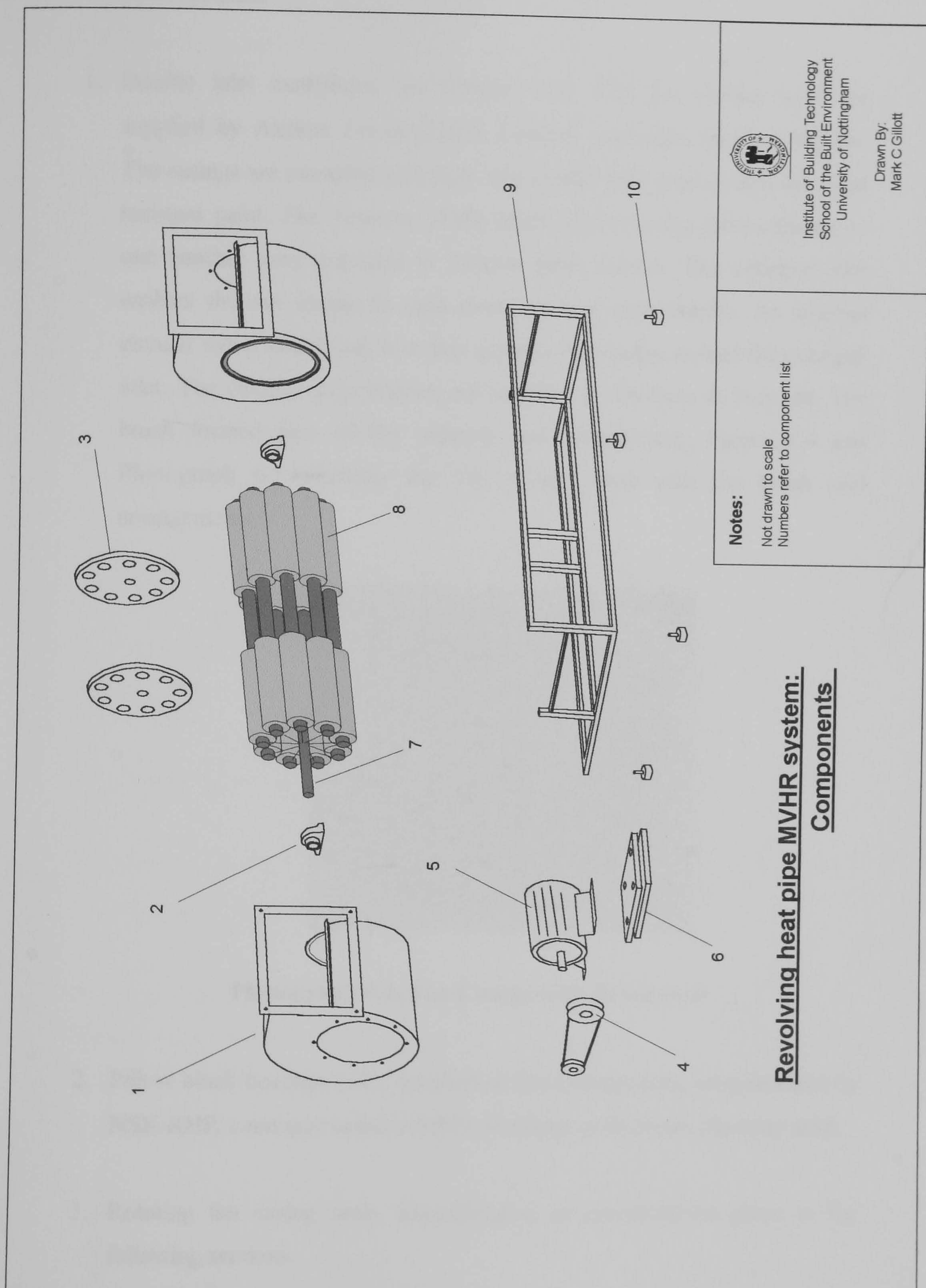
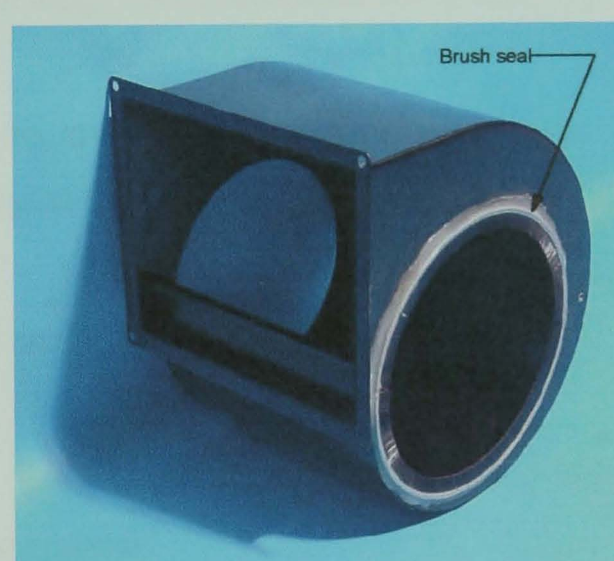


Figure 6.3: Revolving Heat Pipe MVHR System Components

Component List:

1. Double inlet centrifugal fan casings (×2). The fan casings used are supplied by Airflow Developments Limited, catalogue number 90G2W. The casings are manufactured from zinc coated mild steel coated with heat resistant paint. The diameter of the inlets on the casing sides adjacent to one another were enlarged to 270mm from 238mm. The enlarged inlet enables the fan casing to pass over the heat pipe bundle. An internal circular nylon strip brush was then glued to the casing around the enlarged inlet. The circular strip brushes are supplied by Dawson & Son Ltd. The brush formed part of the rotating seal arrangement. Figure 6.4 and Photograph 6.1 illustrate the fan casings used with the brush seal arrangement.



Photograph 6.1: Fan Casing with Brush Seal

2. Pillow block bearings (×2), supplied by RS Components, manufactured by NSK-RHP, catalogue number NP25. Bearings to fit 25mm diameter shaft.

3. Rotating fan casing seals. Manufactured to specifications given in the following sections.

4. Drive belt (×1) and pulleys (×2), supplied by Fenner Power Transmission. The drive belt is a 'V' belt, code SPZ 1060. The pulleys are both 80×1

1210 SPZ Pulleys. The drive pulley has a Taper Lock bush 1210×24 and the driven pulley has a Taper Lock bush 1210×25. Drive ratio is 1:1.

5. 1.5kW, 3-phase, 4-pole (max rpm = 1480), foot mounted motor, supplied by Fenner Power Transmission, catalogue number H1408. A 3-phase motor is used to provide good speed control. The motor is 1.5kW in size to overcome the large starting torque. Initially a 550W motor was used but this struggled to overcome the starting torque.

6. Motor slide rails, supplied by Fenner Power Transmission, catalogue number 076k 0000. The motor is mounted on the slide rails and can be moved to adjust the belt tension.

7. Drive shaft. 800mm long, 25mm diameter silver steel bar.

8. Heat pipe bundle to specifications given in the following sections. The heat pipes are insulated where they pass between the casings.

9. Frame constructed to specification shown in Figure 6.5.

10. Height adjustable feet. General purpose anti-vibration machine mounts supplied by RS components.

Photographs 6.2 to 6.5 illustrate the prototype test rig. Five variations of the system were tested to determine the best configuration. The variations included alterations to the fin type, number of heat pipes, working fluid fill ratio and geometry. The following design variations were built.

1. 9-pipe, wire fin, 10% water fill.
2. 9-pipe, wire fin, 20% water fill.
3. 9-pipe, plate fin, 10% water fill.
4. 9-pipe, plate fin, 10% water fill, tapered bundle.
5. Double row 24-pipe, wire fin, 10% water fill.

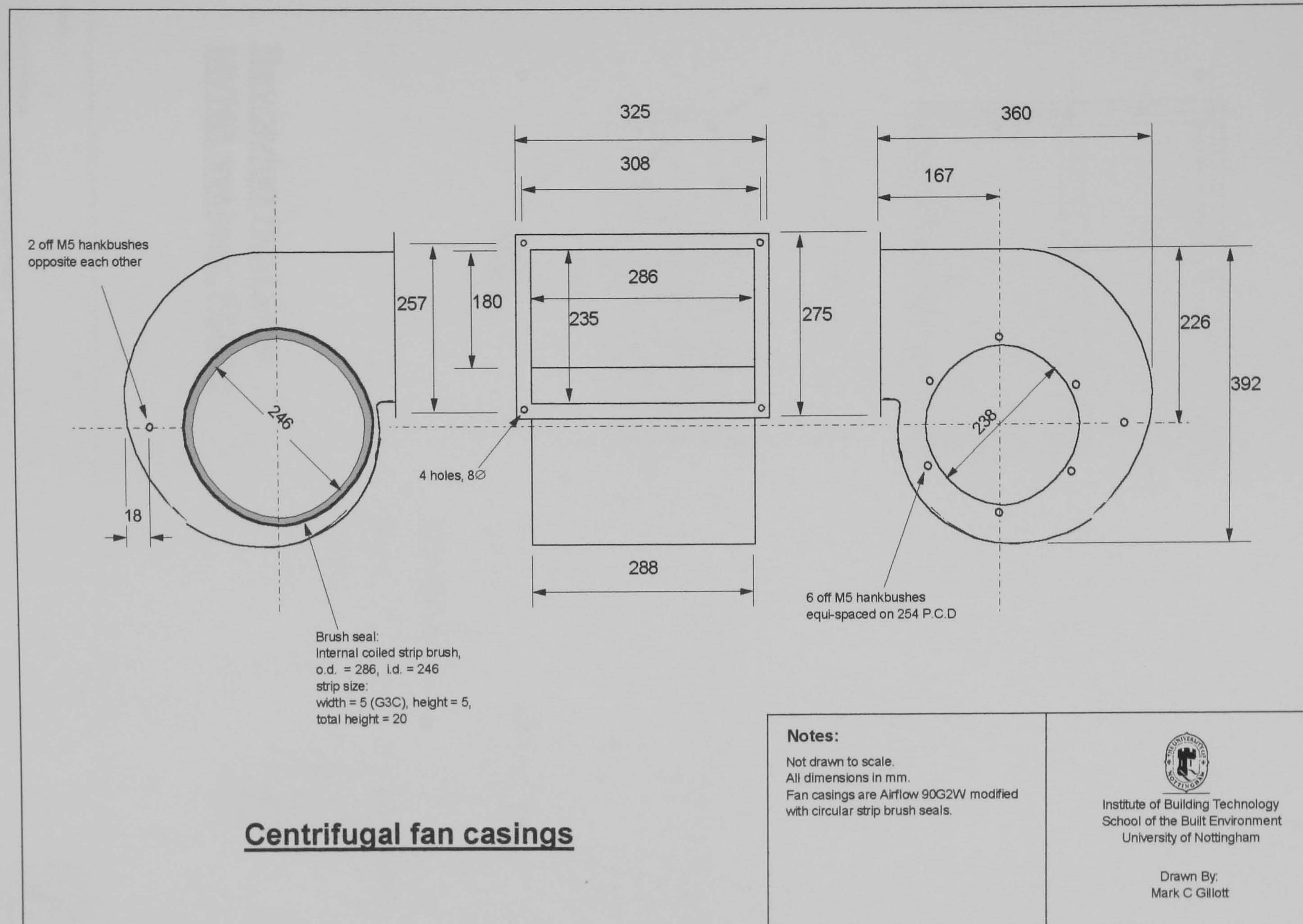
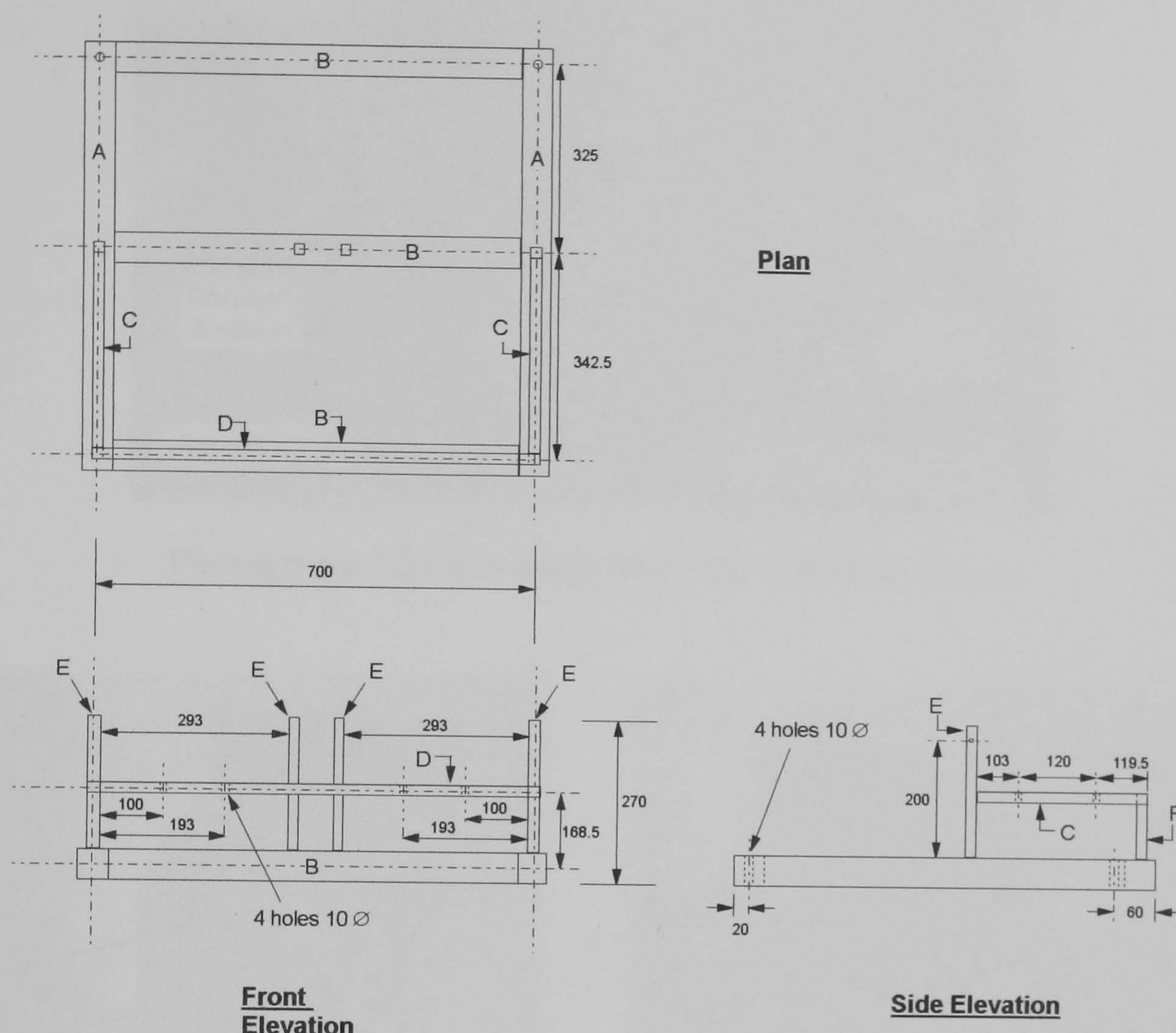


Figure 6.4: Centrifugal Fan Casings



Component List:

section	Ref.	Length & Quantity
40x40x3	A	707.5 (x2)
	B	660 (x3)
25x25x3	C	342.5 (x2)
	D	675 (x1)
	E	230 (x4)
	F	136 (x2)

Revolving heat pipe MVHR system: Frame

Notes:

Not drawn to scale.
All dimension in mm, unless otherwise stated.
Steel box section cut to above specification and welded together.

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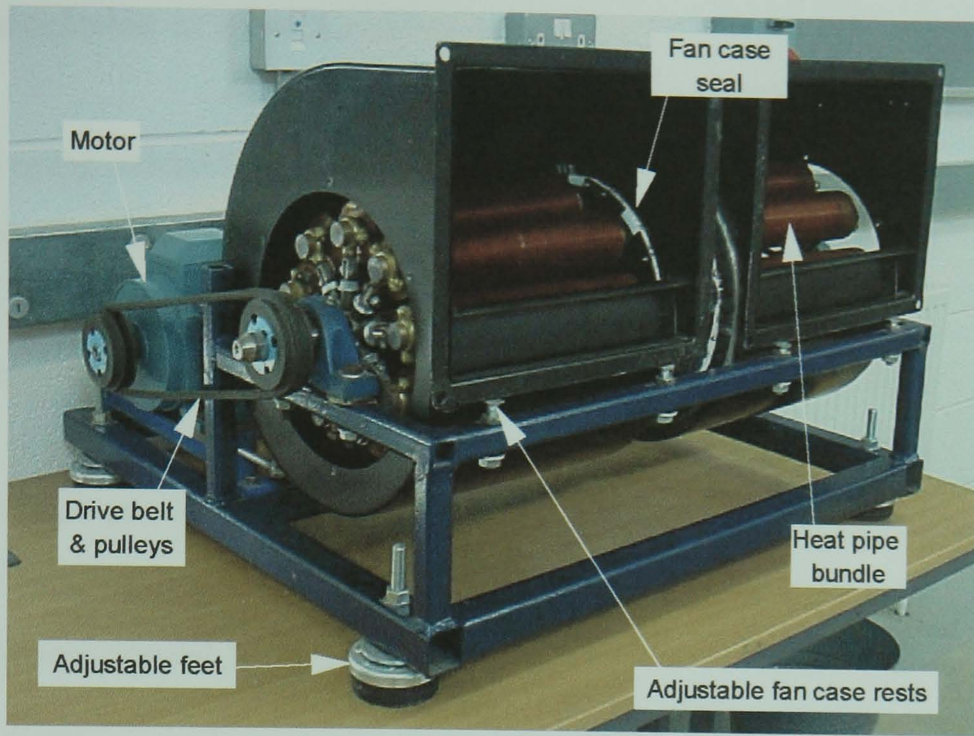
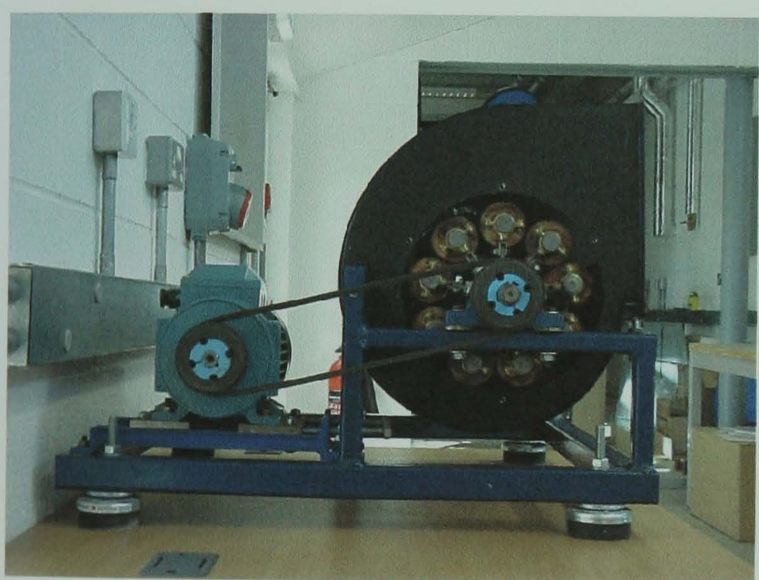
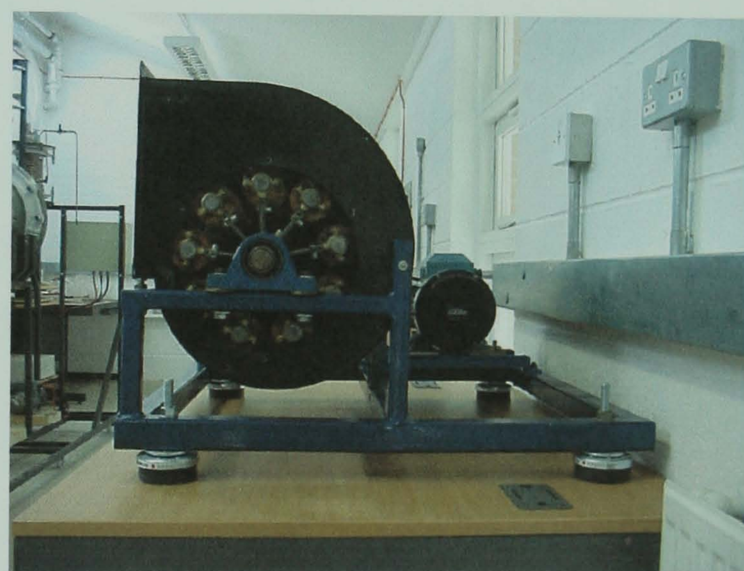
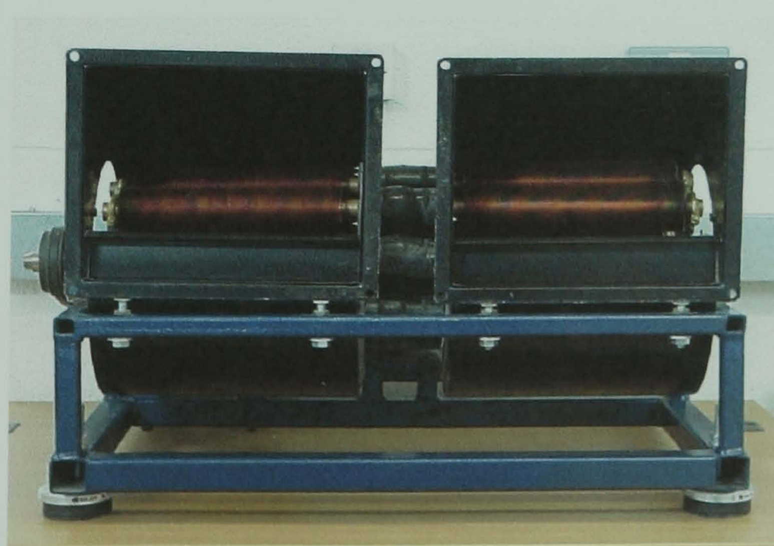
Figure 6.5: Revolving Heat Pipe MVHR System Frame



Photograph 6.2: Revolving Heat Pipe MVHR System

Photographs 6.3 and 6.4: Side Elevations



Photograph 6.5: Front Elevation

6.2.1 Calculation of theoretical heat transfer

The following calculates the maximum theoretical heat transfer the heat pipes should be capable of handling.

The following assumptions have been made:

Temperatures are taken for a winter's day where the outside temperature is 0°C and the inside temperature is 20°C. The heat exchanger is assumed to be 100% efficient, hence ΔT is 20 °C.

Using the sensible heat equation:

$$Q = m\,C\,\Delta T \tag{6.1}$$

Volumetric flow rate = 200m^3/hr

ρ = 1.205 kg/m^3 (density of dry air at 20 °C and 1 atmosphere)

C = 1.005 kJ/kgK (specific heat capacity of dry air at 20 °C and 1 atmosphere)

mass flow rate, m = 200 × 1.205

= 241 kg/hr

= 0.06694 kg/s

Q = 0.06694 × 1005 × 20

= **1346 Watts**

The heat pipes should therefore be capable of transferring 1346W (≈ 1.5kW).

6.2.2 Nine pipe non tapered rigs

This section covers the design of the nine pipe rigs where the heat pipes are horizontal to the axis of rotation. To fit within the fan casings, the external diameter of the heat pipe bundle was limited to 260mm with an overall length of 660mm. A large number of geometric configurations are possible within these limits. However, 22mm copper tube was used for the pipes due to its availability, low cost and good thermal conductivity.

Three variations were built for testing. The first configuration consisted of 9 660mm long, 22mm diameter heat pipes with 19mm high copper wire finning. Copper wire finning was used due to its good heat transfer characteristics. Wire finning induces turbulence and breaks down boundary layers, thereby increasing heat transfer. In conventional applications, wire finning may not be used as it has large pressure drops over its surface, thus incurring fan power penalties. This is not a problem for its use here, as the heat pipes are used both as the impeller and heat exchanger. Wire finning consists of a series of elongated wire loops, spirally wound on to the tube wall and held in position with a binding wire at the base of the loops. The loops and binding wire are then soft soldered to the tube wall to give a metallic bond between the wire fins and the tube [Tube Fins Ltd.]. The wire finning was produced at a maximum density in order to achieve the greatest area of heat exchanger. Figure 6.6 illustrates a section through a wire finned tube.

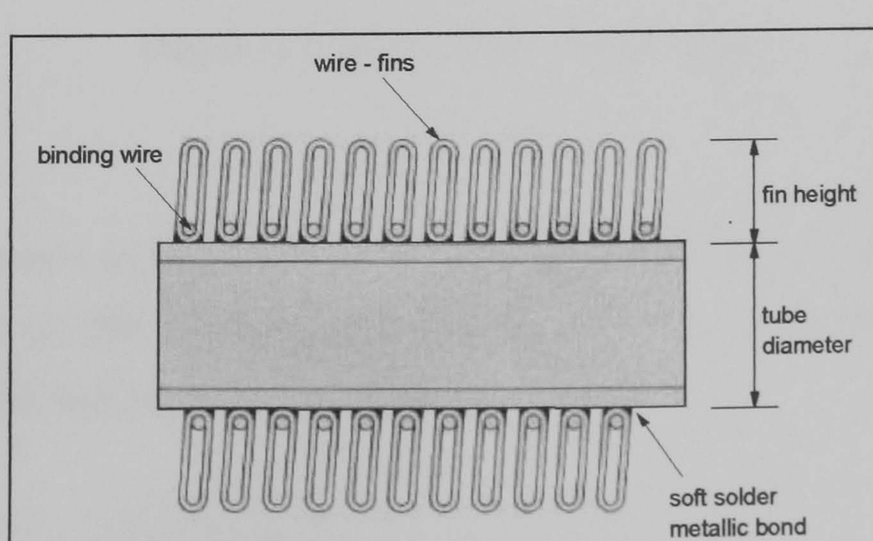


Figure 6.6: Wire Finning

The heat pipes were designed by Isoterix Limited. They supplied 9 heat pipes for assembly into the prototype rig where they would revolve eccentrically 100mm from the axis of rotation. Collectively the heat pipes were to be capable of transferring at least 1500 Watts. Each pipe was charged with 10% by volume of water. Water was used as the working fluid due to its good thermophysical properties. The quantity of charge was influenced by past research which was reviewed in Chapter 4. Isoterix supplied the pipes with wire gauze capillary structures to work in conjunction with the rotational forces in returning the condensate. A double layer of 150μm phosphor bronze screen mesh was used in each heat pipe. The 9 heat pipes weigh 13.8kg. Figure 6.7 illustrates the heat pipe design.

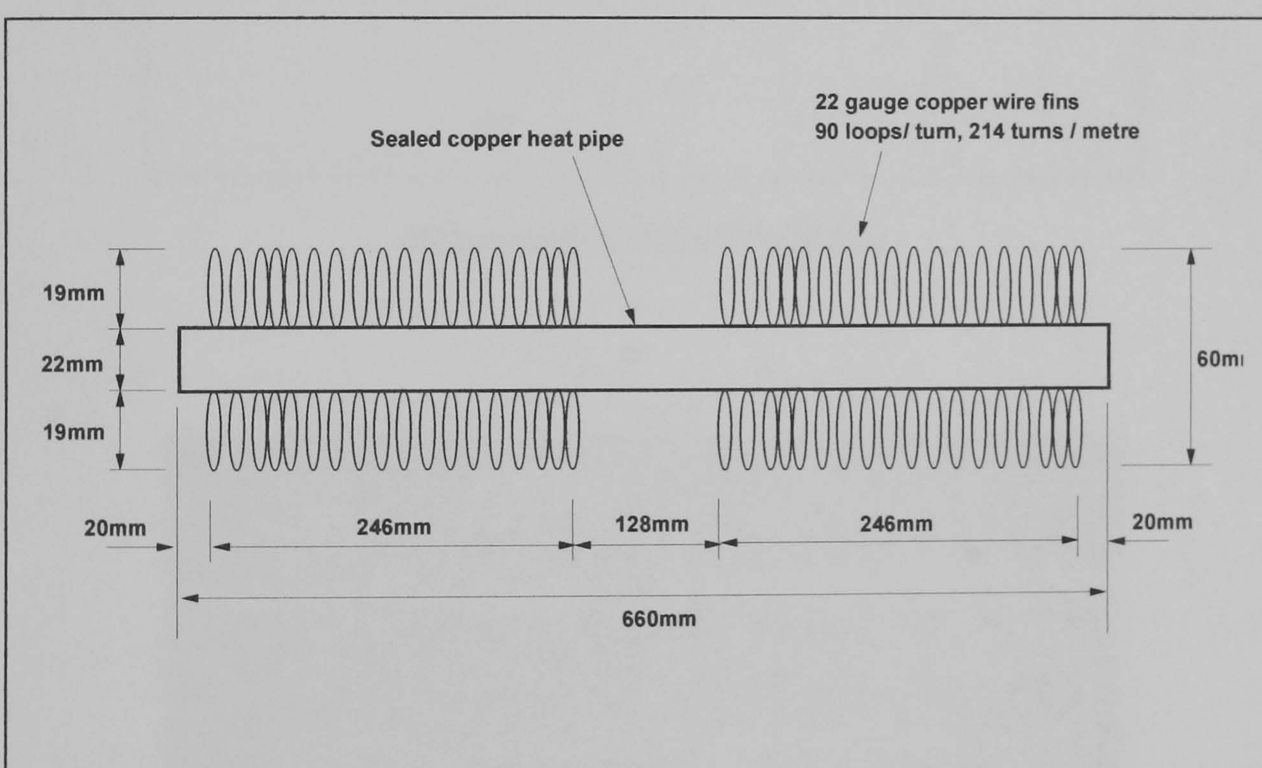


Figure 6.7: 9-Pipe Heat Pipe Design

The arrangement of heat pipes is illustrated in Figure 6.8. The heat pipes are equi-spaced on 200 p.c.d. Photograph 6.6 shows the evaporator/condenser sections of the heat pipes.

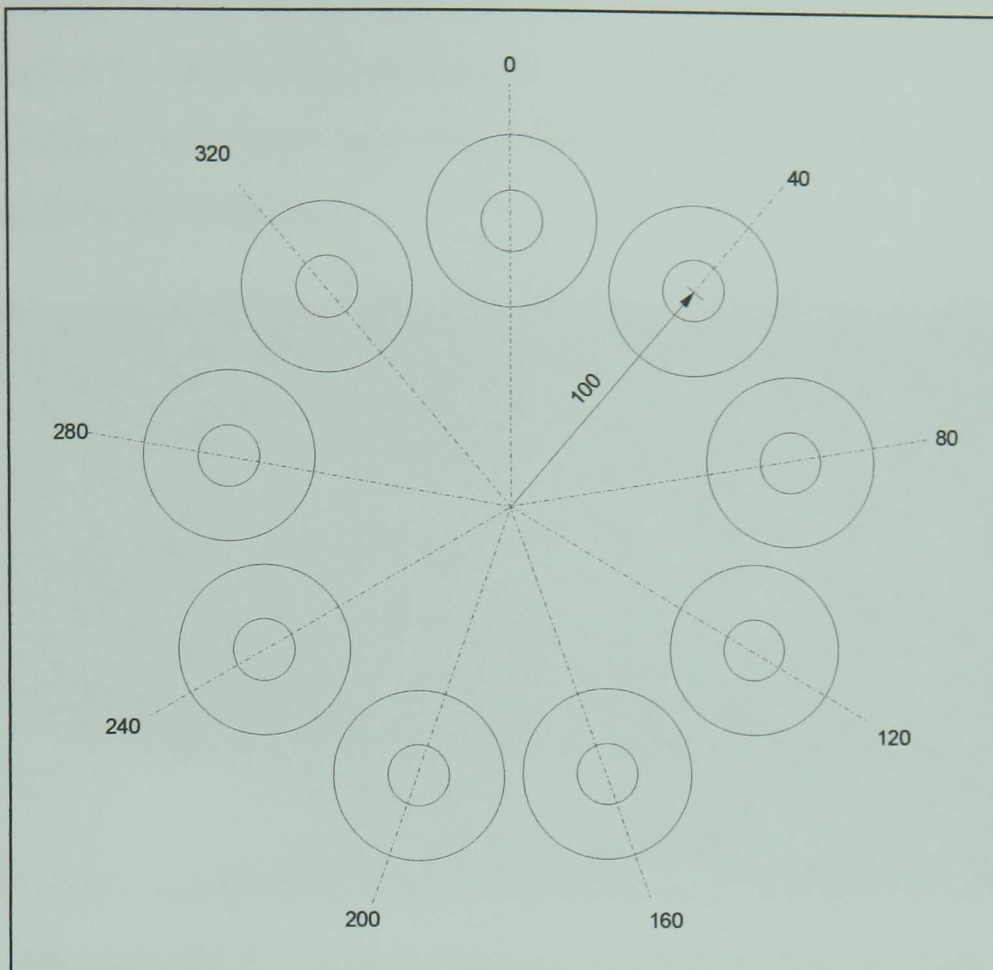
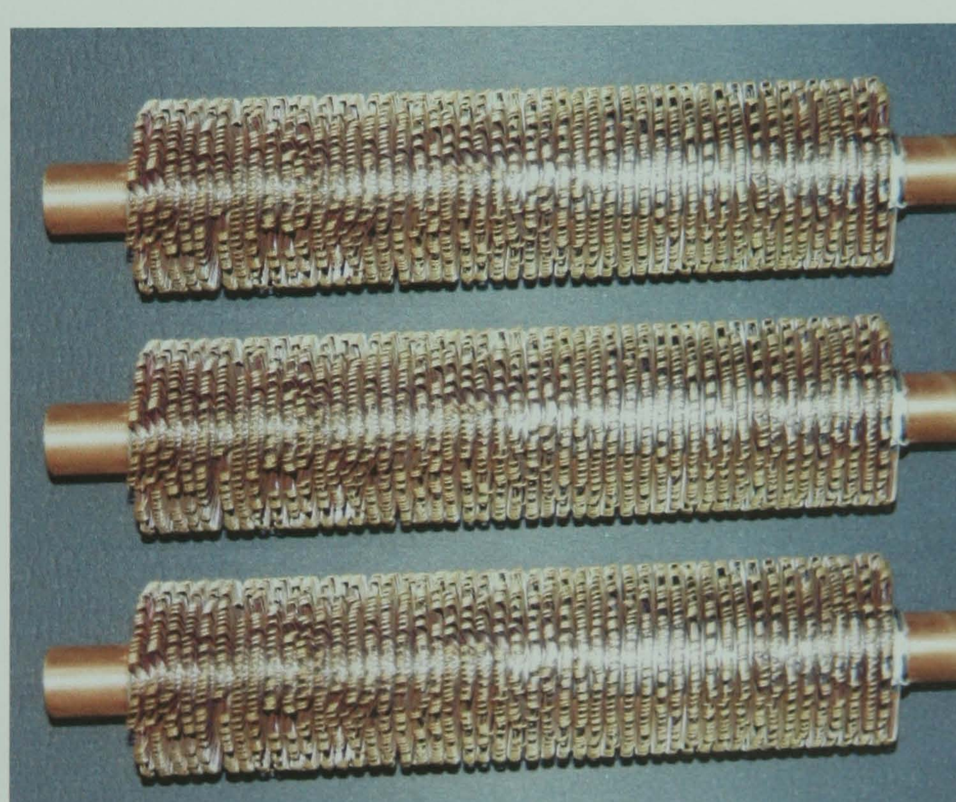
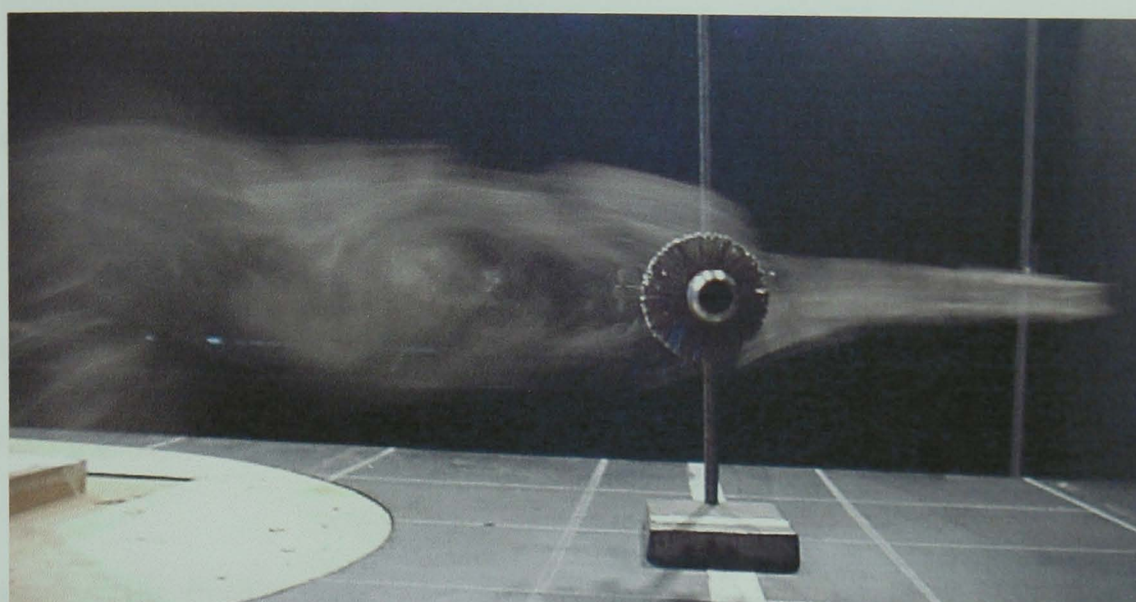


Figure 6.8: 9-Pipe Geometry



Photograph 6.6: Wire Finned Heat Pipes

The turbulent nature of air passing over wire finned tube is shown in Photograph 6.7. The photograph shows smoke passing over the tube in a wind tunnel. Eddies and swirls can clearly be seen down stream of the tube, indicating induced turbulence.



Photograph 6.7: Smoke Patterns Over Wire Finned Tube

Calculation of surface area of wire finning used:

Specification - 22 gauge wire fins (d = 0.711mm ∅), b = 19mm, at 90 loops/turn and 214 turns/metre on 9 pipes (diameter, D = 22mm), finned length, l = 246mm.

Surface area of 1 loop = $2\pi db$

$$= 2\pi \times 0.711 \times 10^{-3} \times 0.019$$

$$= 8.5 \times 10^{-5}\ m^2$$

Number of turns per end of pipe, x:

$$\frac{x}{246} = \frac{214}{1000}$$

$$x = 52.6 \text{ turns / pipe end}$$

number of loops per pipe end = 90×53

$= 4770$

surface area per pipe end = $4770 \times 8.5 \times 10^{-5}$

$= 0.407 m^2$

Therefore the surface area of finning on 9 pipe ends, i.e. the evaporator or condenser is $9 \times 0.407 =$ **$3.7 m^2$**

Surface area of pipes without finning is:

No. of pipes $\times \pi Dl$ $= 9 \times \pi\ 0.022 \times 0.246$

$=$ **$0.153\ m^2$**

Therefore the finning greatly increases the total heat exchanger surface.

Calculation of RPM for steady heat transfer characteristics:

Work by Bontempts et al (1984) showed that for stable revolving heat pipe operation, the centrifugal force must be greater than the gravitational force acting on the working fluid (see Chapter 4.5.3), .

i.e. $\omega^2 (R+r) > g$

$\omega^2 (R+r) > 9.81$

Where: R = 100mm (0.1m)

r = 11mm (0.01m)

When the centrifugal force is equal to the gravitational force:

$$\omega^2 (R+r) = 9.81$$

$$\omega^2 (0.1+0.01) = 9.81$$

$$0.11\omega^2 = 9.81$$

$$\omega^2 = 89.18$$

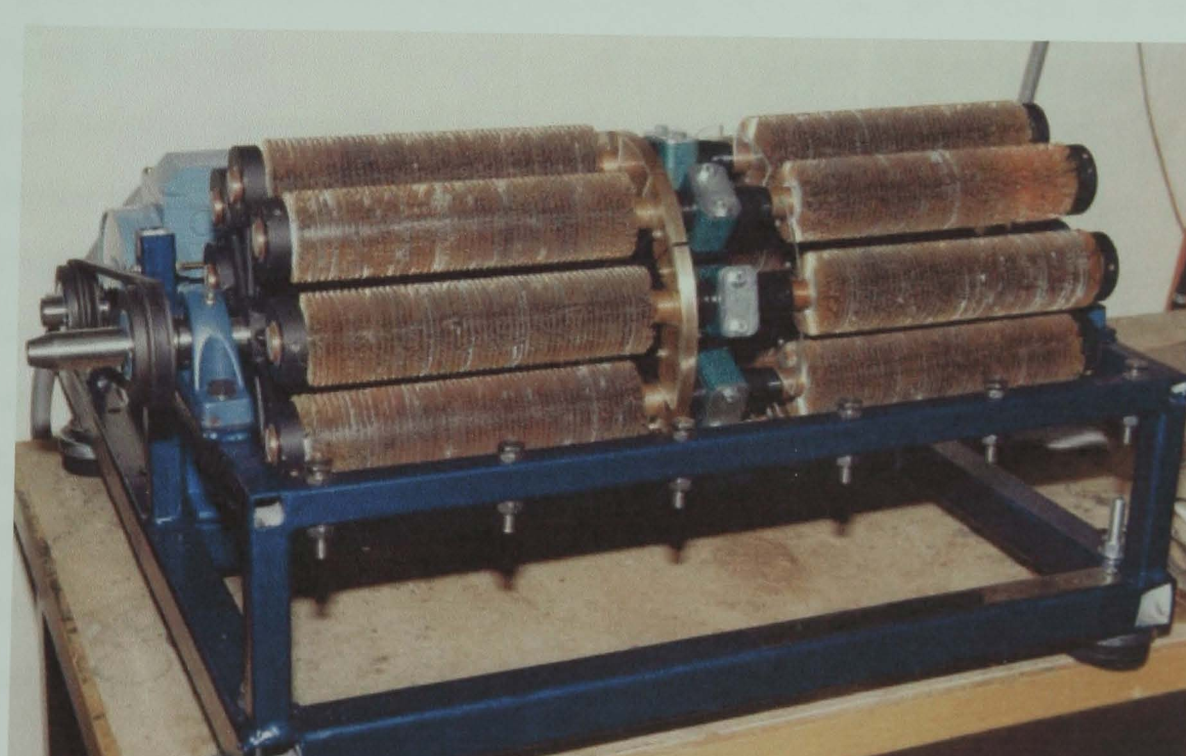
$$\omega = 9.44$$

$$RPM = \omega/2\pi \times 60$$

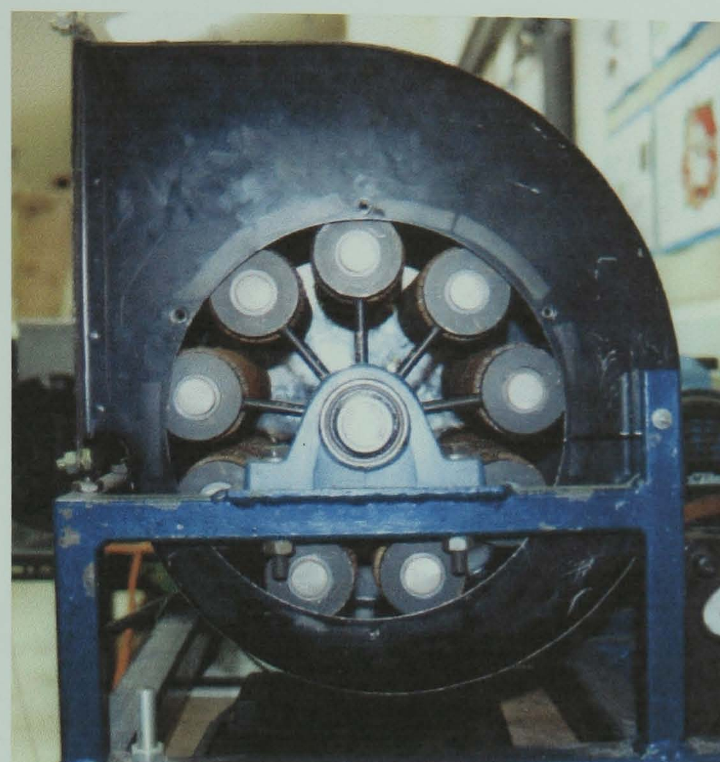
$$= \mathbf{90}$$

Therefore, for steady heat transfer characteristics, the heat pipes should revolve at a speed greater than 90 rpm.

Photographs 6.8 and 6.9 show the assembled 9-pipe wire fin rig with and without fan casings.



Photograph 6.8: 9-Pipe Wire Finned Rig



Photograph 6.9: 9-Pipe Wire Finned Rig (Side View)

Figure 6.9 illustrates the 9-pipe rig construction. The end clamps, shown in Figure 6.10, slide over the ends of the heat pipes and are held in place with grub screws. The middle clamps, shown in Figure 6.11, are moulded plastic clamps which split in half to enable assembly. The mild steel spokes are silver soldered onto the mid steel centre boss. The heat pipes are insulated between the fan casings with tubes of 1 inch thick flexible synthetic insulation. The fan casings are also covered with this insulation in sheet form. Figure 6.12 illustrates the revolving seals used to seal the wall of the fan casings through which the heat pipes pass. They consist of slotted perspex plates with three brass rims which are positioned in place after the pipes are secured. The brush seals run around the circumference of the brass rim to produce a seal.

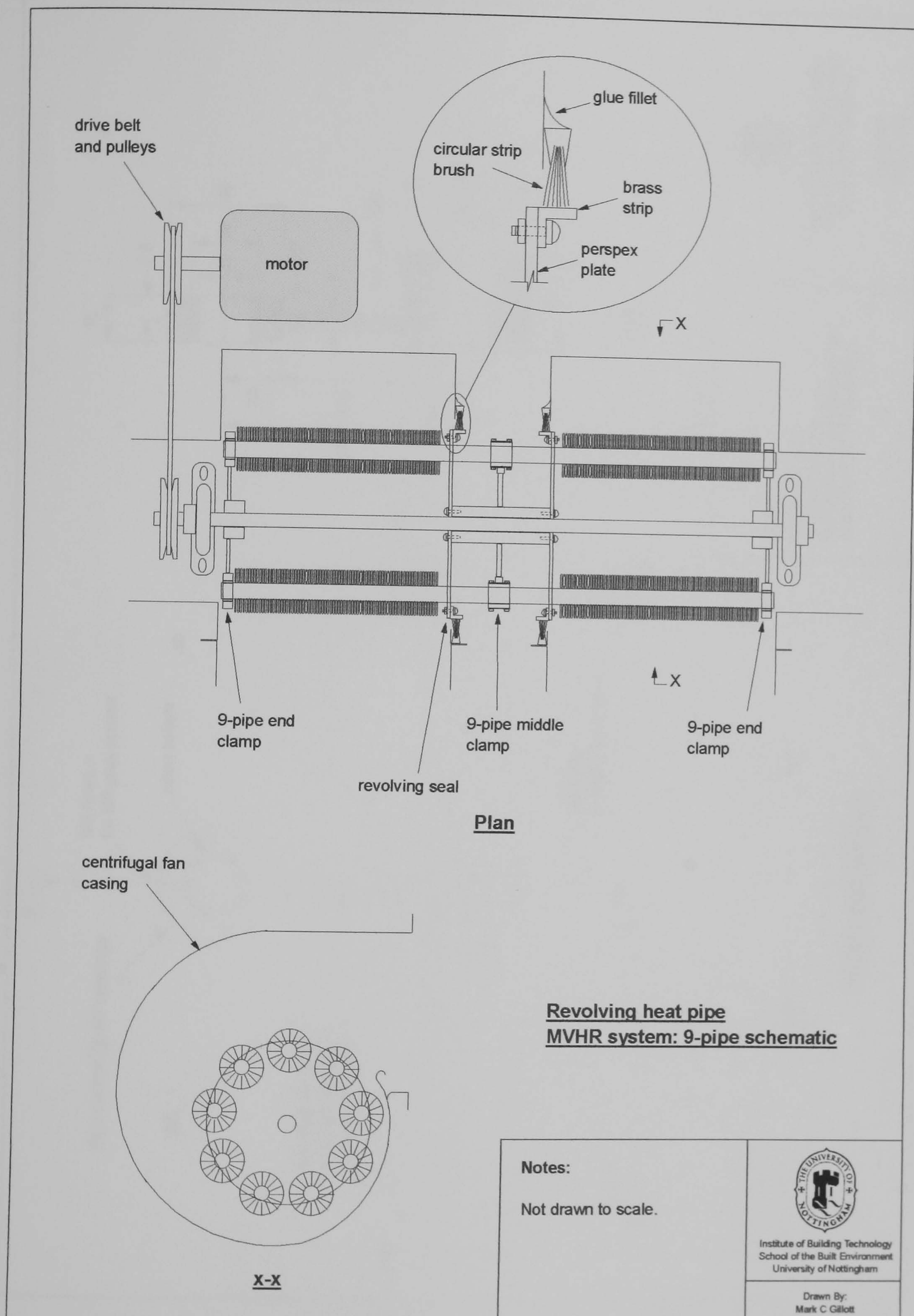


Figure 6.9: 9-Pipe Rig Construction

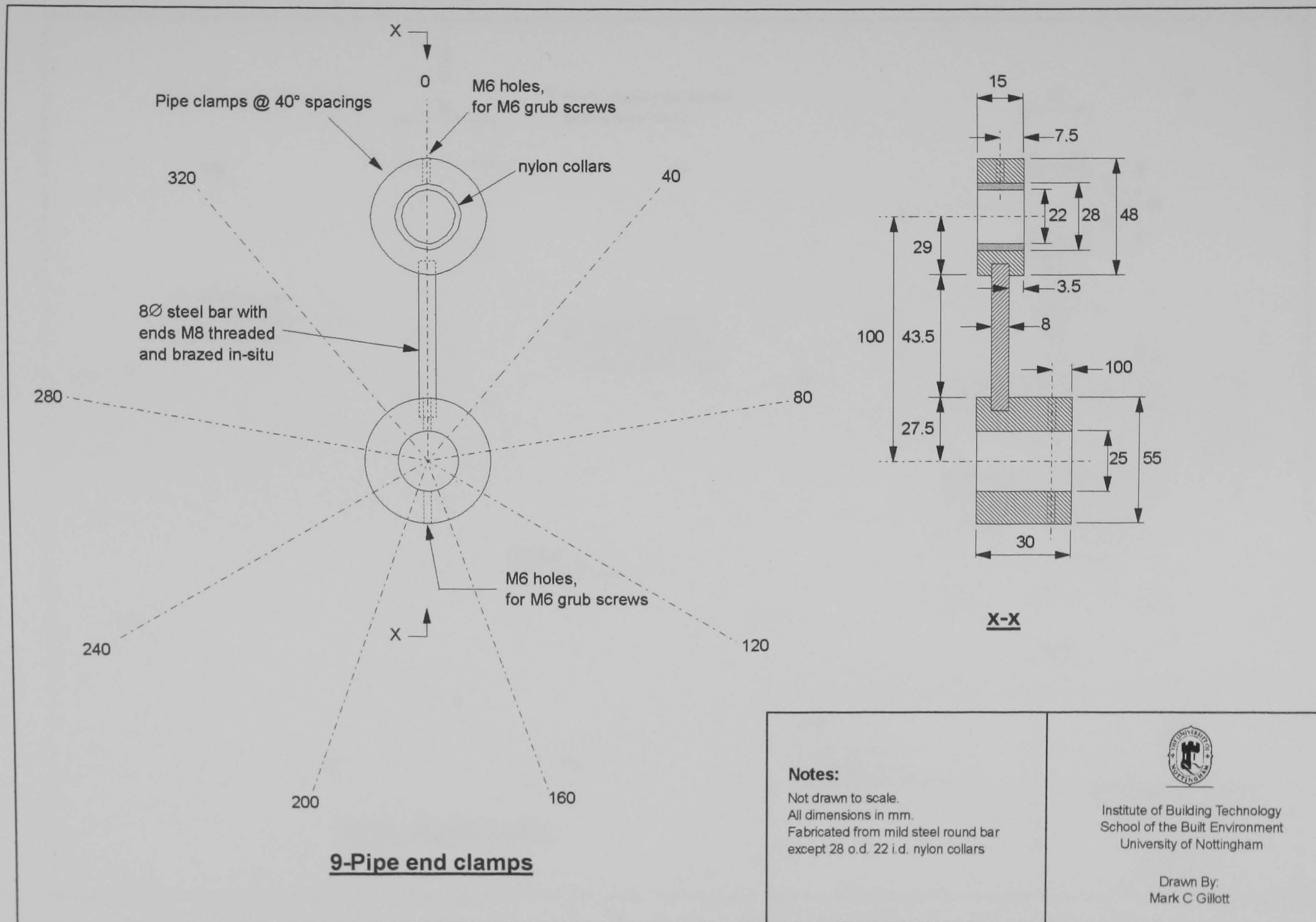


Figure 6.10: 9-Pipe End Clamps

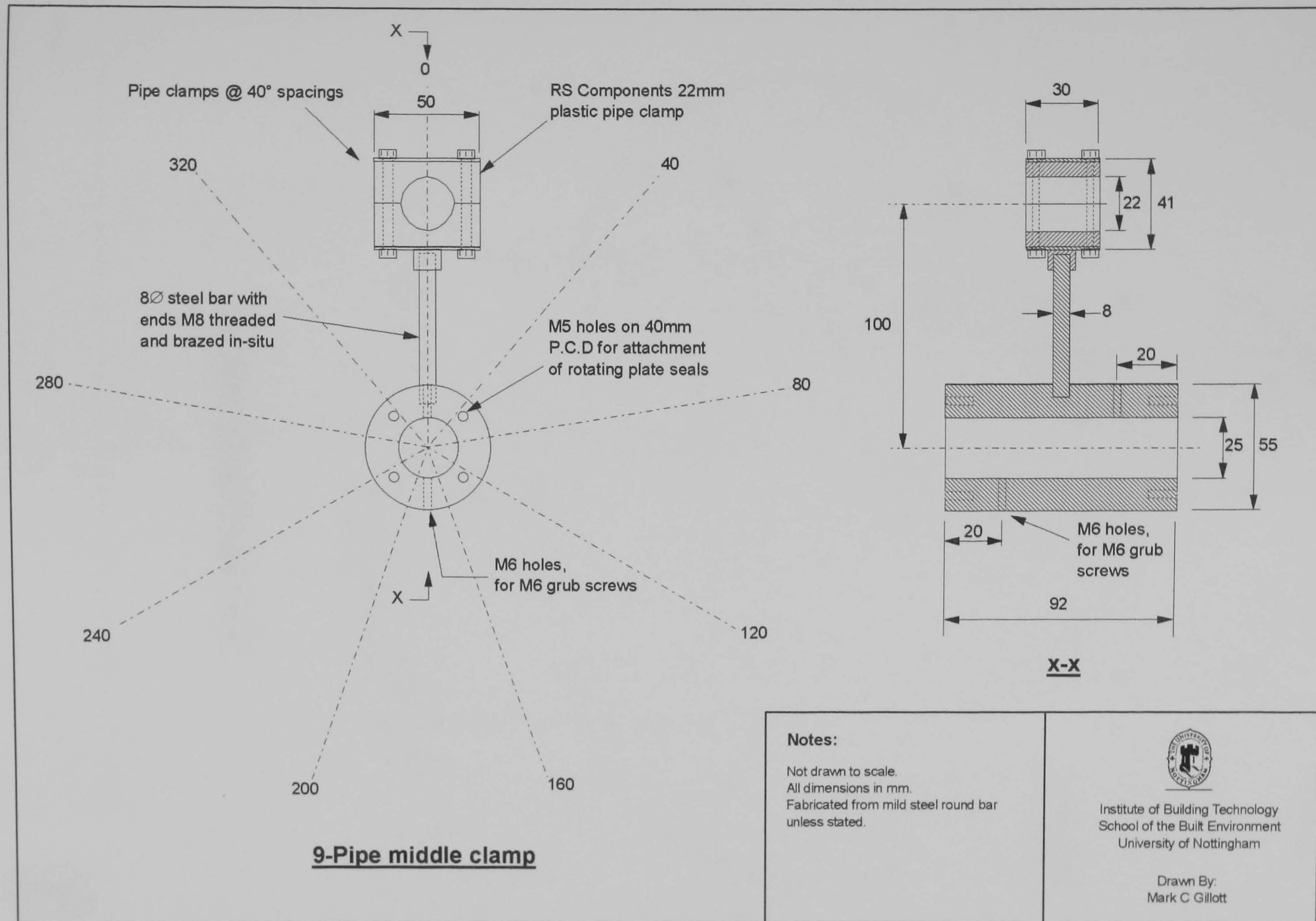
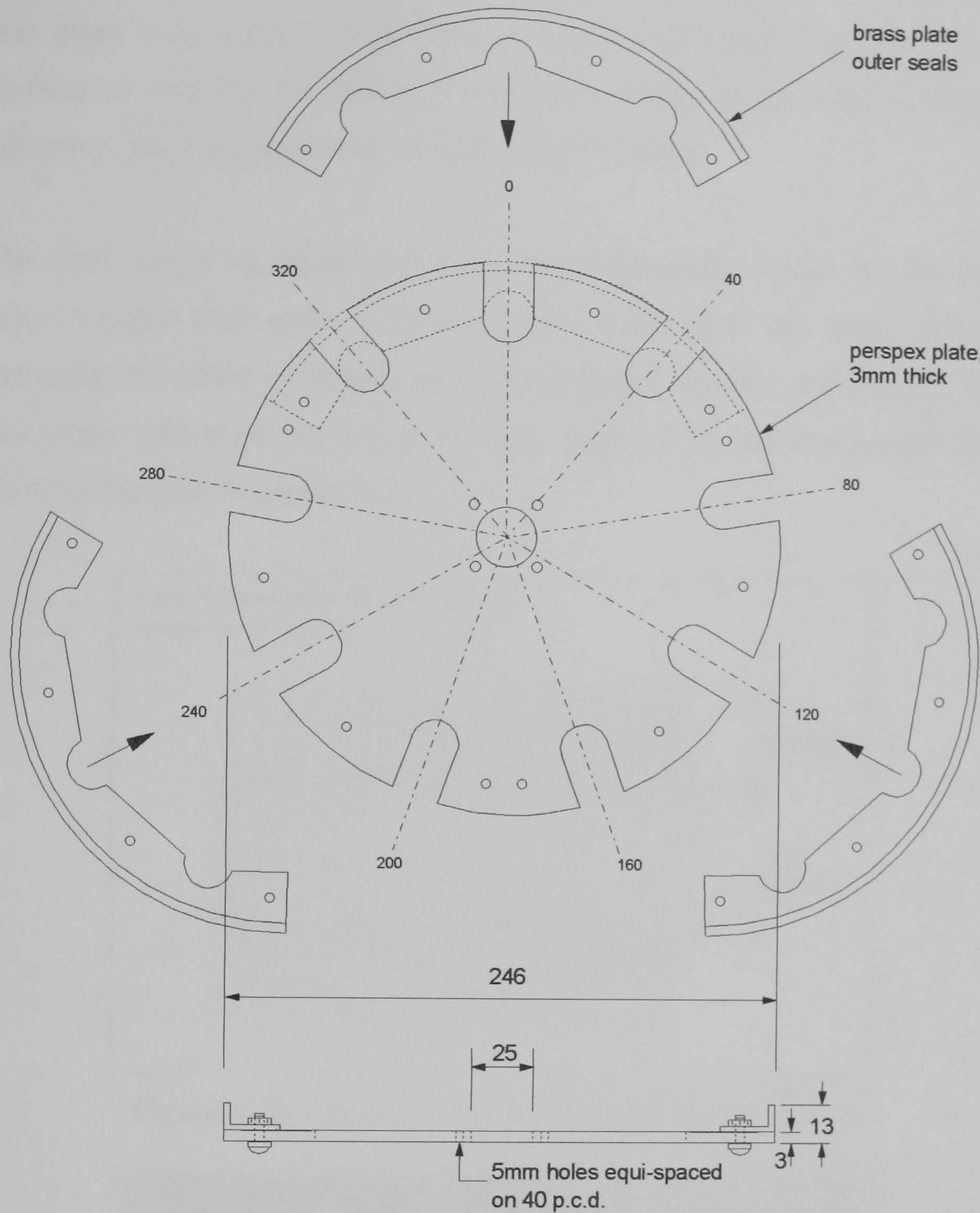


Figure 6.11: 9-Pipe Middle Clamp



9-Pipe Revolving Seal

Notes:

Not drawn to scale.
All dimensions in mm, unless stated.

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Figure 6.12: 9-Pipe Revolving Seal

The second 9-pipe rig used the same components except that the wire finned heat pipes were charged with 20% by volume of water. The charge was doubled to establish the effect of refrigerant charge on the heat exchanger efficiency. The nine pipes with 20% fill weigh 15.42kg.

The third 9-pipe rig again used the same components, except for the heat pipes. Copper heat pipes (∅22mm) with copper plate fins were used to determine the effect of finning on heat exchanger and fan performance. The nine pipes with plate fins weigh 13.75kg. Figure 6.13 and Photograph 6.10 illustrate the type of finning used.

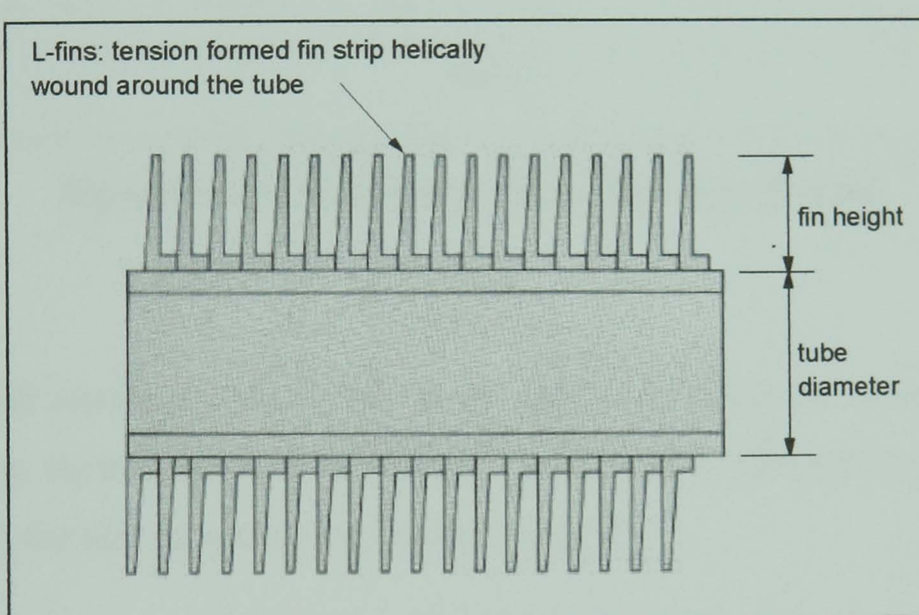


Figure 6.13: Circular Tension Formed Plate Finning

Photograph 6.10: Plate Finned Heat Pipes

The density of plate finning was as high as possible to maximise the surface area available for heat transfer. Figure 6.14 illustrates the finning specification for the heat pipe.

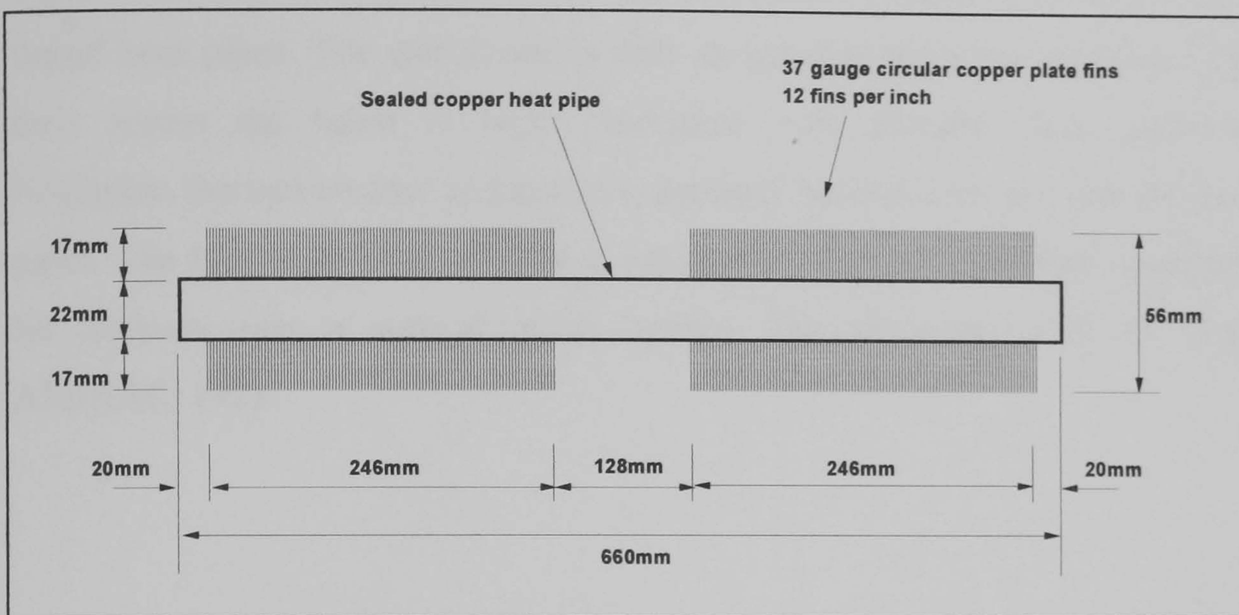


Figure 6.14: Plate Finned Heat Pipe Specification

The overall diameter of the plate finned pipes is less than the overall diameter of the wire finned pipes due to fabrication limitations. The surface area of the finning on the nine pipe ends in one fan casing is:

Length of finning $= 246/25.4$

$= 9.68$ inches

Number of fins at 12 fins/inch $= 9.68 \times 12$

$= 116$

Area for 1 fin (both sides) $= 2 \times (\pi(B/2)^2 - \pi(D/2)^2)$

(where: B = plate fin $\varnothing$ and D = pipe $\varnothing$)

$= 2(\pi(0.028)^2 - \pi(0.011)^2)$

$= 4.166 \times 10^{-3}\ m^2$

Area of finning on 1 pipe $= 116 \times 4.166 \times 10^{-3}\ m^2$

Area of finning on 9 pipes $= \mathbf{4.35\ m^2}$

Calculation of theoretical heat transfer rate, Q:

The following section calculates the theoretical heat transfer rate for the plate finned heat pipes. The calculation is only an approximation as in practice the flow across the tubes is highly turbulent with complex flow patterns. Negligible thermal contact resistance is assumed between the fins and the heat pipes. The heat transfer coefficient is calculated using the simplified equations for airflows over a vertical plane surface. The equations used are from ASHRAE, 1981.

$$Q = h \ \Delta t \ (A_b + \eta_f A_f) \tag{6.2}$$

Where:
Q = heat transfer rate (W)
h = heat transfer coefficient (W/m^2K)
Δt = temperature difference between the surface and bulk of the fluid (°C)
A_b = surface area of pipe without fins (m^2)
A_f = surface area of fins (m^2)
η_f = fin efficiency

A_b is very small relative to the fin area because when the finning is attached to the pipes the area of pipe surface without finning is very small due to the high fin density. For the purpose of this calculation, A_b is assumed to be zero.

ΔT is taken to be 10°C. This is assumed from the following where:

temperature of air onto the evaporators = 20°C = T_1,

temperature of the evaporators = T_2,

temperature of air onto the condensers = 0°C = T_3,

temperature of the condensers = T_4,

area of condensers and evaporators are A_c and A_e respectively,

heat transfer coefficients of condensers and evaporators are h_c and h_e respectively.

$Q_{out} = A_c\, h_c(T_1 - T_2)$

$Q_{in} = A_e\, h_e(T_4 - T_3)$

$Q_{in} = Q_{out}$

If $A_c = A_e$ and $h_c = h_e$

$T_1 - T_2 = T_4 - T_3$

$T_1 + T_3 = T_2 + T_4$

$20 + 0 = T_2 + T_4$

Assuming the heat pipe is highly efficient $T_2 \approx T_4$

$T_2 = T_4 = 10°C$

Therefore ΔT = 10°C.

Calculation of heat transfer coefficient:

For $v < 16$fps (4.9m/s) $\qquad h = 0.99 + 0.21v$ **(6.3)**

For v of 16 to 100fps (4.9 to 30.5m/s) $\qquad h = 0.5\,(v)^{0.8}$ **(6.4)**

N.B. The ASHRAE equations use imperial units, where h is Btu/hrft^{2}F and h is fps.

Calculation of velocity, v:

The velocity of air over the fins varies with the distance to the axis of rotation. Therefore, an average value for velocity is assumed where the radius is taken to be the distance from the centre of the heat pipes to the axis of rotation. The velocity at a radius of 100mm and at a typical speed of 800 rpm is:

$$\begin{aligned} v &= 2\pi R\,(800/60) \\ &= 8.4\text{m/s} \\ &= 27.56\text{fps} \end{aligned}$$

The velocity v, is the velocity of air that is slipping over the pipes. This value of velocity is high as the revolving heat pipes are acting as a fan and therefore some of the air is being moved by the pipes. The value will be lower than that calculated and its magnitude depends on how efficient the pipes are at moving the air.

Substituting v into equation 6.4:

$$\begin{aligned} h &= 0.5\,(27.56)^{0.8} \\ &= 7\ \text{Btu/hr.ft}^2.\text{F} \\ &= 39.7\ \text{W/m}^2\text{K} \end{aligned}$$

Calculation of fin efficiency, η_f:

For a cylindrical plate fin,

$$\eta_f = \frac{\tanh(mr_i\phi)}{mr_i\phi} \tag{6.5}$$

$$m = \sqrt{\frac{2h}{tk}} \tag{6.6}$$

$$\phi = [(r_o/r_i) - 1][1+0.35\ln(r_o/r_i)] \tag{6.7}$$

Plate thickness, $t = 0.24 \times 10^{-3}$m

The thermal conductivity of copper, k = 394 W/mK

Fin radius, r_o = 56/2

= 28mm

Tube radius, r_i = 22/2

= 11mm

Substituting values into equation 6.6:

$$m = \sqrt{\frac{2 \times 39.7}{0.24 \times 10^{-3} \times 394}}$$

$$m = 28.98$$

Substituting values into equation 6.7:

$$\phi = [(28/11) - 1][1+0.35\ln(28/11)]$$
$$\phi = 2.05$$

Substituting values into equation 6.5:

$$\eta_f = \frac{\tanh(28.98 \times 0.011 \times 2.05)}{28.98 \times .011 \times 2.05}$$

$$\eta_f = 88\%$$

Area of finning = $4.35m^2$

Substituting values into equation 6.2:

$$Q = h\ \Delta t\ (A_b + \eta_f A_f)$$
$$= 39.7 \times 10\ (0 + 0.88 \times 4.35)$$
$$= \mathbf{1520W}$$

The calculated theoretical heat transfer rate meets the system requirement of 1.5kW. However, the actual heat transfer, in practice, will be less due to limiting factors such as heat pipe performance. The interaction of the tubes with one another will also effect the heat transfer. The temperature difference, Δt, will not be as high as 10°C as the majority of fins will transfer heat to air that has already been heated by the other fins. As already explained the value of the velocity will be much lower. A lower air velocity will reduce the heat transfer coefficient which in turn will reduce the heat transfer.

If there is no finning on the pipes the theoretical heat transfer rate is:

Finding the Reynolds number:

$$Re = \frac{Dv}{\nu} \tag{6.8}$$

$$= (0.022 \times 8.4) / 1.3 \times 10^{-5}$$

$$= 14215$$

Finding the heat transfer coefficient:

$$\frac{hD}{k_f} = 0.24(Re)^{0.6} \tag{6.9}$$

$$= 0.024 \times 0.24\ (14215)^{0.6} / 0.022$$

$$= 81.21\ W/m^2{}^\circ C$$

Finding the rate of heat transfer:

$$Q = h\ \Delta t\ A_b$$

$$= 81.21 \times 20 \times 0.153$$

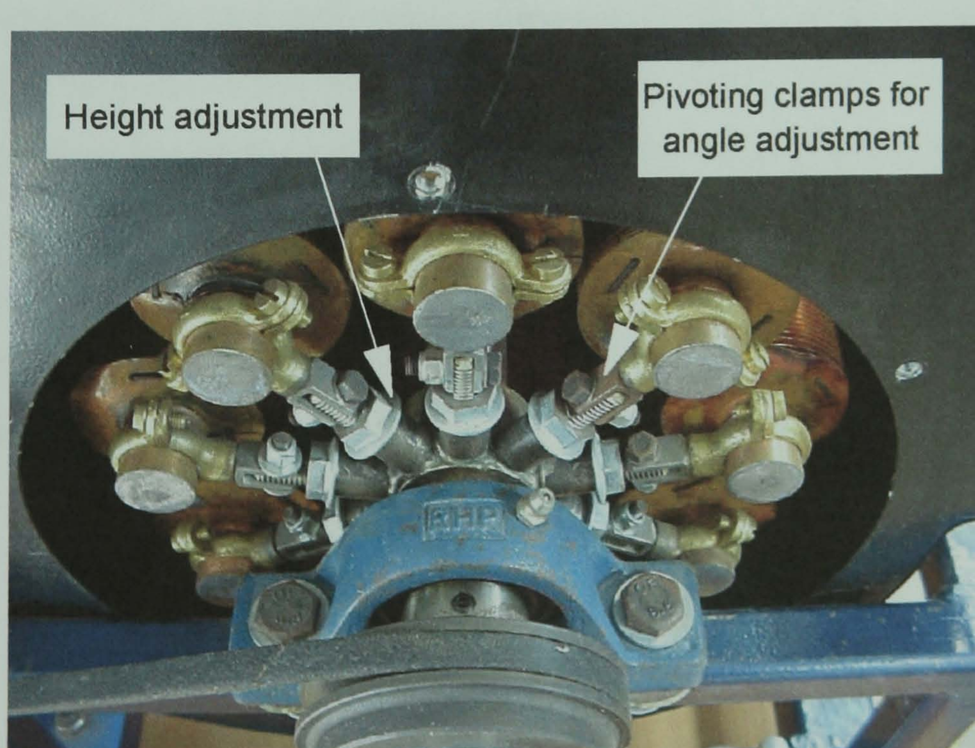
$$= \mathbf{249\ W}$$

From the results it is seen that the estimated heat transfer of the heat pipes with finning is far greater than if no finning was used. The large area of finning with its high theoretical heat transfer rate ensures, in practice, good heat transfer whilst fulfilling the requirement of impelling the air.

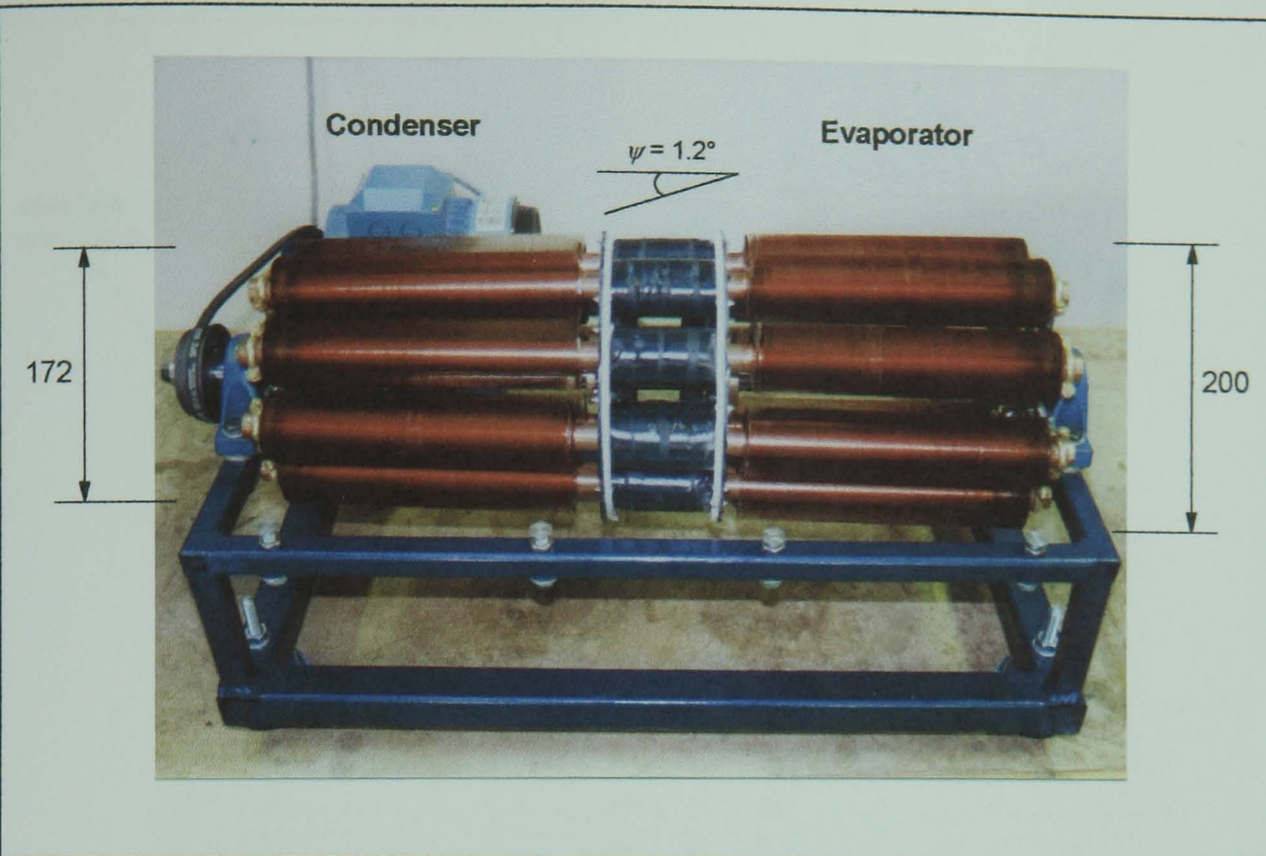
6.2.3 Nine pipe tapered rig

The forth prototype system uses the 9 copper plate finned heat pipes (10% fill) with different clamps that allow the heat pipe bundle to be tapered. The radius of revolution increases along the length of the pipes from the condenser ends to the evaporator ends. By tapering the heat pipe bundle, a component of force is generated along the wall of the pipes from centrifugal acceleration. This force will work in returning the condensate to the evaporator ends. Testing will determine whether this arrangement has efficiency benefits that merit the more complex construction.

The heat pipe clamps pivot to enable adjustment of the angle of the heat pipes' axes relative to the axis of rotation. The length of each spoke on the clamps can also be adjusted to alter the distance of the heat pipes from the axis of rotation. The correct adjustment of the clamp angles and height at the three clamping positions along the lengths of the heat pipes, enable the heat pipe bundle to be tapered. Photograph 6.11 shows the adjustable clamps and Photograph 6.12 shows the assembled tapered heat pipe bundle.



Photograph 6.11: Adjustable Clamps For Tapered Rig



Photograph 6.12: Tapered Heat Pipe Bundle

The tapered bundle shown is at the maximum achievable taper angle. The maximum taper angle is restricted by the maximum and minimum heat pipe bundle diameters at the evaporator and condenser ends respectively. The diameter of the heat pipe bundle at the evaporator end is limited by the size of the fan casing. The heat pipe bundle diameter at the condenser end, can only be reduced to a size where the finning of adjacent heat pipes are touching. The 9 copper plate finned heat pipes were used rather than the wire finned pipes because they have smaller fin heights. This allows close spacing of the pipes at the condenser end, which gives a large taper angle. Figure 6.15 illustrates a schematic of the 9-pipe tapered bundle rig and Figure 6.16 illustrates the adjustable clamps used to hold the heat pipes in a tapered bundle.

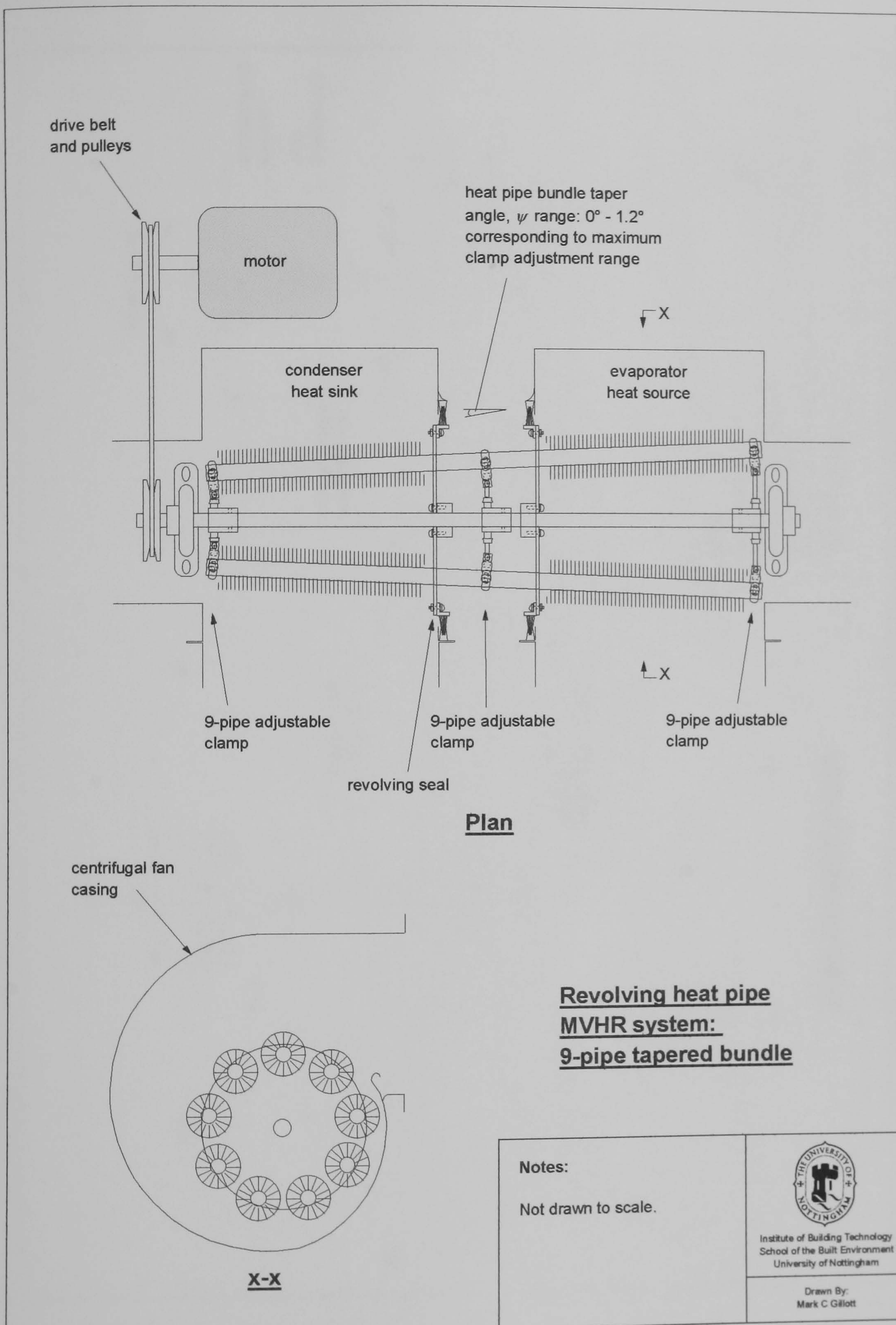


Figure 6.15: 9-Pipe Tapered Heat Pipe Bundle Rig

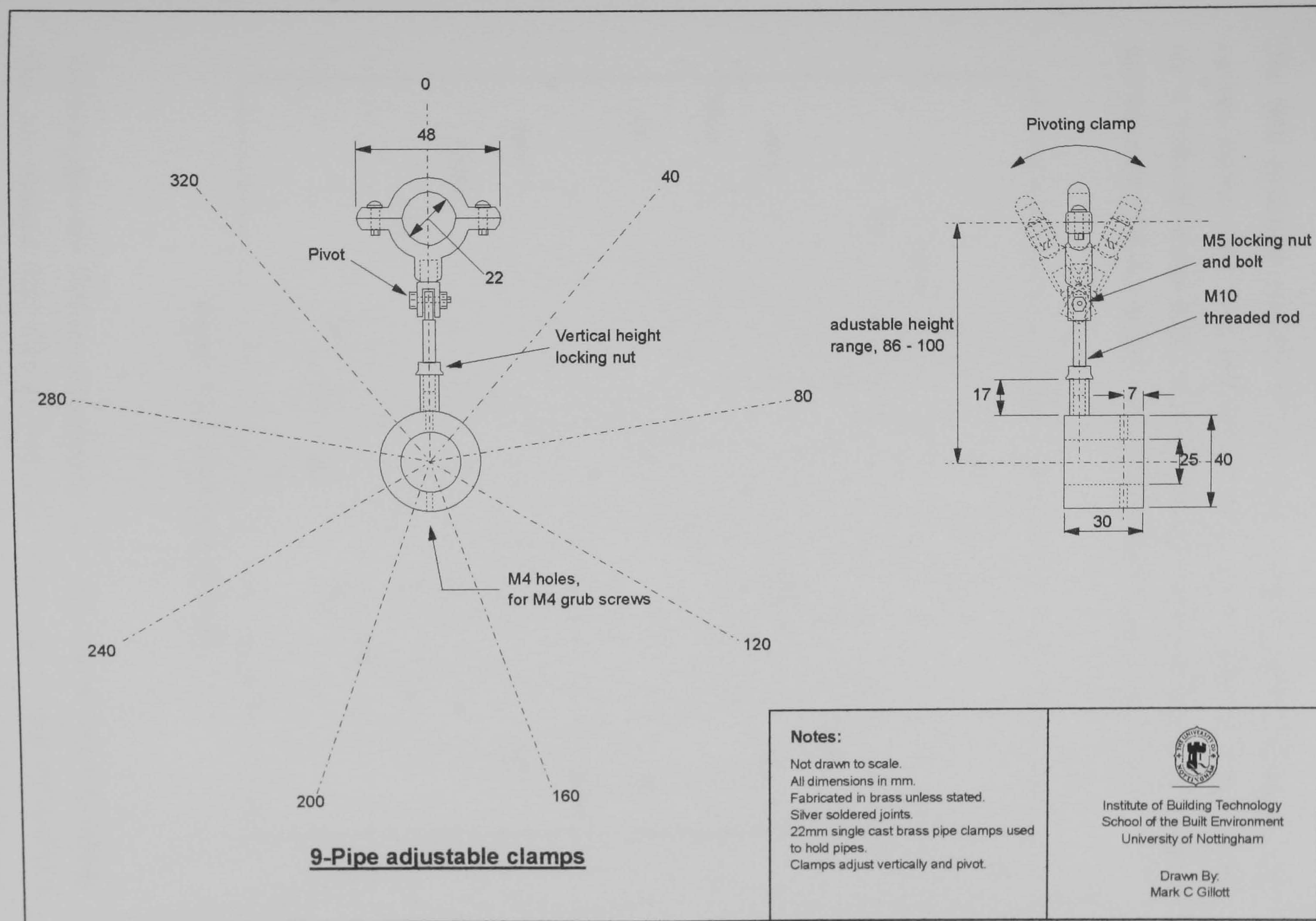


Figure 6.16: 9-Pipe Adjustable Clamps

6.2.4 Double row 24 pipe rig

The fifth prototype consists of the same specification of framework, fan casings, motor, drive belts and pulleys. However, the heat pipes and clamps are a different design with an alternative geometric configuration. Twenty four pipes are used in a double row configuration shown in Figure 6.17.

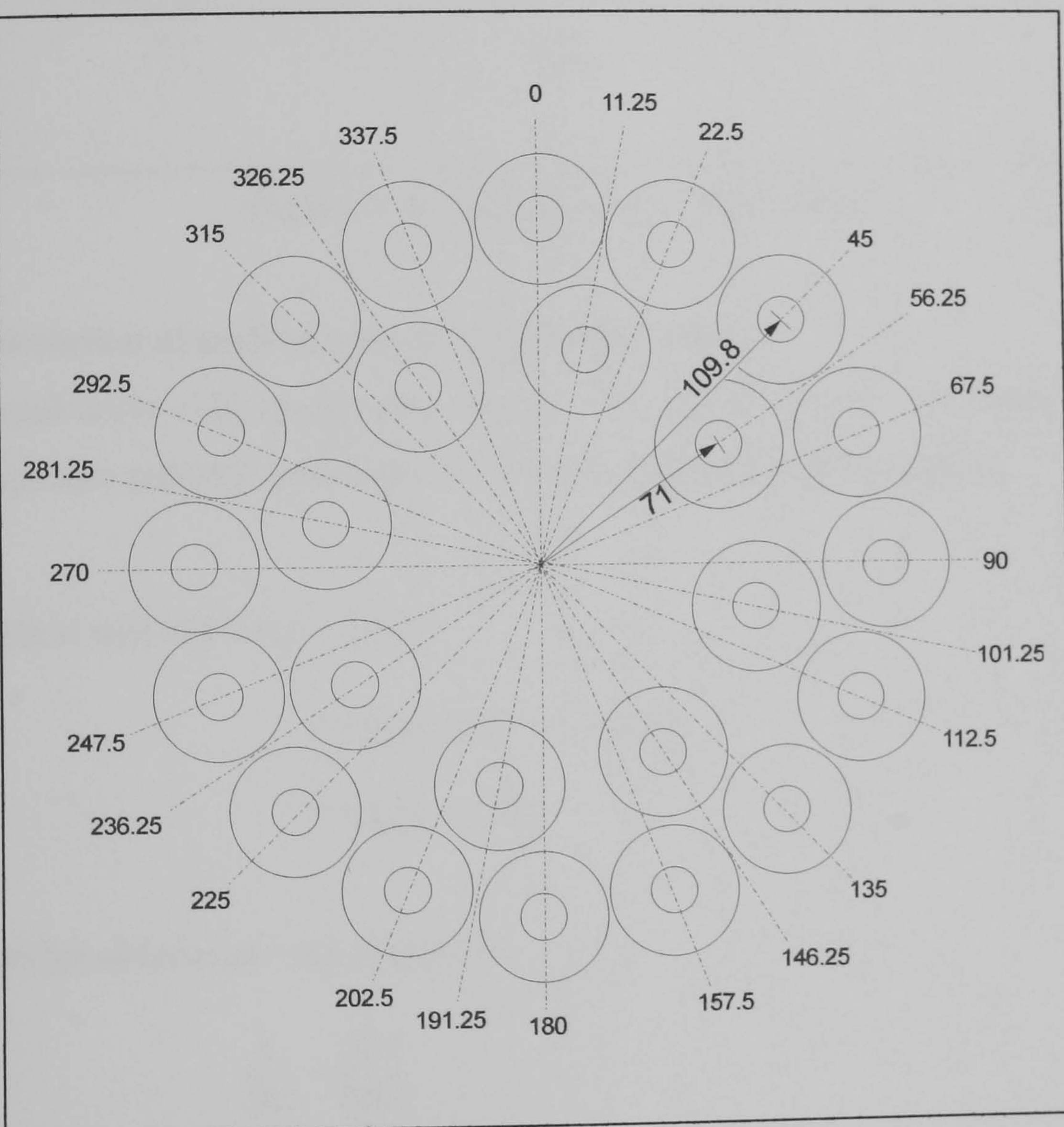


Figure 6.17: 24-Pipe Geometry

The heat pipes are 15mm copper tubes with ½" high copper wire loop finning. They are charged with 10% by volume of water and, as with the previous pipes, they have a double layer of 150μm phosphor bronze screen mesh wick. Figure 6.18 illustrates the heat pipe specification.

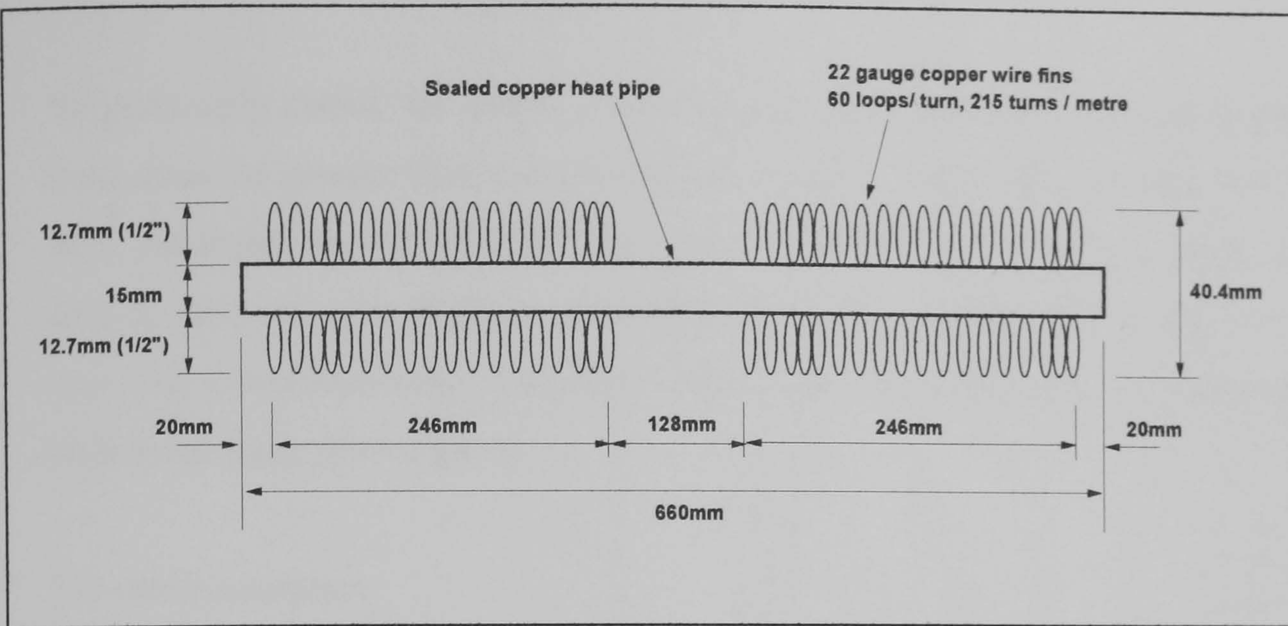


Figure 6.18: 24-Pipe Heat Pipe Design

Calculation of surface area of wire finning used:

Specification - 22 gauge wire fins ($d = 0.711\text{mm}\ \varnothing$), $b = 12.7\text{mm}$, at 60 loops/turn and 215 turns/metre on 24 pipes (246mm finned length, l).

Surface area of 1 loop $= 2\pi db$

$$= 2\pi \times 0.711 \times 10^{-3} \times 0.0127$$

$$= 5.675 \times 10^{-5}\ \text{m}^2$$

Number of turns per end of pipe, x:

$$\frac{x}{246} = \frac{215}{1000}$$

$$x = 53 \text{ turns / pipe end}$$

number of loops per pipe end $= 60 \times 53$

$$= 3180$$

surface area per pipe end $= 3180 \times 5.675 \times 10^{-5}$

$$= 0.1805\text{m}^2$$

Surface area for 24 pipes = **4.33 m^2**

Calculation of RPM for steady heat transfer characteristics:

As previously stated, for stable revolving heat pipe operation the centrifugal force must be greater than the gravitational force acting on the working fluid. At a particular angular velocity, the centrifugal force acting on the working fluid in the inner row of pipes is less than that acting on the fluid in the outer row. The calculation must, therefore, look at the forces acting on the working fluid in the inner row of pipes.

For stable operation:

$$\omega^2 (R+r) > g$$

$$\omega^2 (R+r) > 9.81$$

Where: $R = 71\text{mm}\ (0.071\text{m})$

$r = 7.5\text{mm}\ (0.0075\text{m})$

When the centrifugal force is equal to the gravitational force:

$$\omega^2 (R+r) = 9.81$$

$$\omega^2 (0.071+0.0075) = 9.81$$

$$0.0785\omega^2 = 9.81$$

$$\omega^2 = 124.97$$

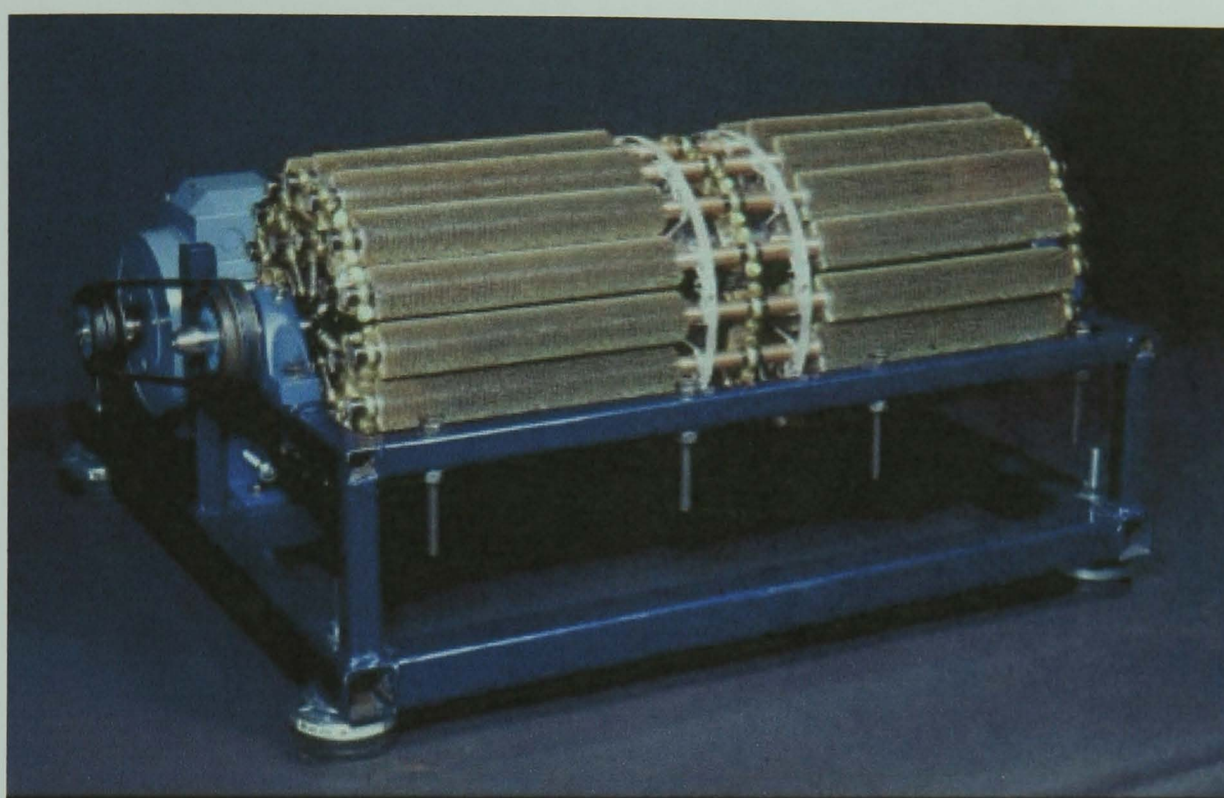
$$\omega = 11.17$$

$$\text{RPM} = \omega/2\pi \times 60$$

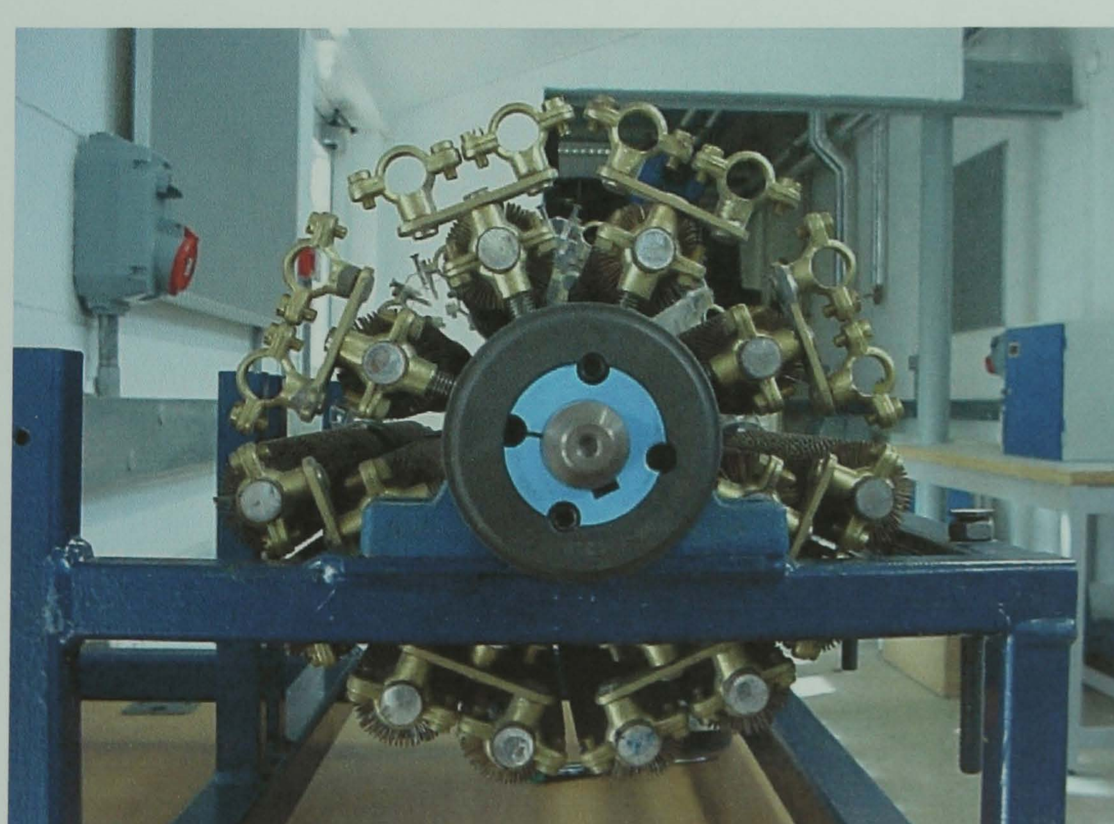
$$= \mathbf{107}$$

Therefore for steady heat transfer in all 24 heat pipes, they should revolve at a speed greater than 107 rpm.

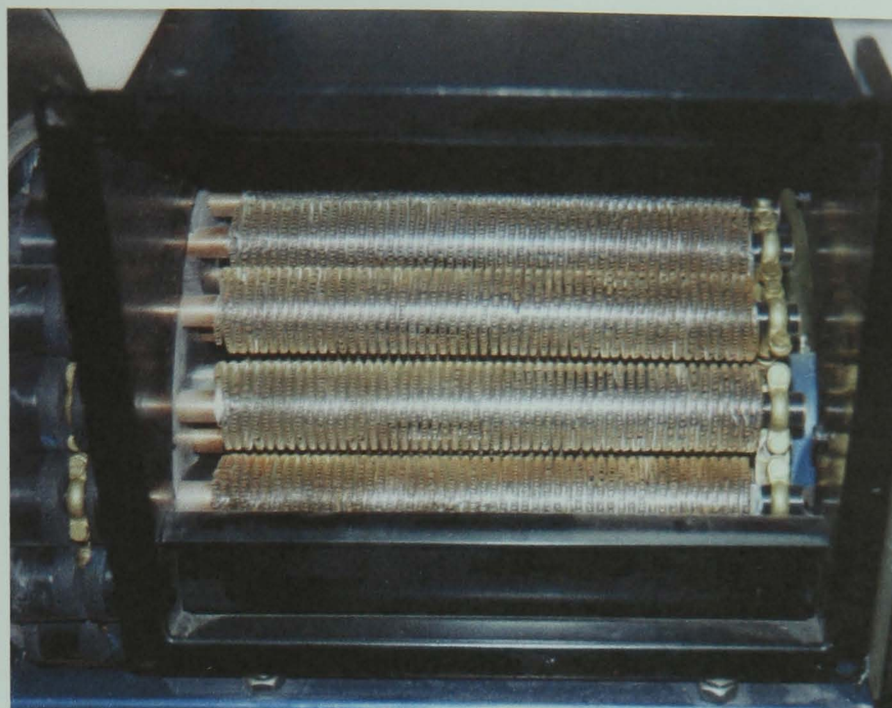
The heat pipes are clamped at their ends and middle by 15mm brass pipe clamps brazed together to hold the pipes in the configuration shown in Figure 6.17. The 24 heat pipes weigh 19.2kg. Photographs 6.13, 6.14 and 6.15 show the 24 pipe rig arrangement.



Photograph 6.13: 24-Pipe Wire Finned Rig

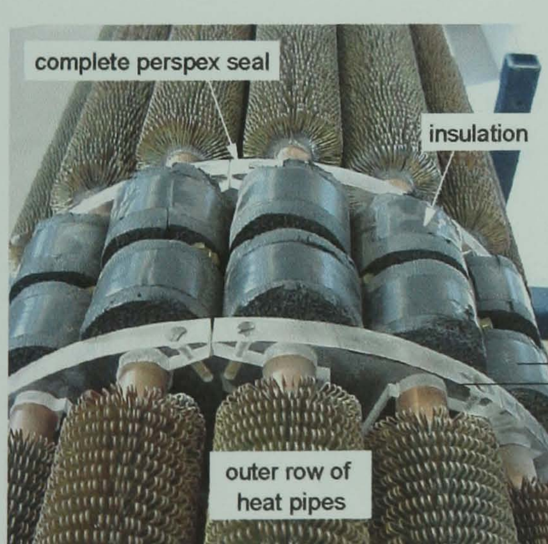


Photograph 6.14: 24-Pipe Rig: Pipe Clamps

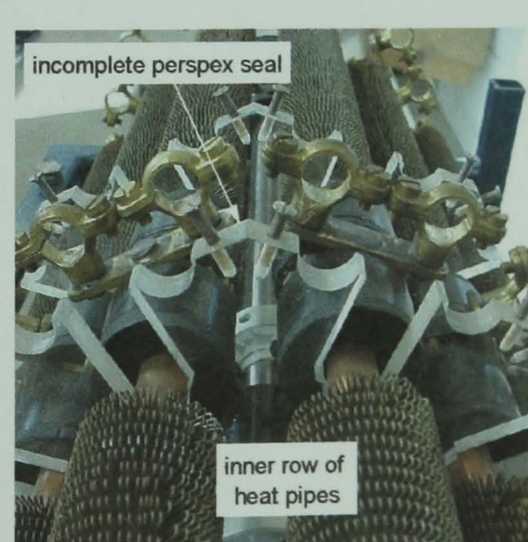


Photograph 6.15: 24-Pipe Rig: Fan Outlet

The rotating fan case seals are 10mm thick circular perspex plates, with eight detachable sections which are screwed in place once the pipes are secured in position. Photographs 6.16 and 6.17 show the complete and incomplete seals. Figure 6.19 illustrates the 24-pipe revolving seal pattern.



Photograph 6.16: 24-Pipe Rig: Both rows with complete seal



Photograph 6.17: 24-Pipe Rig: Inner row only with incomplete seal

Figure 6.20 illustrates the 24-pipe rig schematic and Figure 6.21 illustrates the 24-pipe clamp detail.

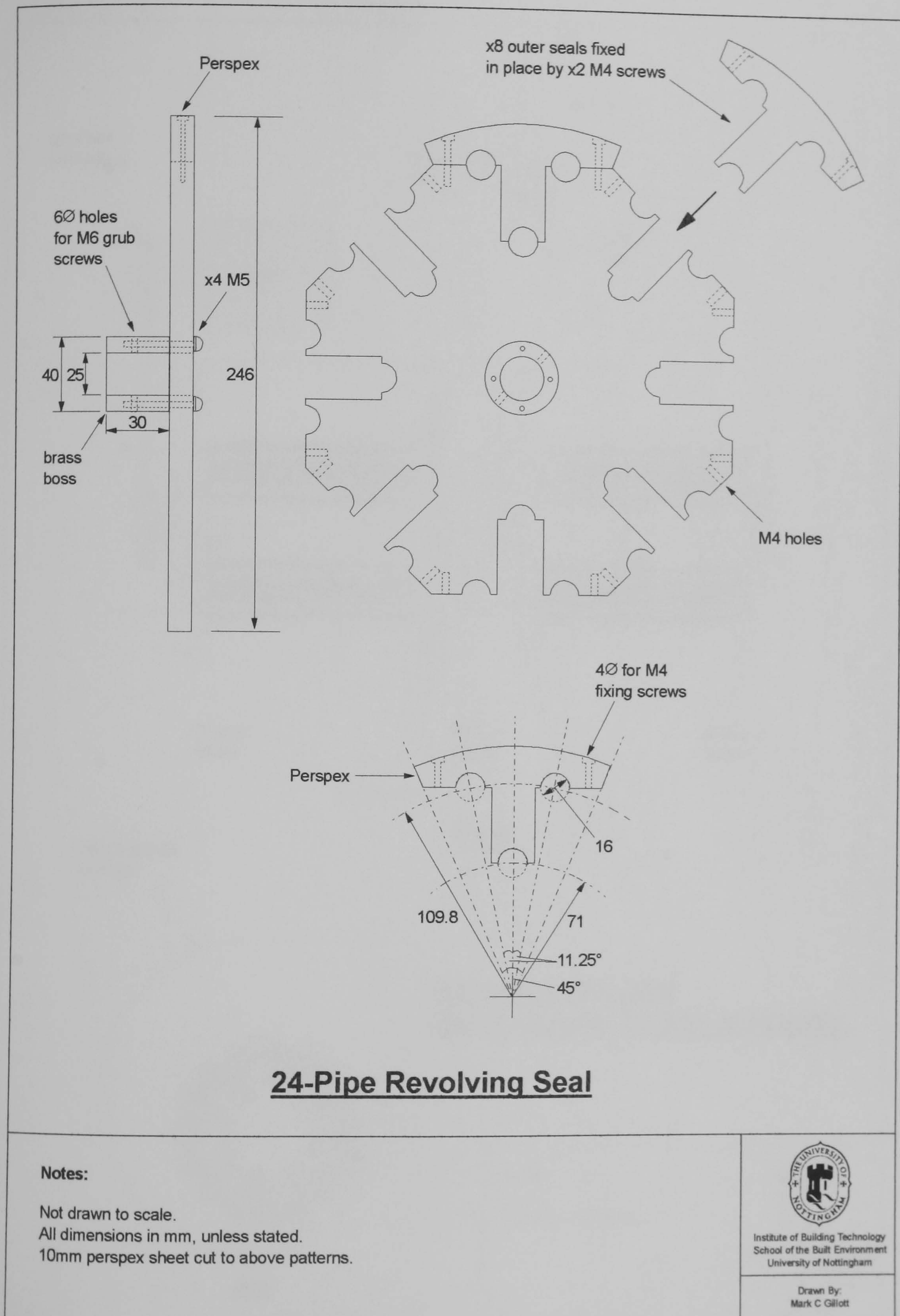


Figure 6.19: 24-Pipe Revolving Seal

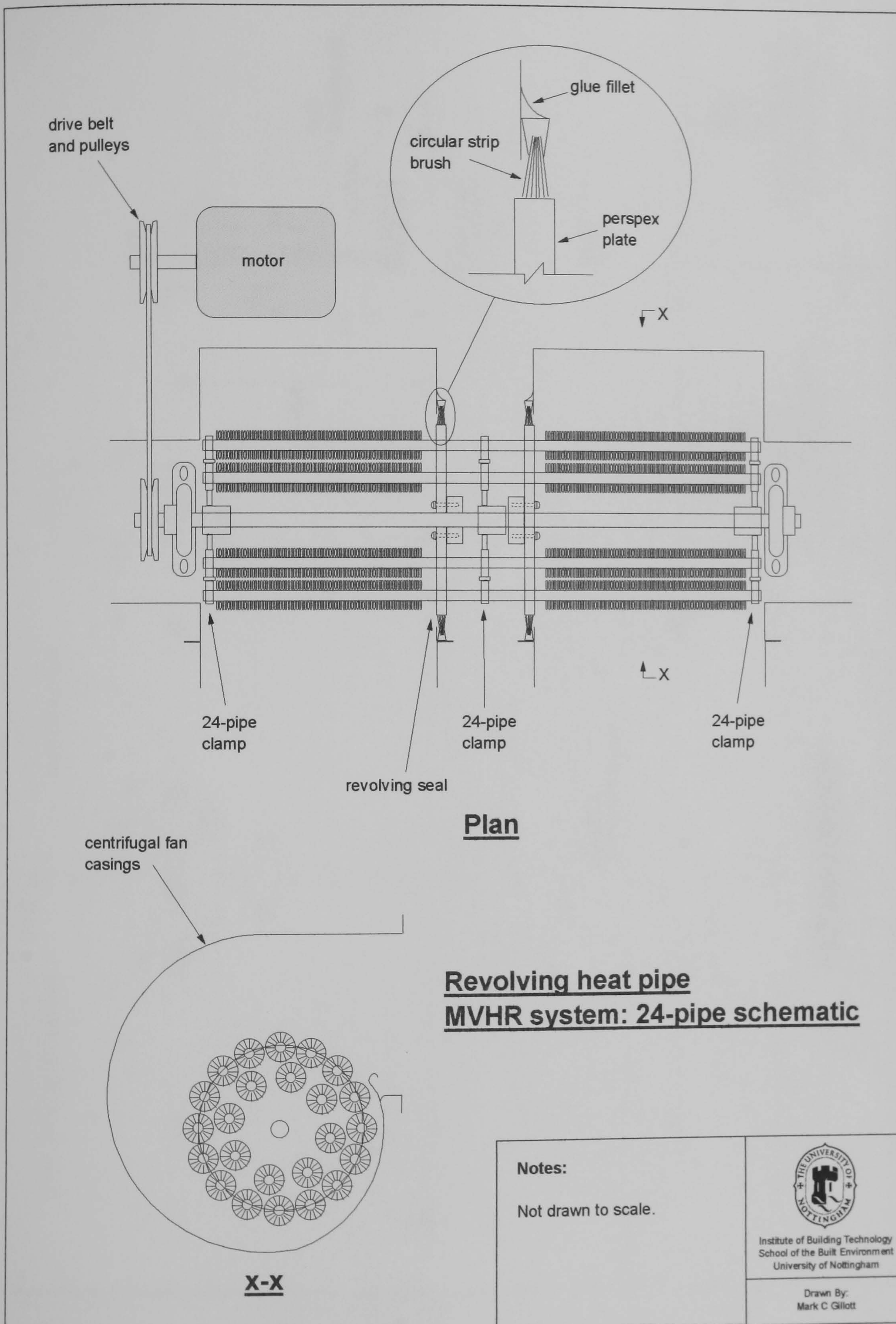


Figure 6.20: 24-Pipe Rig Schematic

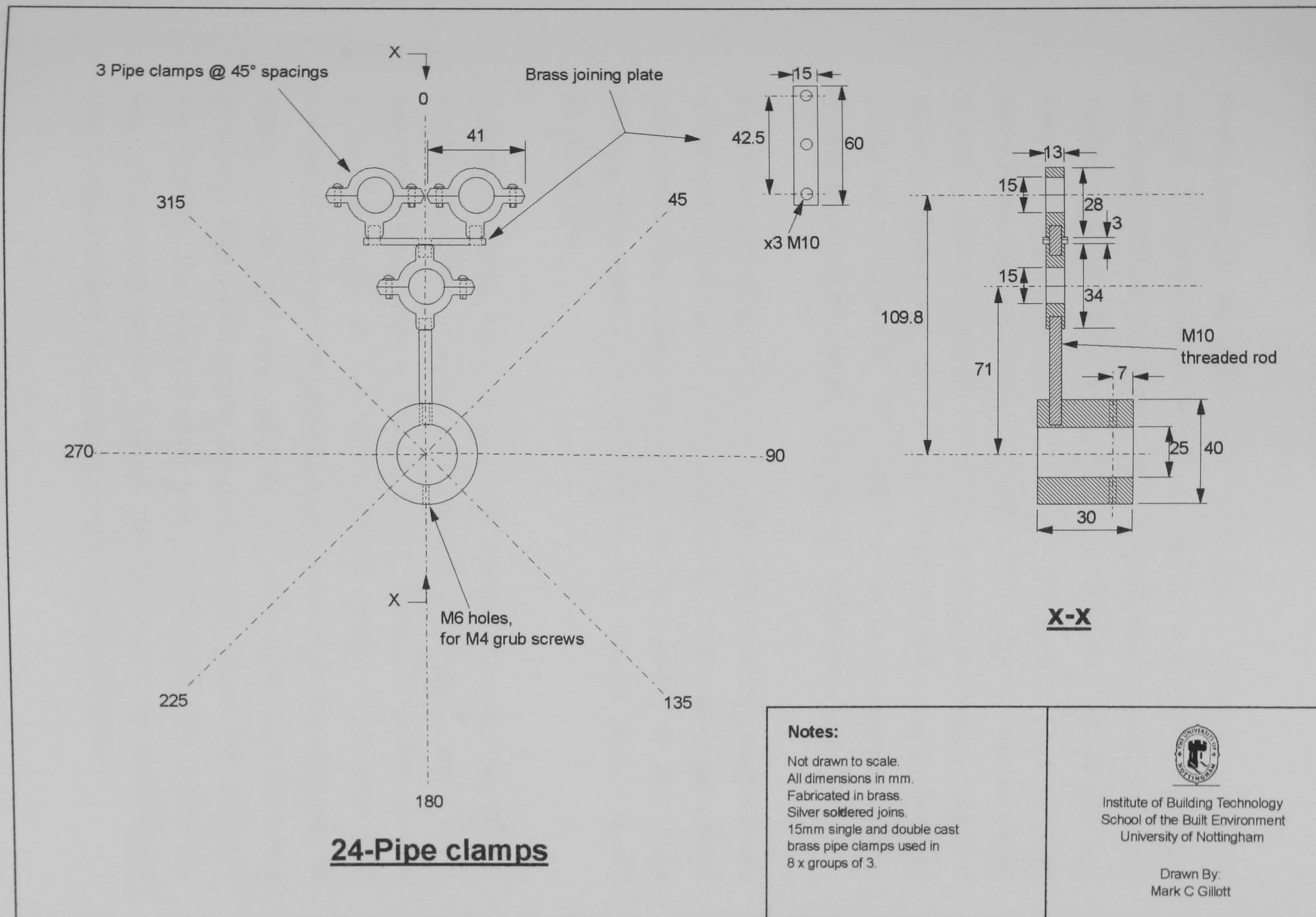


Figure 6.21: 24-Pipe Rig Clamps

6.3 Conclusions

Several prototype rig designs have been constructed for use as MVHR systems. The prototypes incorporate revolving heat pipes which work as both heat exchangers and air impellers. To comply with legislation, the systems are required to provide $200m^3/hr$ of fresh air continuously when installed in a typical four bed detached house. At this flow rate, and for typical operating temperatures, there will be approximately 1.5kW of thermal energy available to recover. This is a theoretical maximum for a heat exchanger efficiency of 100%. In practice, this value will not be reached and, therefore, less than 1.5kW will be transferred. The 9-pipe wire fin heat exchanger surface is theoretically more than capable of transferring this quantity of heat.

The surface area of the finning on the heat pipes in one fan casing is $3.7m^2$, $4.33m^2$ and $4.35m^2$ for the 9 wire finned pipes, 24 wire finned pipes and 9 plate finned pipes respectively. For steady heat transfer, the rpm of the revolving pipes must be greater than 90 for the 9 pipes and greater than 107 for the 24 pipes.

Appendix C shows details of the latest prototype revolving heat MVHR system. The system is smaller than the initial prototypes. It uses a 200W single phase 6-pole motor to rotate the pipes at 800rpm. The motor is mounted in the cold supply air duct and is directly coupled to the shaft. This arrangement means the heat generated in the motor can be used to pre-heat the supply air. The heat pipes used are very much lighter than those used previously. Thin walled copper tube (0.5mm wall) is used with aluminium wire loop fins. The weight of all the pipes is 8.4kg. So that the pipes do not flex or bend during rotation, 'Table Z' type (hard drawn) thin walled copper tube is used. Testing of this prototype is currently being undertaken.

7. TESTING OF MVHR SYSTEMS

This chapter covers the testing and performance evaluation of the prototype MVHR systems. The systems were tested in terms of their heat exchanger efficiency and fan performance. A commercial MVHR system was also tested so that the performance of the prototypes could be compared to the performance of an existing system.

7.1 Ventilation heat recovery tests

Figure 7.1 and Photograph 7.1 show the experimental set-up of the test rig. To avoid confusion, air in the warm air side is referred to as exhaust air, and air in the cool air side is referred to as supply air, i.e. representing the supply and exhaust air flows to and from a dwelling. In order to simulate the working conditions of a ventilation heat recovery system, the temperatures of both inlets to the prototype systems were carefully controlled.

Ductwork insulated with 1 inch thick flexible synthetic insulation was connected to the prototypes. The inlet ducts are 1230mm long circular ducts (238mm ∅). The outlet ducts are 1475mm long with a rectangular section of 152 × 235mm. Two 55mm long transition ducts were used to join the two outlet ducts to the outlets on the fan casings.

Warm exhaust inlet air was heated by a 3kW electric duct heater, as shown in photograph 7.2. Cool supply inlet air was chilled using a heat exchanger supplied with chilled water. The outputs from the heater and chiller can be controlled enabling the inlet air to be provided at a range of constant temperatures. The volume flow rates can be controlled by changing the speed of rotation or by adjusting the openings at the end of the outlet ducts.

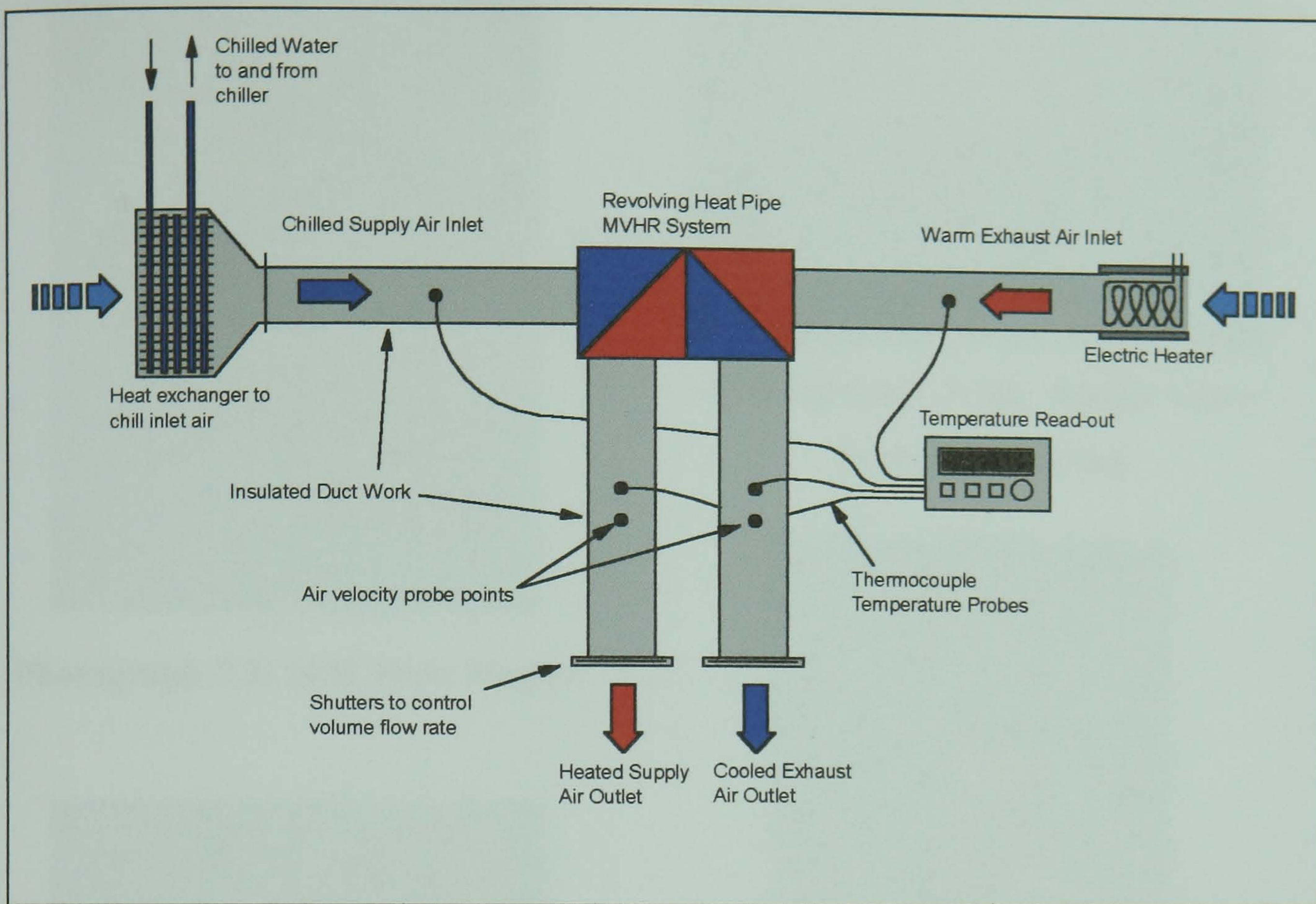
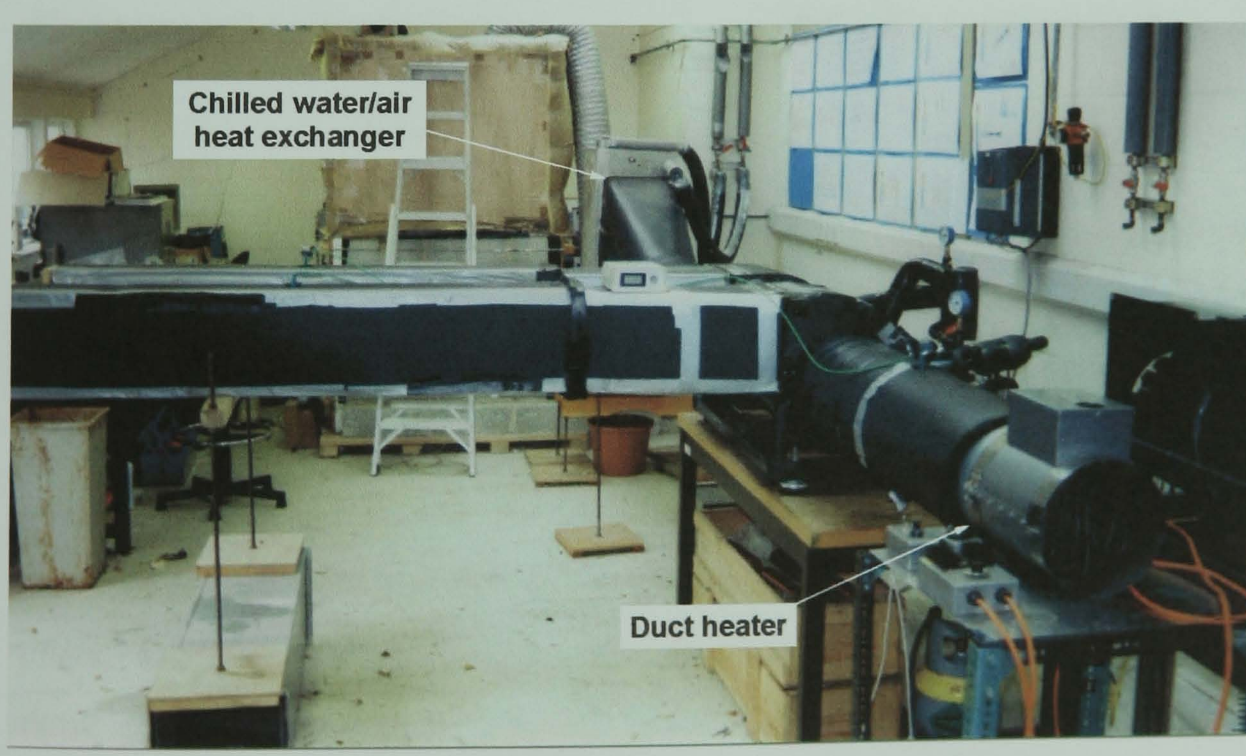
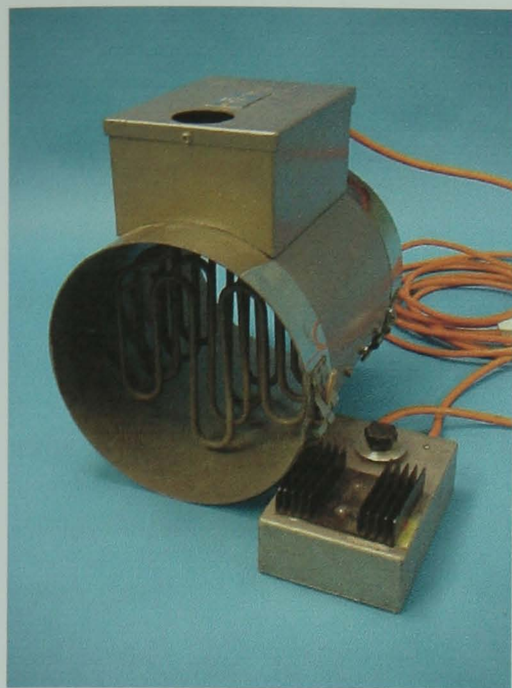


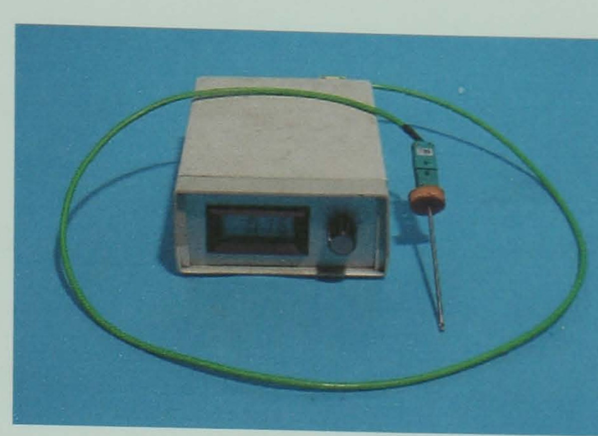
Figure 7.1: Schematic of Ventilation Heat Recovery Test Set-up



Photograph 7.1: Laboratory Set-up for Ventilation Heat Recovery Tests



Photograph 7.2: 3kW Duct Heater



Photograph 7.3: Air Temperature Probe and display

Photograph 7.4: Air velocity Probe (hot wire anemometer)

Photograph 7.5: Tachometer

The temperature of the air in the ducts was measured using k-type thermocouple probes, as shown in Photograph 7.3. Volume flow rates were obtained by measuring velocities in the outlet ducts using the hot wire anemometer shown in Photograph 7.4. Temperature and velocity measurements were taken at four points across the duct sections. This enabled the calculation of average figures for values of temperature and velocity in the ducts. The electrical power consumption of the motor was obtained from a three phase energy meter (model ELCONTROL VIPD3). The speed of rotation of the heat pipe bundle was measured using the digital contact tachometer shown in Photograph 7.5.

Results were obtained for several different speeds of rotation. Inlet temperatures were kept constant for a full range of speeds. At each speed of rotation, the volume flow rates were altered to cover a range between a maximum and minimum value. The flow rates were altered using the control shutters. The exhaust and supply air flow rates, measured in the outlet ducts, were kept the same at each volume flow rate so that the heat exchanger efficiency, η_t, was close to the heat exchanger effectiveness, ε. Readings for temperature and power were recorded at each volume flow rate. The procedure was repeated for the following sets of inlet temperatures: 0°C and 20°C; 0°C and 25°C; 0°C and 30°C; 5°C and 20°C; 5°C and 25°C; 5°C and 30°C, where the former and latter values refer to the supply and exhaust air inlet temperatures respectively. The test results can be found in Appendix D.

The results of the tests on each of the prototypes are discussed in the following sections.

7.1.1 Nine pipe, wire fin, 10% fill:

The prototype was tested at 200rpm intervals between the rotation speeds 100rpm and 1480rpm. Over the range of rotational speeds, inlet air temperatures and volume flow rates between 13m^3/hr and 823m^3/hr, the temperature efficiency ranged between 16% and 67%. The temperature efficiency decreased with increased volume flow rate. The power input to the system ranged between 84.1Watts and 539Watts. Power consumption was greatest at the higher speeds and increased with volume flow rates. The coefficients of performance (COPs) range between 0.15 and 9.51. The COPs increased with increased volume flow rate. Figure 7.2 illustrates typical characteristics for the prototype system for inlet temperatures of 0°C and 20°C. Figure 7.3 illustrates the effect of flow rate on efficiency for a range of inlet temperatures at 900rpm running speed. The relationships of COP against flow and power against flow are almost linear. To illustrate the relationship between efficiency and flow rate, lines of best fit (linear trend) were plotted.

The outlet volume flow rates were the same for each set of results so that the temperature efficiency, η_t, was close to the heat exchanger effectiveness, ε. The effectiveness is based on the ratio of mass flow rates in the supply and exhaust air flows. The effectiveness values will almost be the same as the efficiency values, because the temperatures and corresponding densities of air in each outlet are very similar. An example calculation of the heat exchanger efficiency and effectiveness is given below.

For the following conditions:

rpm = 100, $T_1 = 0$ °C, $T_2 = 7$ °C, $T_3 = 20$ °C, $T_4 = 13$ °C (exhaust air outlet)

Air density at 7°C = 1.26kg/m^3 and at 13°C = 1.23kg/m^3,

flow rate = 77.2m^3/hr.

Heat exchanger efficiency,

From equation 3.1:

$$\eta_t = \left(\frac{T_2 - T_1}{T_3 - T_1}\right)$$

$$\eta_t = \left(\frac{7-0}{20-0}\right)$$

$$= \mathbf{35\%}$$

Heat exchanger effectiveness,

From equation 3.3:

$$\varepsilon = \frac{m_s(T_2 - T_1)}{m_e(T_3 - T_1)}$$

$$\varepsilon = \frac{77.2 \times 1.26(7-0)}{77.2 \times 1.23(20-0)}$$

$$= \mathbf{35.8\%}$$

This demonstrates that the values of efficiency and effectiveness are very similar. This will generally be true for all values of temperature studied, since the density of air over the conditions varies only slightly.

7.1.2 Nine pipe, wire fin, 20% fill:

The prototype was tested at 500, 900 and 1300rpm. For volume flow rates between 13m^3/hr and 720m^3/hr the efficiency ranged between 20% and 55%, the input power ranged between 116 and 400Watts, and the COP ranged between 0.15 and 8.69.

For the same operating speeds and flow rates, the power input to this system, compared to the power input to the 10% charged system, was more. This is probably because more power is required to drive the heavier heat pipes with more working fluid. The average efficiencies for the 10% charged system and the 20% charged system were almost the same. For speeds of 500, 900 and 1300rpm and over the full range of flow rates and inlet temperatures, the average efficiency for the 10% charged system was 39.1% and the average efficiency for the 20% charged system was 39.7. Figures 7.4 and 7.5 illustrate typical system characteristics.

7.1.3 Nine pipe, plate fin, 10% fill:

For speeds of 500, 900 and 1300rpm over a range of flow rates between 26m^3/hr and 707m^3/hr the efficiency ranged between 20% and 50%. The average efficiency over the range of conditions was 34%. The power input ranged between 110Watts and 380Watts and the COPs ranged between 0.18 and 5.9. Figures 7.6 and 7.7 illustrate typical system characteristics. The efficiencies of the plate finned systems are less than for the wire finned systems.

7.1.4 Nine pipe, plate fin, 10% fill, tapered:

The same range of tests were carried out on this system as those carried out on the non-tapered system. The efficiencies ranged between 15% and 60%. The average efficiency over the range of conditions was 36.4%, slightly higher than the non-tapered system. However, the average efficiency at 1300rpm (30%) was lower than the average efficiency at 1300rpm for the non-tapered system (32%). This may be due to the condensate returning at a rate which is higher than the rate of evaporation, resulting in the evaporators becoming flooded which would reduce the efficiency of heat transfer. The component of force along the pipe walls acting on the condensate, due to the angled heat pipes, may be too large at the higher speeds. Thick films reduce conductive heat transfer and nucleate boiling. The input power ranged between 84Watts and 237Watts. The power input is probably less for this system because the clamps used have less mass and, therefore, the load on the motor is less. The COP ranges between 0.25 and 8.92. Figures 7.8 and 7.9 illustrate typical system characteristics.

7.1.5 Twenty four pipe, double row, wire fin, 10% fill:

Of the five different systems tested, this system performed the best. For speeds of 500, 900 and 1300rpm over a range of flow rates between 13m^3/hr and 797m^3/hr the efficiency ranged between 33% and 60%. The input power ranged between 42Watts and 447Watts and the COPs ranged between 0.21 and 26.29. Figures 7.10 and 7.11 illustrate typical system characteristics.

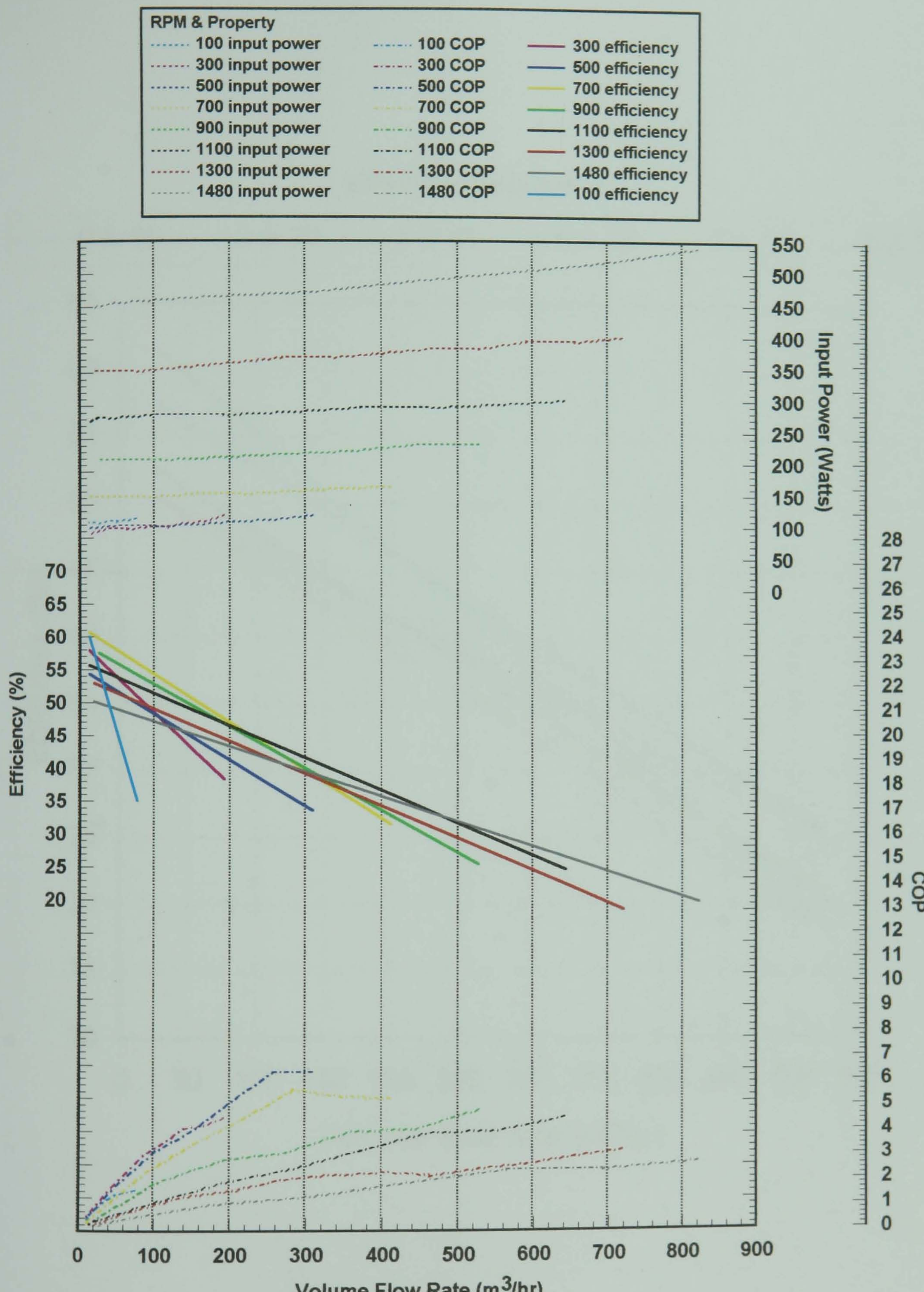


Figure 7.2: Prototype MVHR characteristics

9-pipe, wire fin, 10% fill at 0 °C and 20 °C inlet temperatures

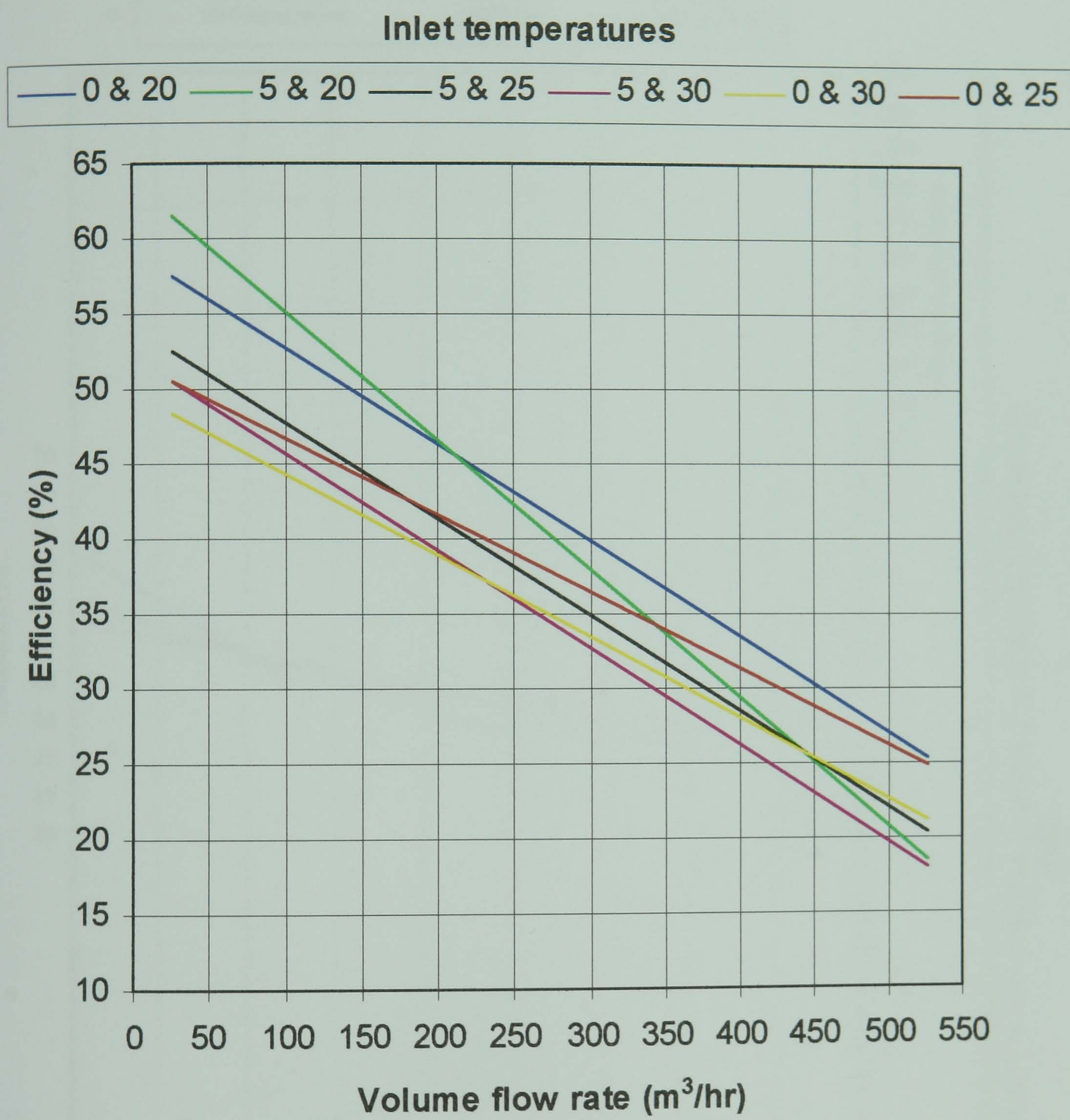


Figure 7.3: Prototype MVHR characteristics

9-pipe, wire fin, 10% fill at 900 rpm

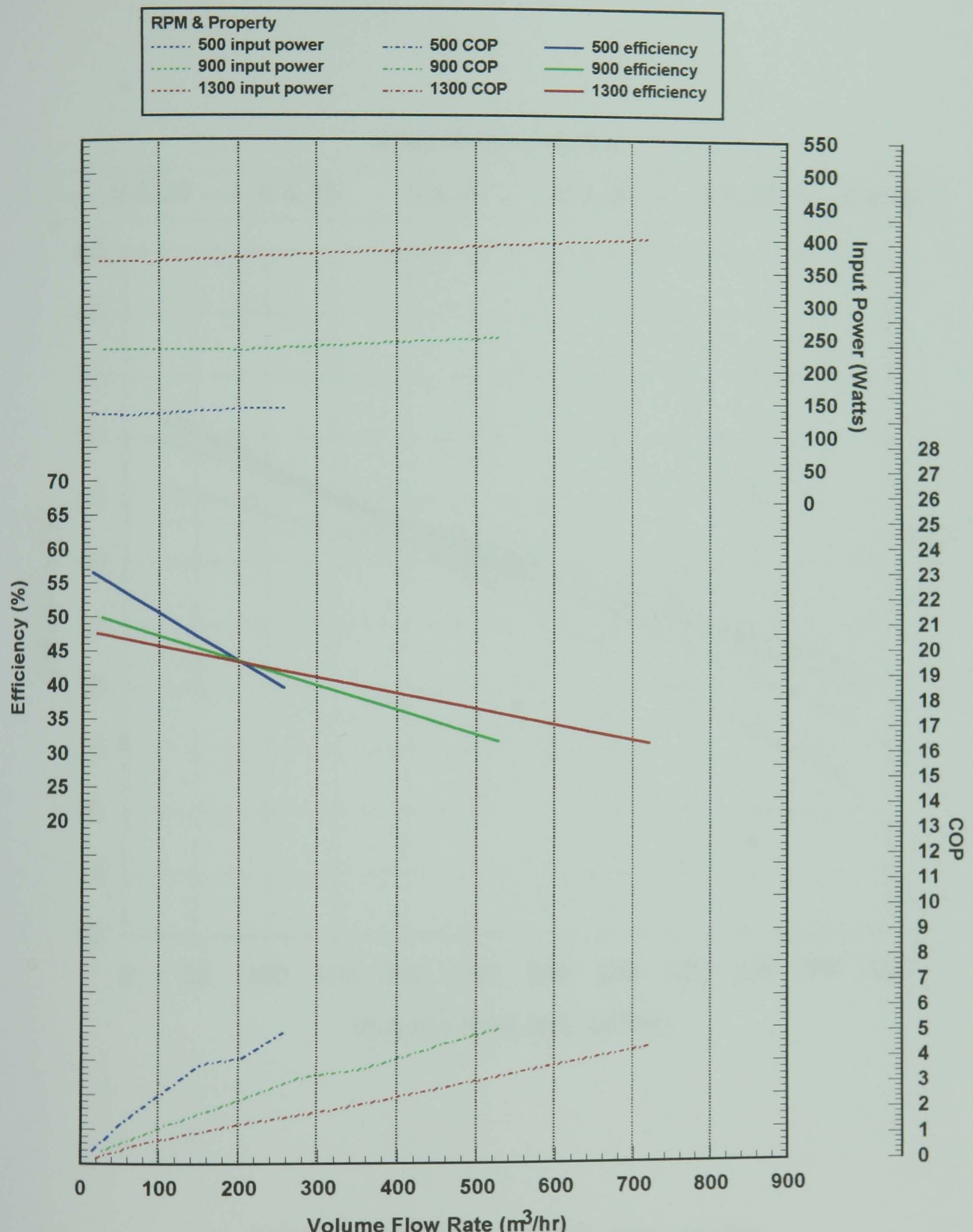


Figure 7.4: Prototype MVHR characteristics

9-pipe, wire fin, 20% fill at 0 °C and 20 °C inlet temperatures

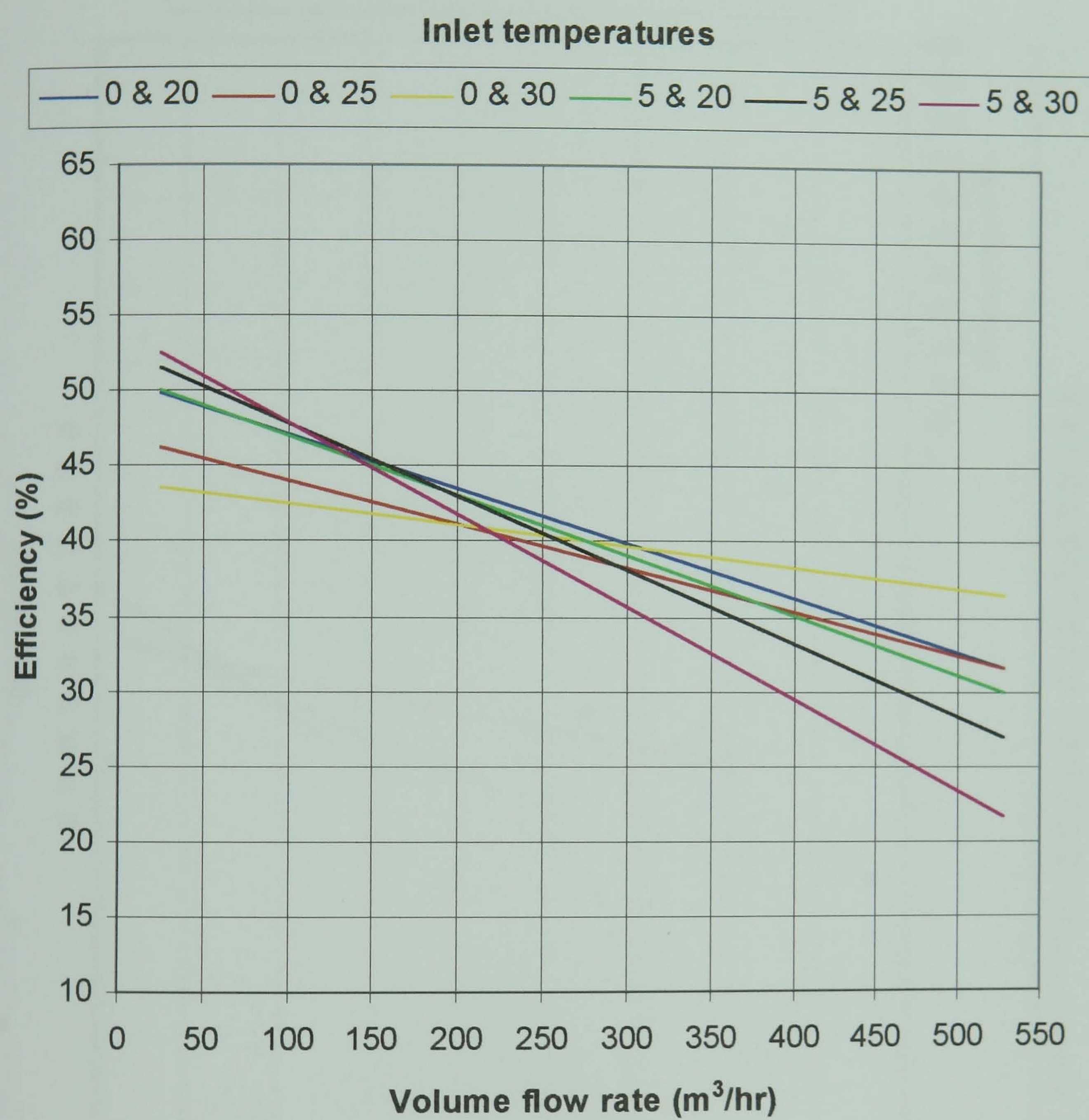


Figure 7.5: Prototype MVHR characteristics

9-pipe, wire fin, 20% fill at 900 rpm

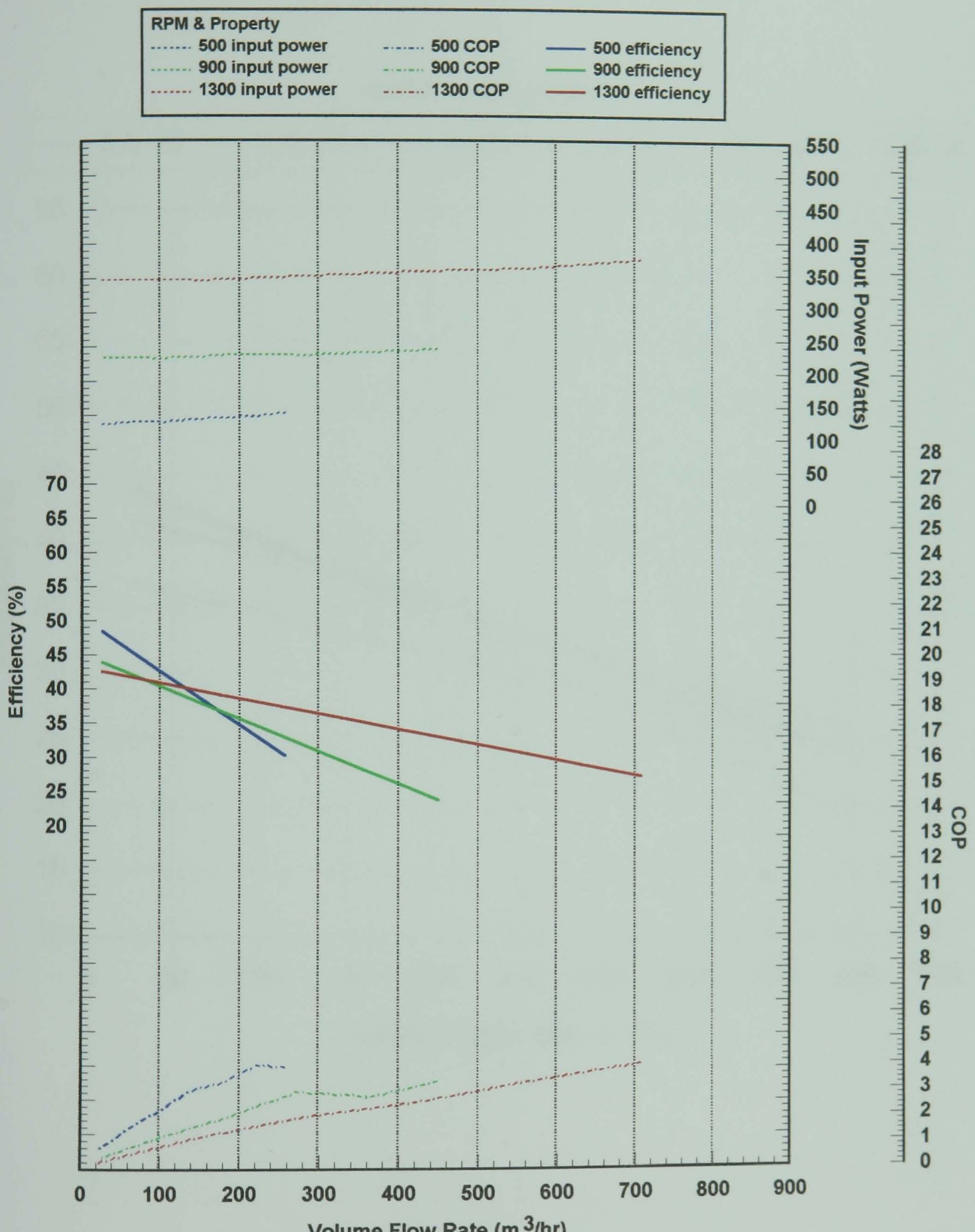


Figure 7.6: Prototype MVHR characteristics

9-pipe, plate fin, 10% fill at 0 °C and 20 °C inlet temperatures

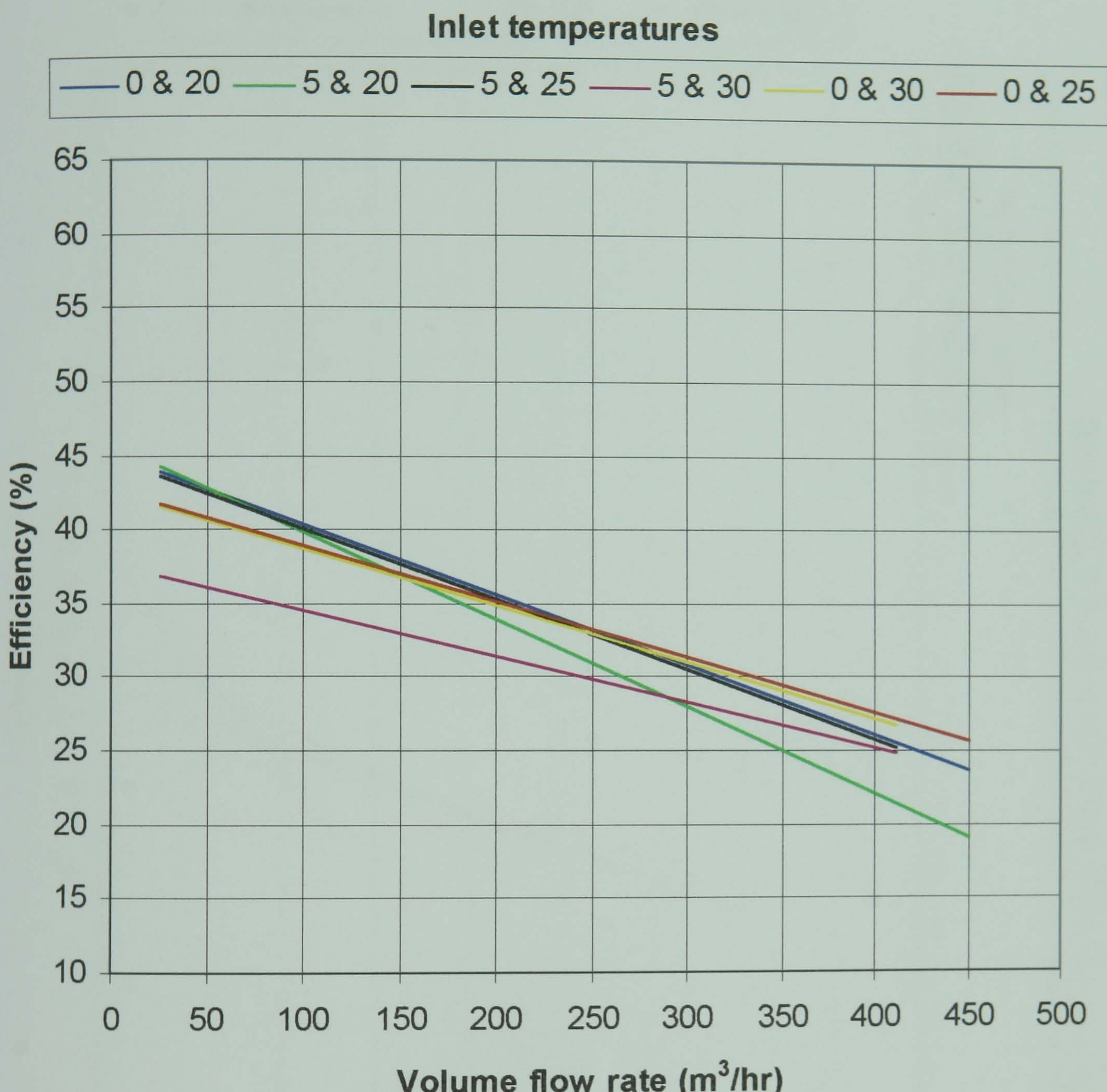


Figure 7.7: Prototype MVHR characteristics

9-pipe, plate fin, 10% fill at 900 rpm

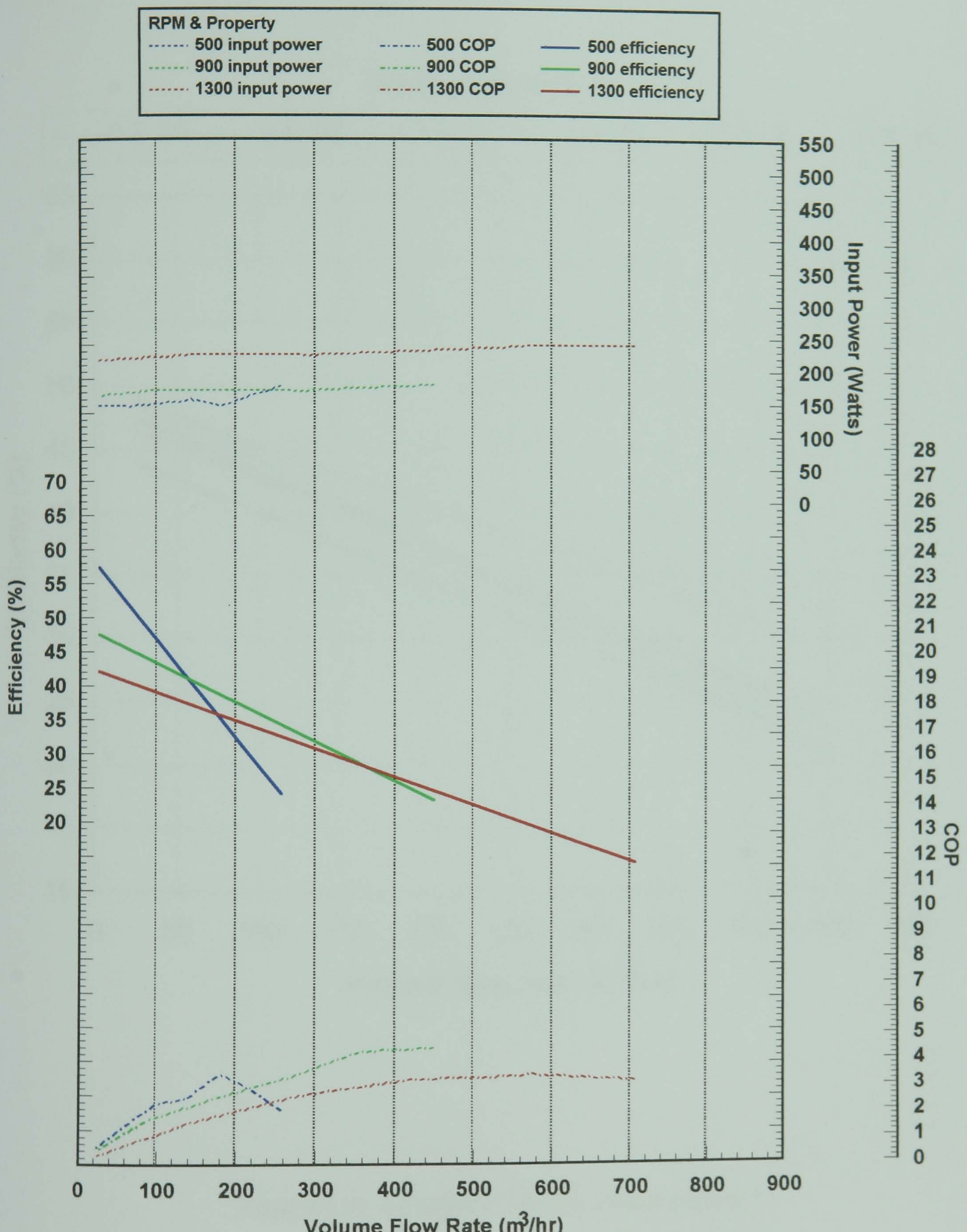


Figure 7.8: Prototype MVHR characteristics

9-pipe, plate fin, 10% fill, tapered at 0 °C and 20 °C inlet temperatures

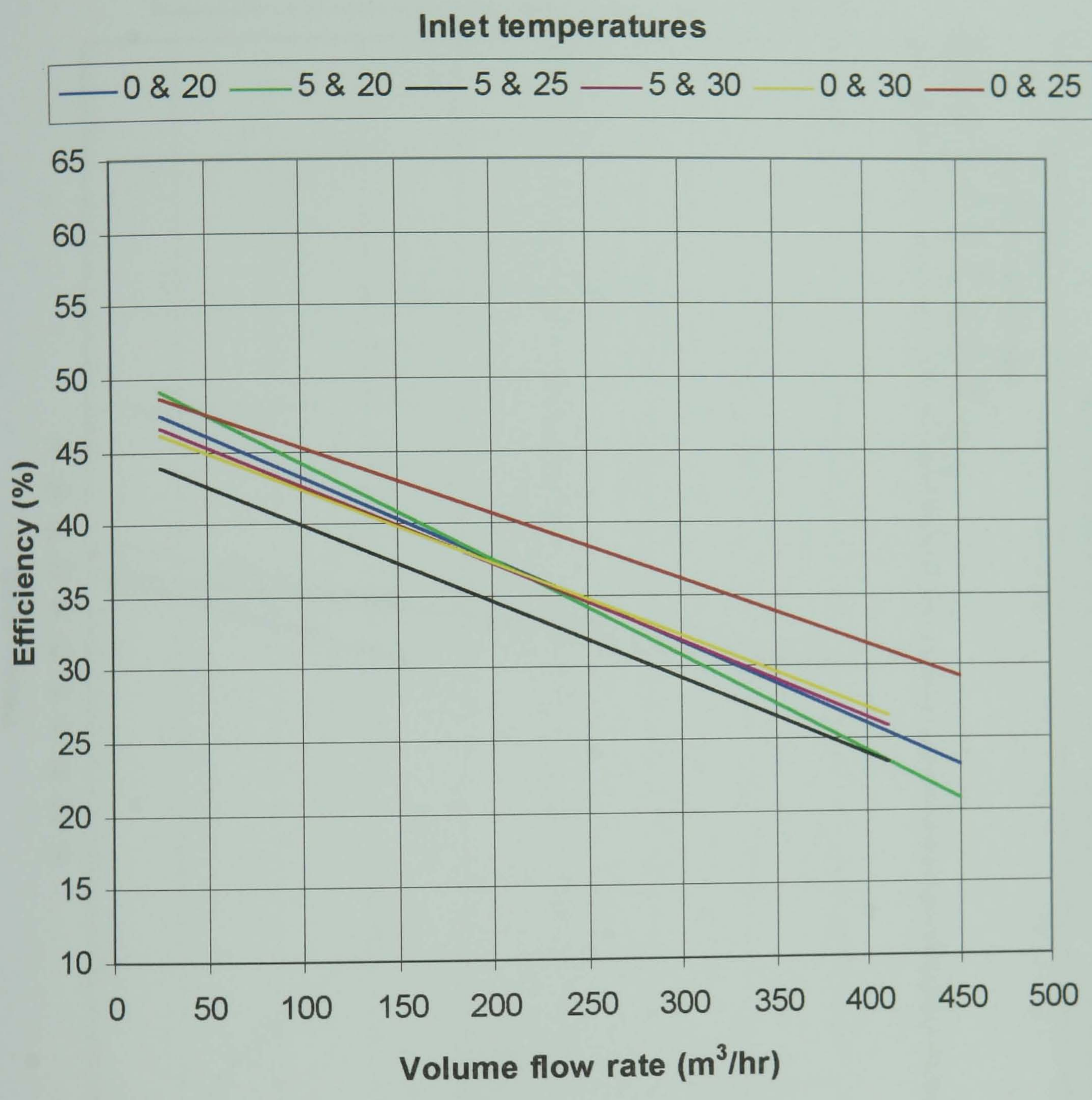


Figure 7.9: Prototype MVHR characteristics
9-pipe, plate fin, 10% fill, tapered at 900 rpm

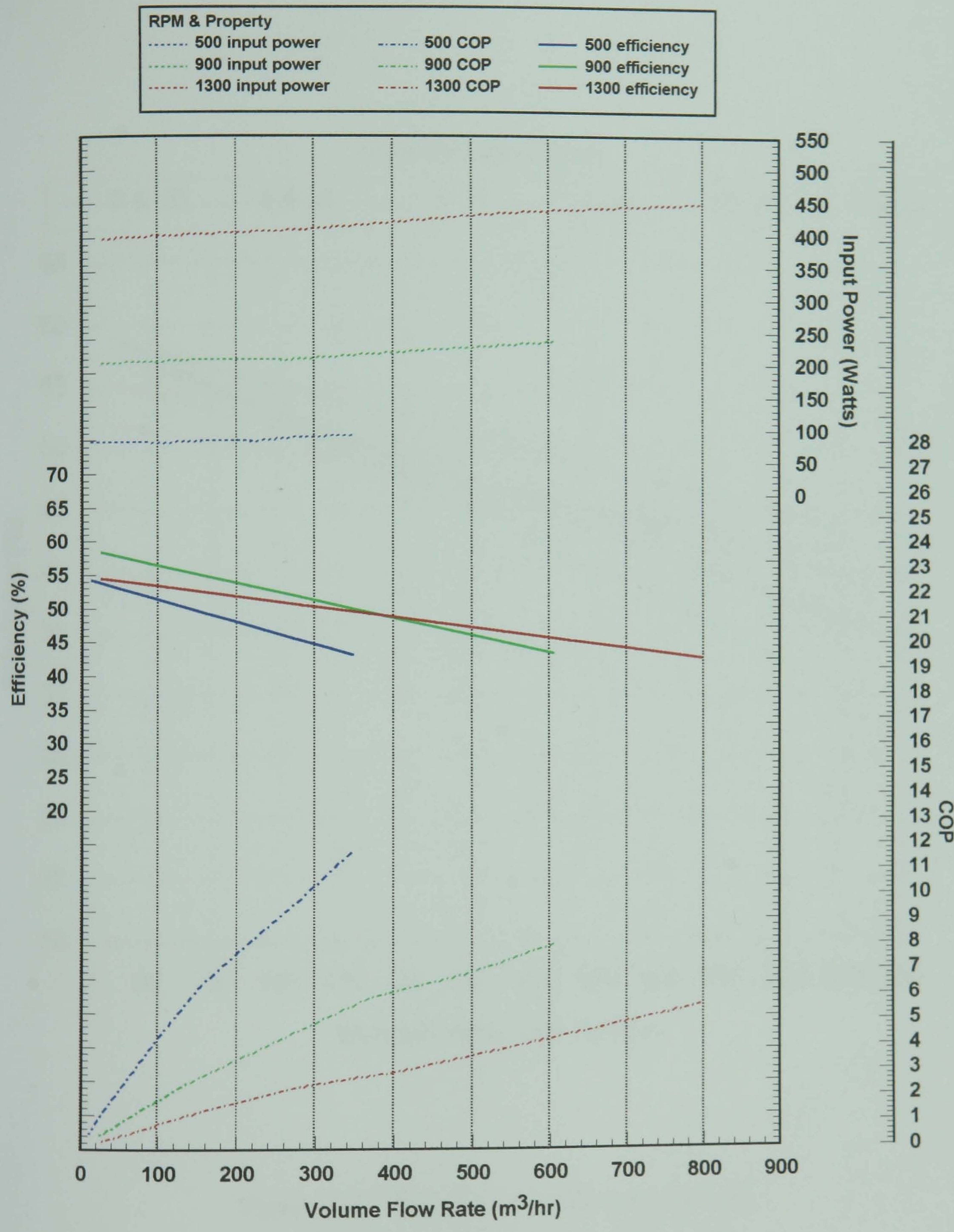


Figure 7.10: Prototype MVHR characteristics

24-pipe, wire fin, 10% fill at 0 °C and 20 °C inlet temperatures

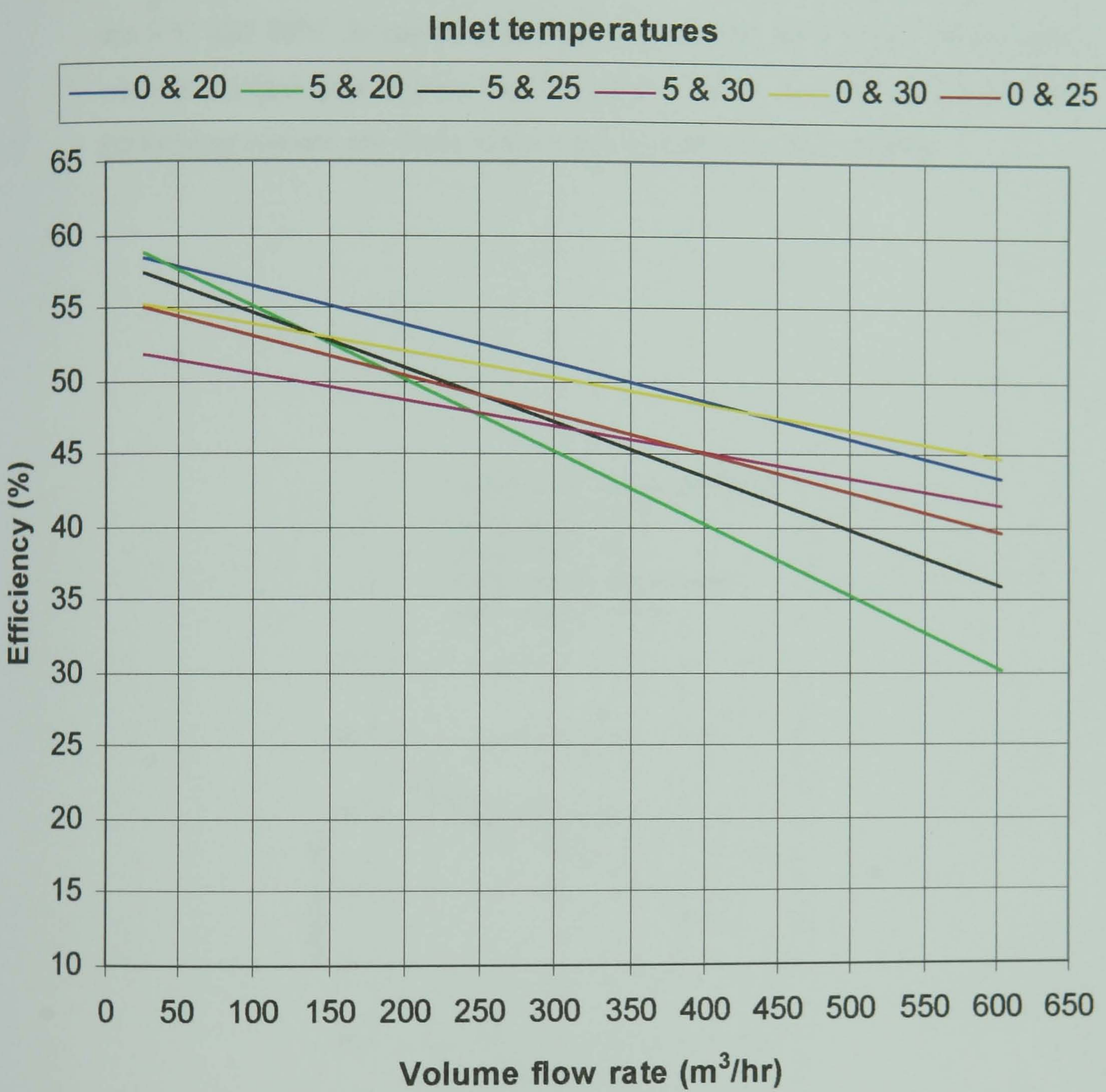


Figure 7.11: Prototype MVHR characteristics

24-pipe, wire fin, 10% fill at 900 rpm

7.1.6 Comparison of prototype systems

Figures 7.12 to 7.14 illustrate the efficiency/flow rate characteristics for the five prototype systems at 500, 900 and 1300rpm when the inlet temperatures are 0°C and 20°C. It can be seen that at all speeds the 24-pipe, double row, wire finned heat pipe system has the highest values of efficiency. The worst performing systems are those where the heat pipes had plate finning.

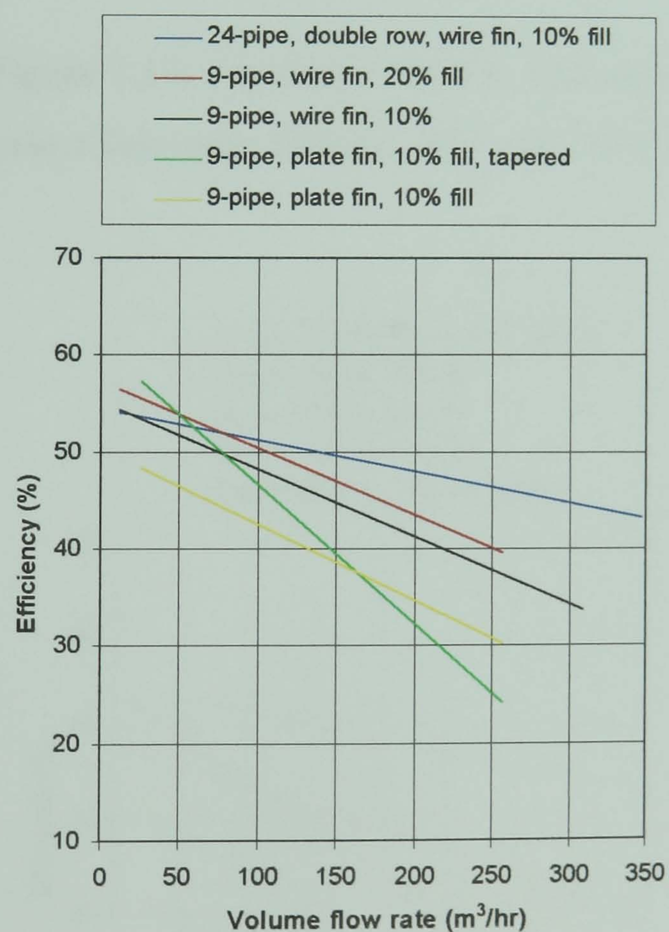


Figure 7.12: Prototype MVHR characteristics:

Efficiency against flow rate, 500rpm, 0°C and 20°C inlet temperatures

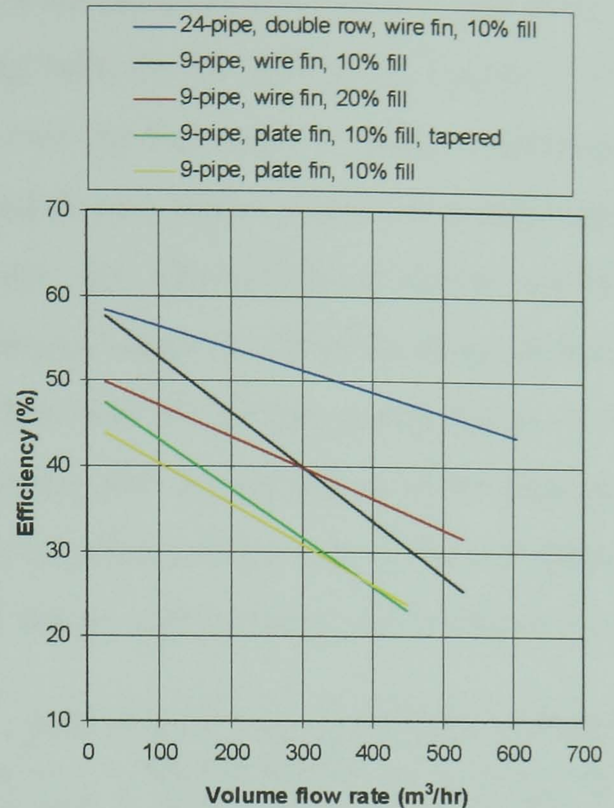


Figure 7.13: Prototype MVHR characteristics:

Efficiency against flow rate, 900rpm, 0°C and 20°C inlet temperatures

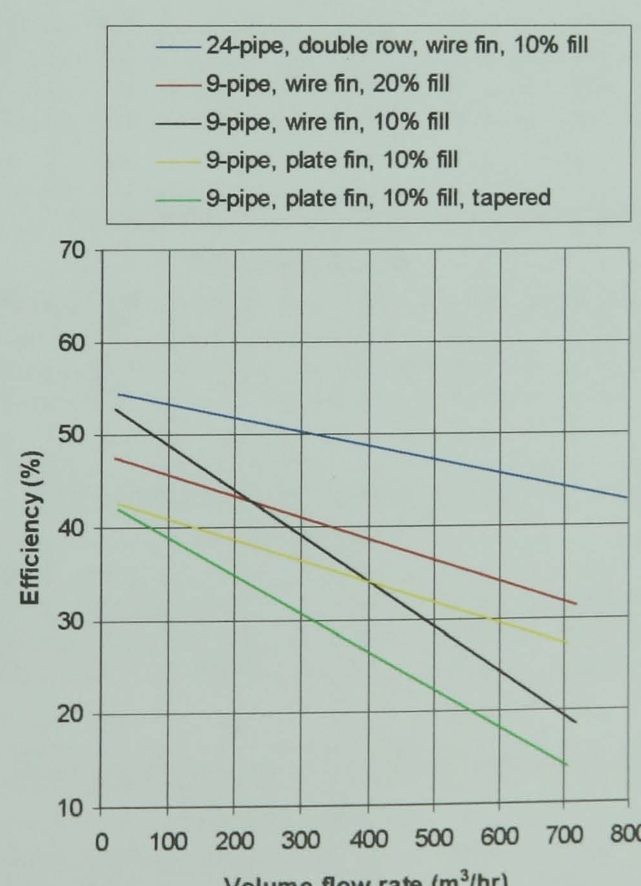


Figure 7.14: Prototype MVHR characteristics:

Efficiency against flow rate, 1300rpm, 0°C and 20°C inlet temperatures

Figures 7.11 to 7.14 do not show conclusively the effect that the working fluid charge and tapering have on the efficiency. Figure 7.15 illustrates the results from all the tests over the full range of inlet temperatures. Consequently, the curves are generated from a larger sample of results which reduces the effects of experimental error. The characteristics shown are for the design flow rate of 200m^3/hr. Efficiency values at 200m^3/hr were found by plotting graphs of efficiency against flow rate for all the results. Lines of best fit were produced for each set of results and the equations corresponding to these lines were used to find values of efficiency for a flow rate of 200m^3/hr. These values are shown in Table 7.1 below with average values shown for each prototype.

9-pipe, wire fin, 10% - temperature efficiencies, η_t at 200m^3/hr								
RPM	Inlet temperatures (°C)						average η_t (same rpm)	average η_t for prototype
	0 & 20	0 & 25	0 & 30	5 & 20	5 & 25	5 & 30		
500	41.2	40.5	39.8	43.7	37.6	39.0	40.3	
900	46.3	41.6	38.9	46.5	41.3	39.3	42.3	
1300	44.0	44.8	41.3	49.6	42.3	41.4	43.9	42.2
average η_t (same temps)	43.9	42.3	40.0	46.6	40.4	39.9		
9-pipe, wire fin, 20% - temperature efficiencies, η_t at 200m^3/hr								
RPM	Inlet temperatures (°C)						average η_t (same rpm)	average η_t for prototype
	0 & 20	0 & 25	0 & 30	5 & 20	5 & 25	5 & 30		
500	43.5	39.1	38.4	38.3	33.9	40.4	38.9	
900	43.5	41.1	41.1	43.0	43.1	41.9	42.3	
1300	43.4	41.5	42.0	41.8	41.9	38.4	41.5	40.9
average η_t (same temps)	43.5	40.6	40.5	41.0	39.6	40.2		
9-pipe, plate fin, 10% - temperature efficiencies, η_t at 200m^3/hr								
RPM	Inlet temperatures (°C)						average η_t (same rpm)	average η_t for prototype
	0 & 20	0 & 25	0 & 30	5 & 20	5 & 25	5 & 30		
500	34.7	33.6	32.0	27.1	34.3	27.8	31.6	
900	35.6	35.1	34.8	34.0	35.3	31.4	34.3	
1300	38.6	33.8	34.7	36.3	34.6	31.7	34.9	33.6
average η_t (same temps)	36.3	34.2	33.8	32.5	34.7	30.3		
9-pipe, plate fin, tapered, 10% - temperature efficiencies, η_t at 200m^3/hr								
RPM	Inlet temperatures (°C)						average η_t (same rpm)	average η_t for prototype
	0 & 20	0 & 25	0 & 30	5 & 20	5 & 25	5 & 30		
500	32.312	38.019	37.928	29.884	35.319	38.438	35.3	
900	37.526	40.672	37.264	37.519	34.556	37.245	37.5	
1300	34.838	36.151	35.269	37.289	34.902	30.584	34.8	35.9
average η_t (same temps)	34.9	38.3	36.8	34.9	34.9	35.4		
24-pipe, double row, wire fin, 10% - temperature efficiencies, η_t at 200m^3/hr								
RPM	Inlet temperatures (°C)						average η_t (same rpm)	average η_t for prototype
	0 & 20	0 & 25	0 & 30	5 & 20	5 & 25	5 & 30		
500	48.025	48.805	51.213	47.926	47.668	47.807	48.6	
900	53.82	50.393	52.084	50.12	50.933	48.707	51.0	
1300	51.797	47.991	47.634	52.872	53.434	49.966	50.6	50.1
average η_t (same temps)	51.2	49.1	50.3	50.3	50.7	48.8		

Table 7.1: Temperature Efficiencies for MVHR Prototypes at 200m^3/hr

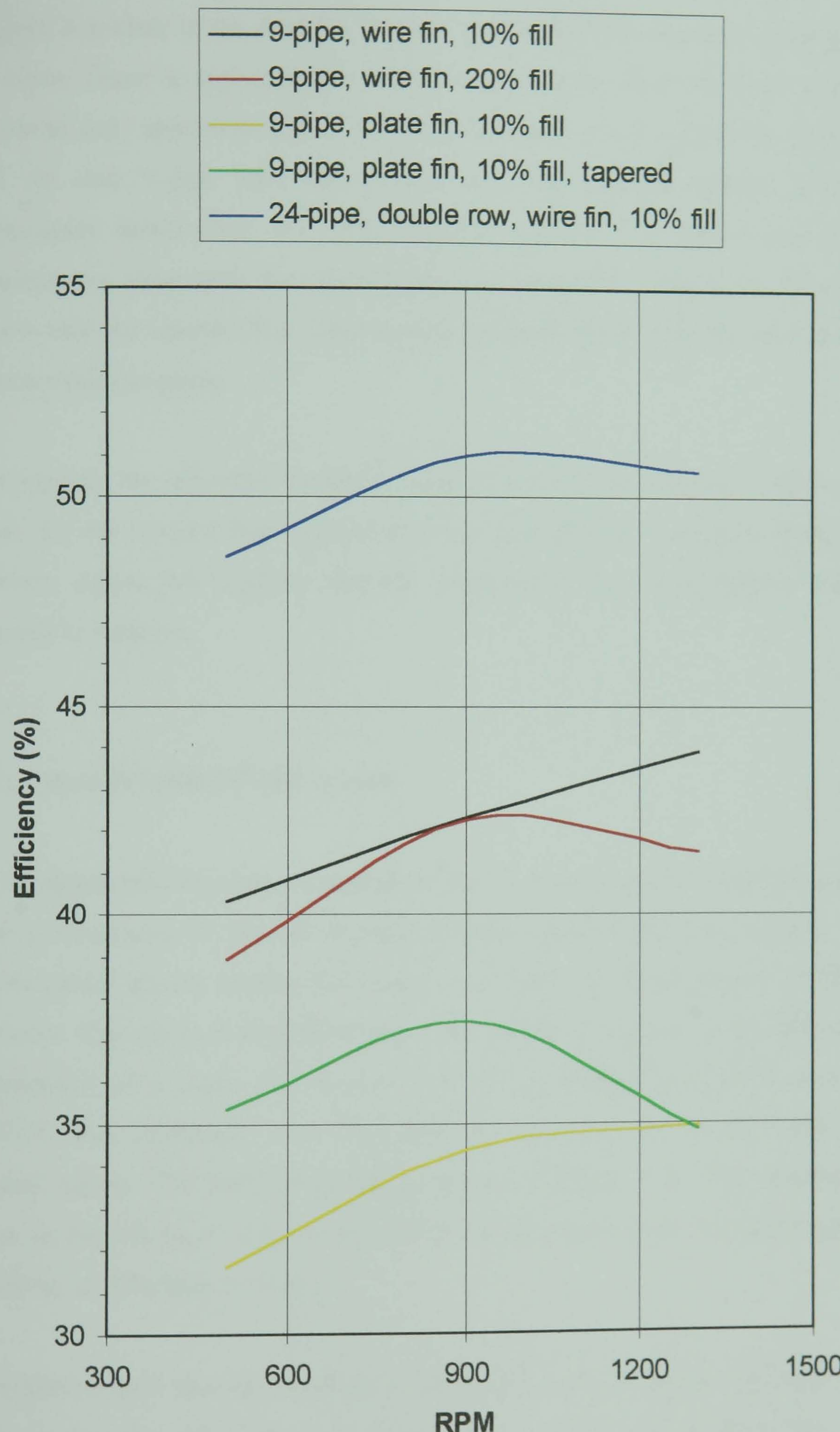


Figure 7.15: Prototype MVHR characteristics: Efficiency against RPM at 200m^3/hr

Again, it is clear to see that the 24-pipe system performs better than the other systems. There is, approximately, a 10% improvement over the 9-pipe wire fin systems and, approximately, a 15% improvement over the plate fin systems. Of the two 9-pipe wire fin systems the 10% charged system performs marginally better than the 20% charged system. The worst performing systems are those with the plate finned heat pipes. The chart does, however, show that the tapered heat pipe bundle performs better than the non-tapered horizontal heat pipes.

In general, the efficiency increases with increased rpm. However, this is not true for the tapered heat pipe system. At 1300rpm the average efficiency is lowest. Again this suggests that the evaporators may be flooded at higher speeds of rotation.

7.1.7 Baxi WH300 MVHR system

A domestic MVHR system available in the UK was tested to obtain an idea of the performance of current systems already employed in UK housing. The commercial system chosen for testing was Baxi Air Management's WH300 system. The unit is a cross flow plate heat exchanger system which essentially comprises of a case, two forward curved centrifugal fans and a polymer UPVC heat exchanger core. The two fans are powered by 220-240V a.c. mains supply. The WH300 system is shown in Figure 7.16. Baxi market the unit in the UK as a 'state of the art' cross flow plate heat exchanger system with up to 70% heat recovery.

Ventilation heat recovery tests were carried out using the same procedure used for testing the revolving heat pipe systems. The volume flow rate was controlled by altering the fan speed. At each fan speed, measurements of temperature, flow rate, voltage and current were taken to enable the calculation of temperature efficiency and power. Test results are given in Appendix D and Figure 7.17 illustrates efficiency and power against flow rate for a range of different inlet temperatures.

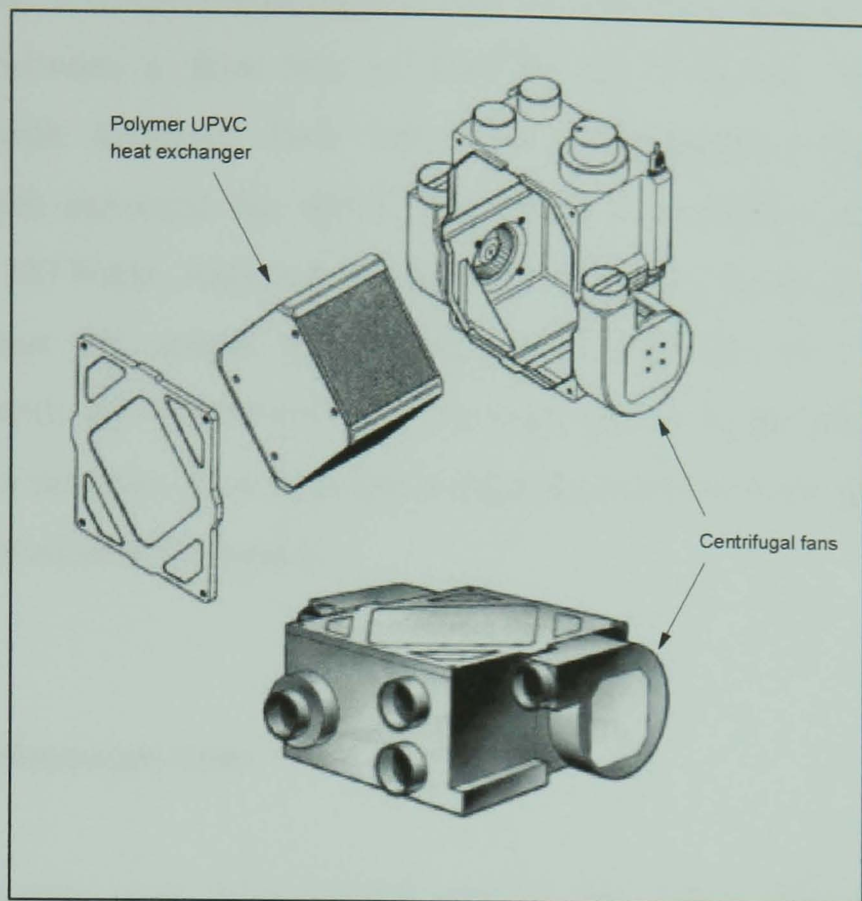


Figure 7.16: Baxi WH300 MVHR System

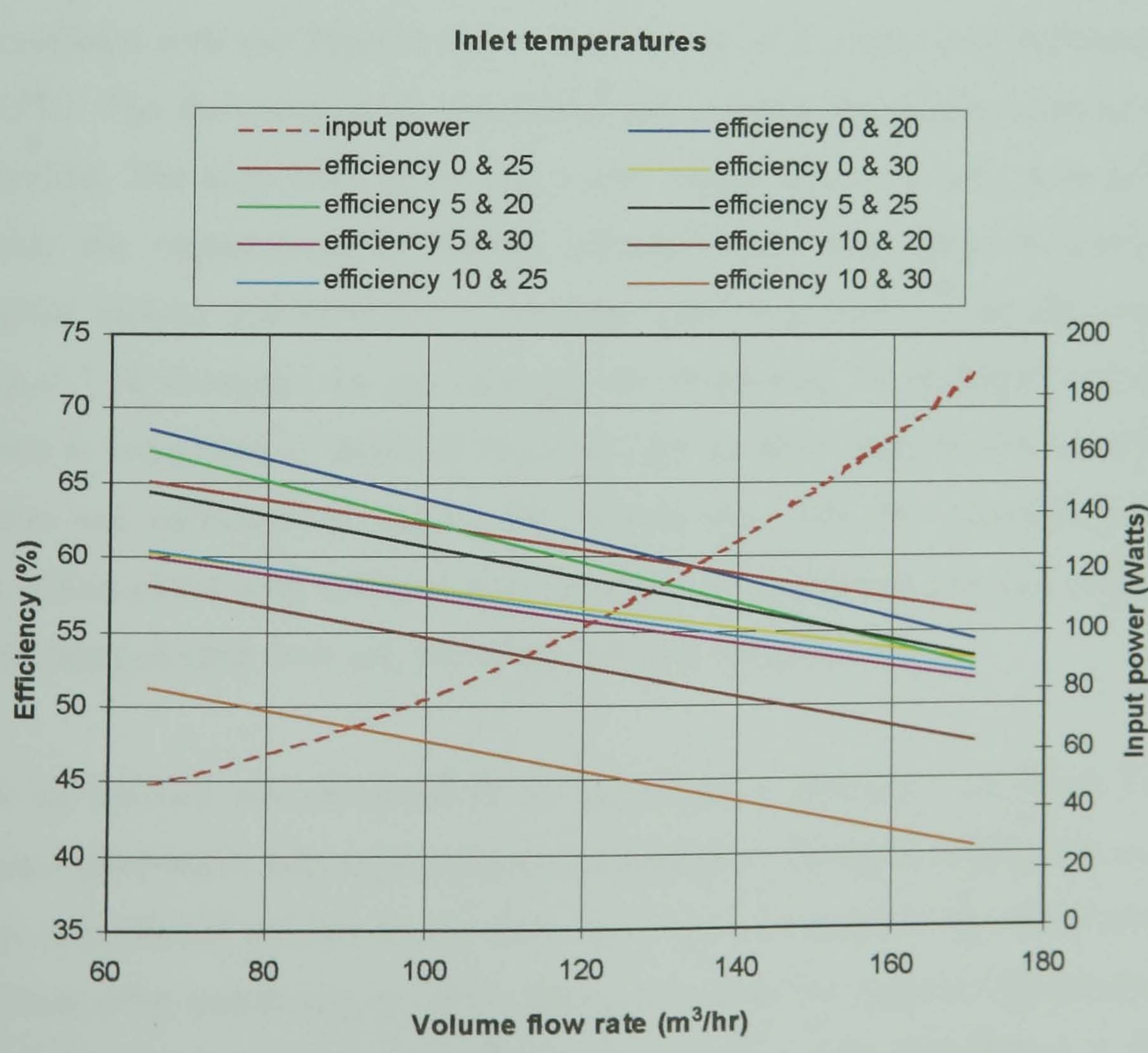


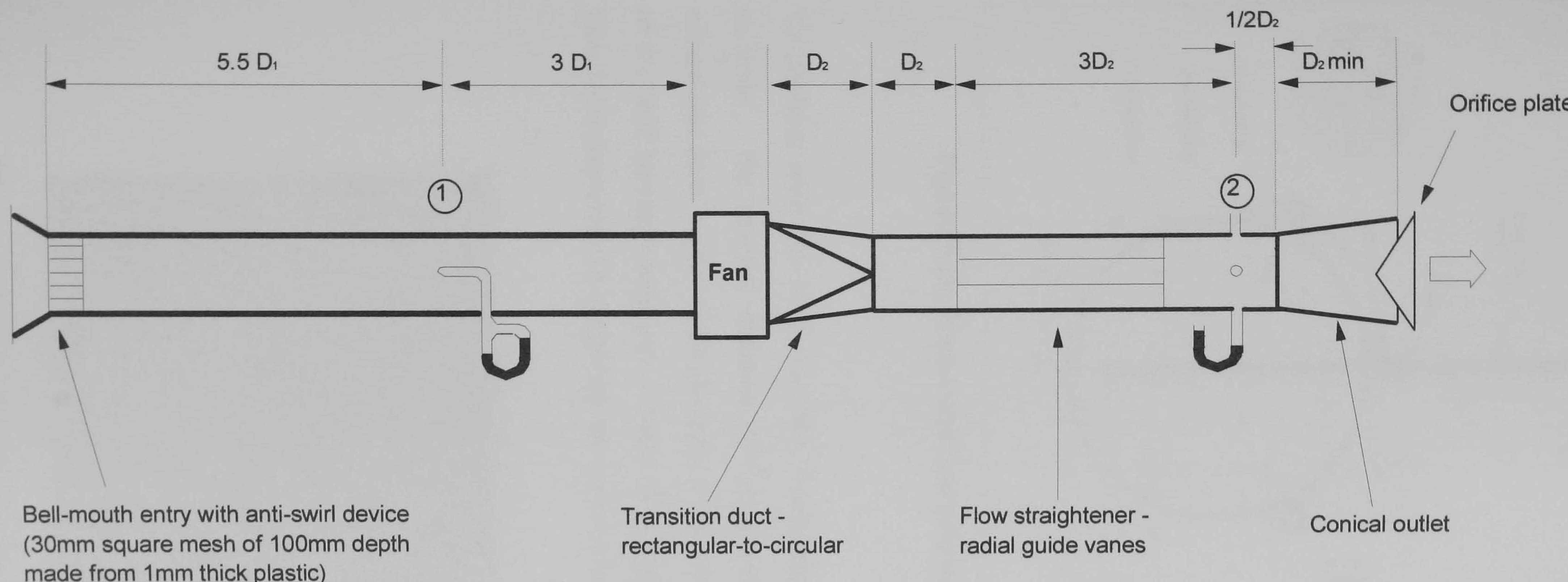
Figure 7.17: Baxi WH300 Ventilation Heat Recovery Characteristics

Testing of the WH300 system showed that the efficiency ranged between 40% and 70% between a flow rate of $65m^3/hr$ and $170m^3/hr$. The efficiency decreased with increased flow rate. Power consumption and flow rate increased with increased fan speed. The power consumption increased from 48Watts to 187Watts. Although the system is meant to supply up to $200m^3/hr$ at the highest fan speed, only a maximum flow rate of $171m^3/hr$ was achievable with the ventilation heat recovery set up in the laboratory. The reduced flow rate may have been due to high losses in the ducts and across the chilled-water/air heat exchanger.

7.2 Fan performance tests

Fan performance tests were carried out on the 9-pipe wire and plate fin prototypes, the 24-pipe wire fin prototype and the Baxi WH300 system to determine their pressure flow characteristics. The tests were undertaken in accordance with the Type D test – ducted inlet and outlet, from BS848:Part 1:1980. The ductwork used was fabricated to meet the requirements of the Standard. The tests were performed on the supply airways of each system. To satisfy the requirements of BS848, measurements were taken in ducts of circular section and therefore a transition piece was used on the fan outlet. Figure 7.18 illustrates the experimental set-up showing the attached ductwork which is comprised of different elements such as flow straighteners, transition pieces and measurement points. The volume flow rate was controlled using six orifice plates with different sized openings. Velocity and pressure readings were taken at each flow rate including fully open and closed.

The air velocity was measured in the inlet duct at position 1 of Figure 7.18, using a pitot-static tube connected to a manometer. The head of the pitot-static tube was aligned parallel to the duct and was positioned at the minimum of 24 measuring points spaced along three symmetrically disposed diameters of the duct. The location of measurements complies with BS848:Part 1. The average of these values was used to calculate the flow rate. Figure 7.19 illustrates the pitot-static traverse measuring points.



The lengths shown on the diagram are shown as a proportion of the diameter at point 1 (D_1=238mm) and point 2 (D_2=220mm)

Figure 7.18: Fan Performance Test Set-up:
Ducted Inlet and outlet, Type D Test, BS848:Part 1:1980

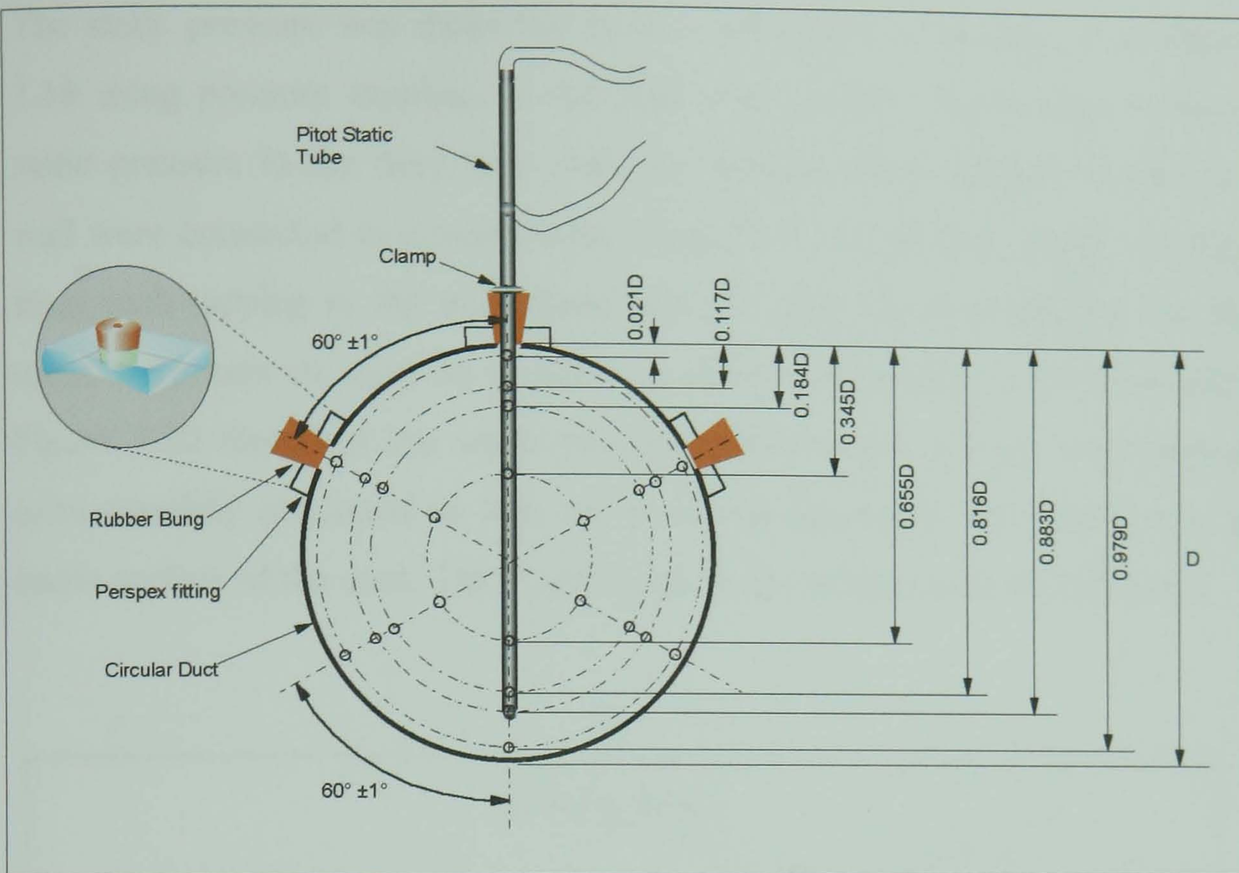
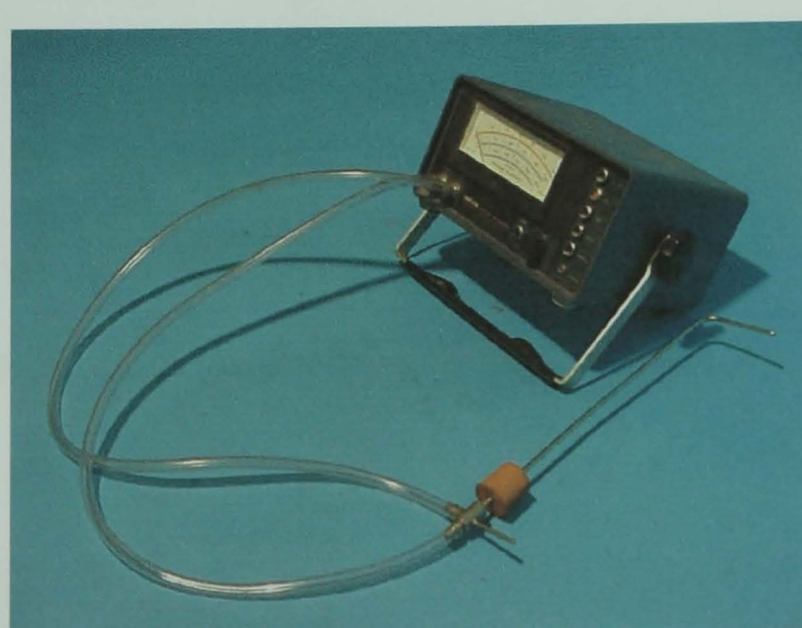


Figure 7.19: Pitot-static traverse measuring points

To conform with BS 1042-2.1:1983, 'Measurement of fluid flow in closed conduits', the external diameter of the pitot-static tube (3.9mm) did not exceed one forty eighth of the diameter of the duct (238mm) and the diameter of the total pressure hole was 1.1mm. Photograph 7.6 shows the pitot-static tube and manometer for measuring the air velocity in the duct.



Photograph 7.6: Pitot-static tube and Manometer

The static pressure was measured in the outlet duct at position 2 of Figure 7.18 using pressure tappings in the duct wall. To find the average value of static pressure in the duct, four pressure tappings perpendicular to the duct wall were connected to a manometer using PVC tubing. The length of tubing from each tapping to the manometer was the same for each tapping. All the tubes were kept as short as possible to reduce inaccuracy in measurements. Figure 7.20 illustrates the static pressure measurement set-up. The tappings were carefully produced so that the bore was normal to and flush with the inside surface of the duct. They were sized in accordance with BS848:Part 1.

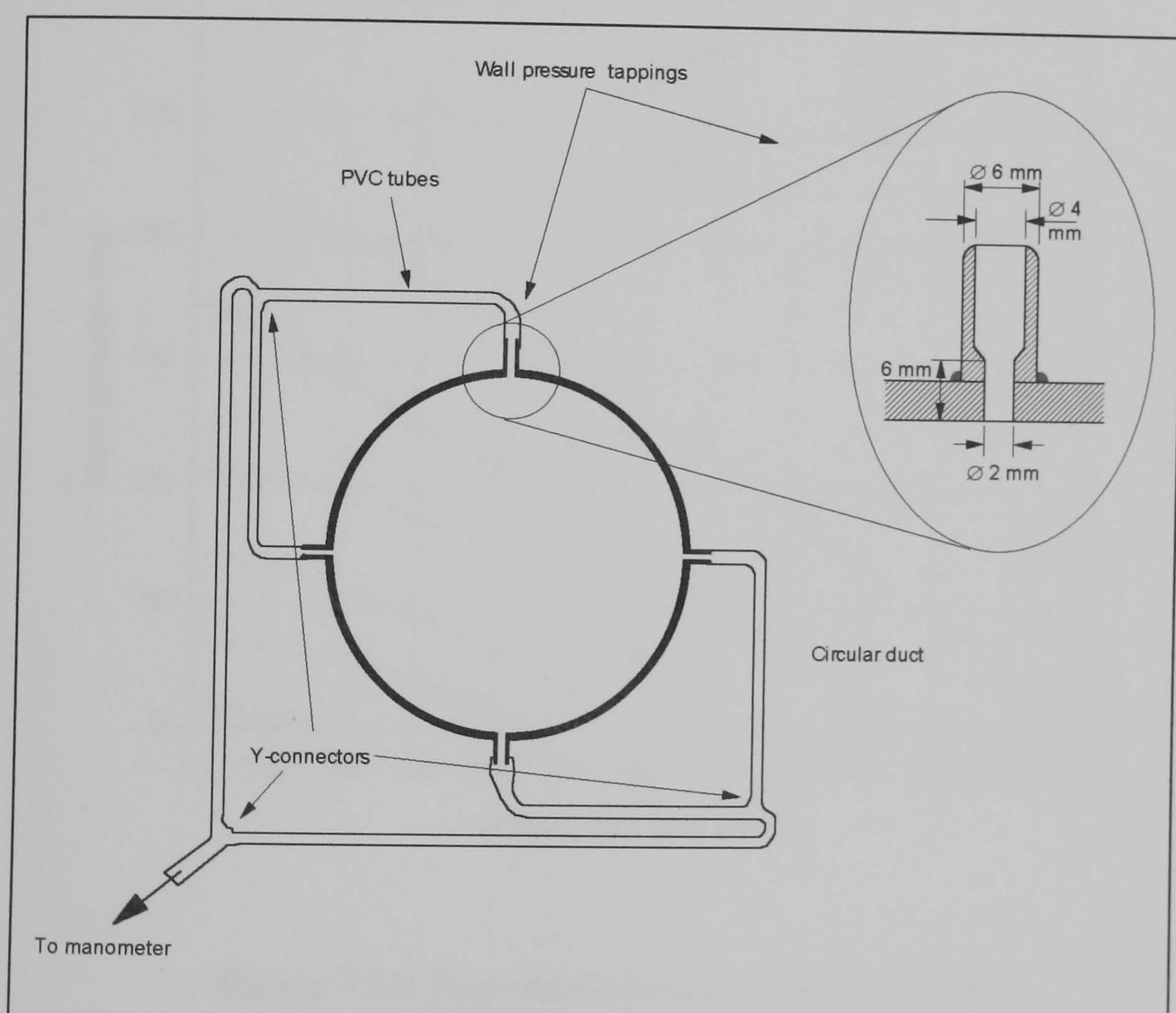


Figure 7.20: Static Pressure Measurement Set-up

All readings of velocity and pressure were taken once the flow in the ducts had stabilised. The measurements were repeated for a range of fan speeds.

7.2.1 Fan performance test results and discussion

The results of the fan performance tests can be found in Appendix E. Figures 7.21 to 7.24 illustrate the static pressure/flow rate relationships established from testing. The curves shown are lines of best fit through the results of tests over a range of fan speeds.

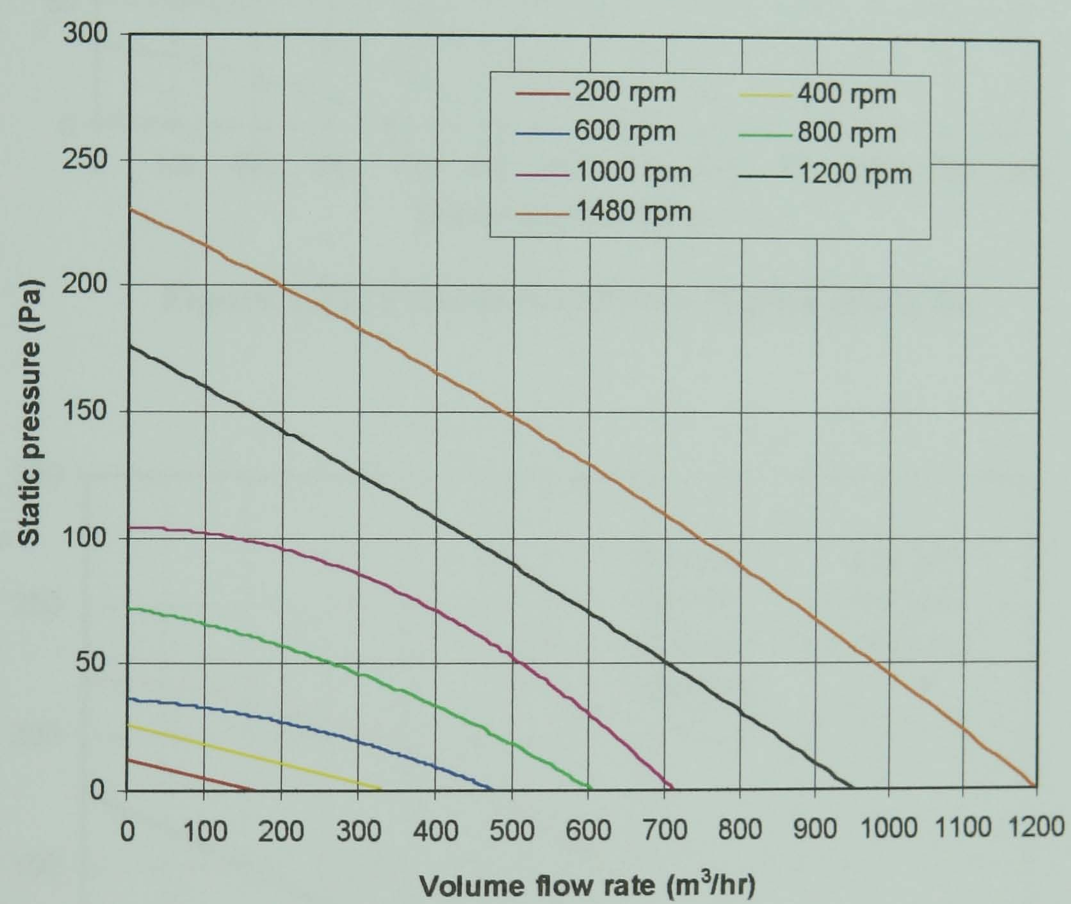


Figure 7.21: Fan characteristic: 9-pipe, wire fin

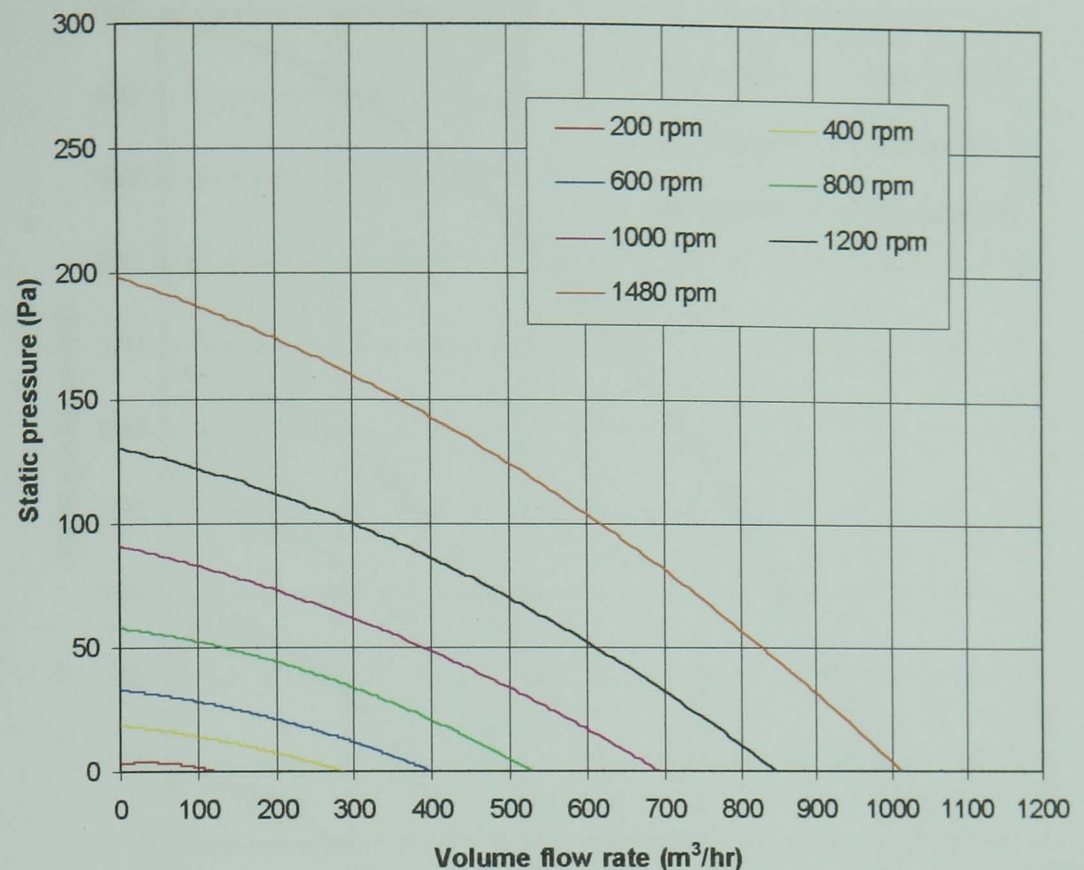


Figure 7.22: Fan characteristic: 9-pipe, plate fin

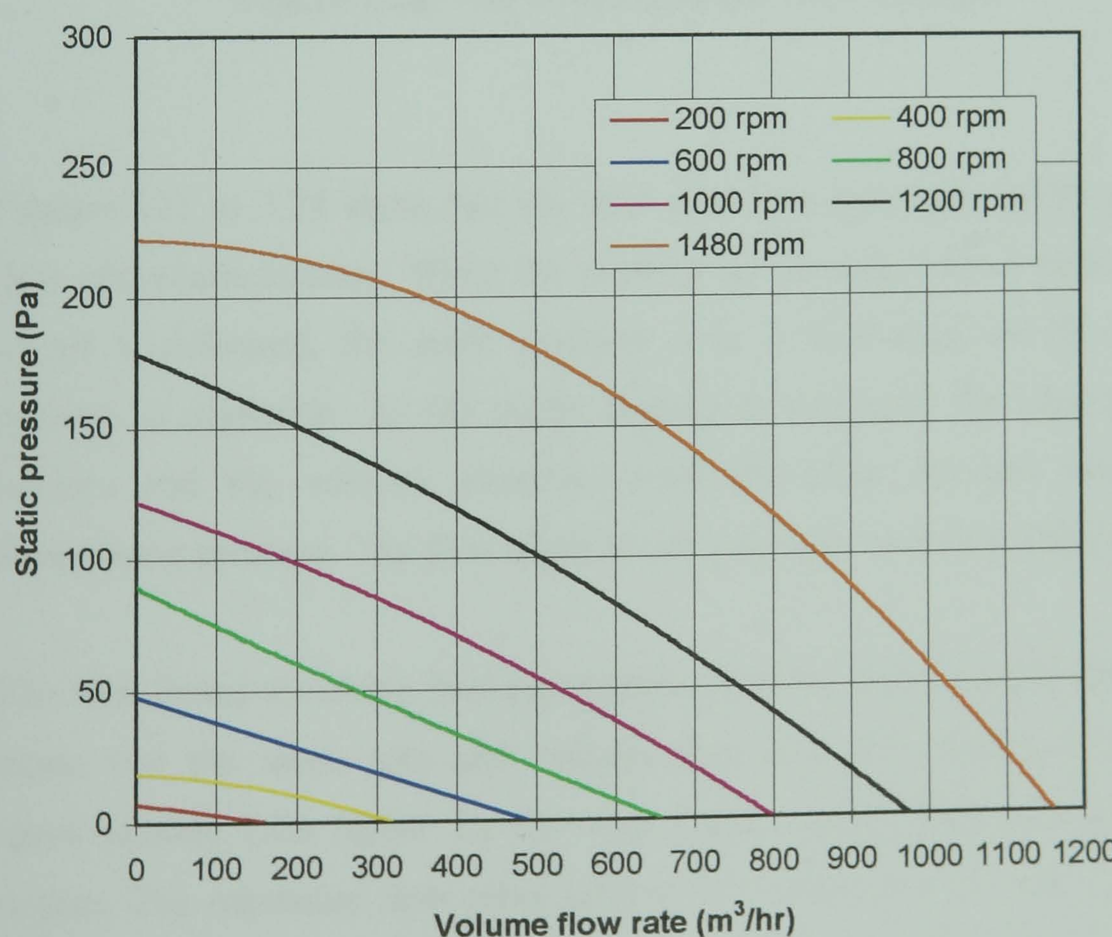


Figure 7.23: Fan characteristic: 24-pipe, wire fin

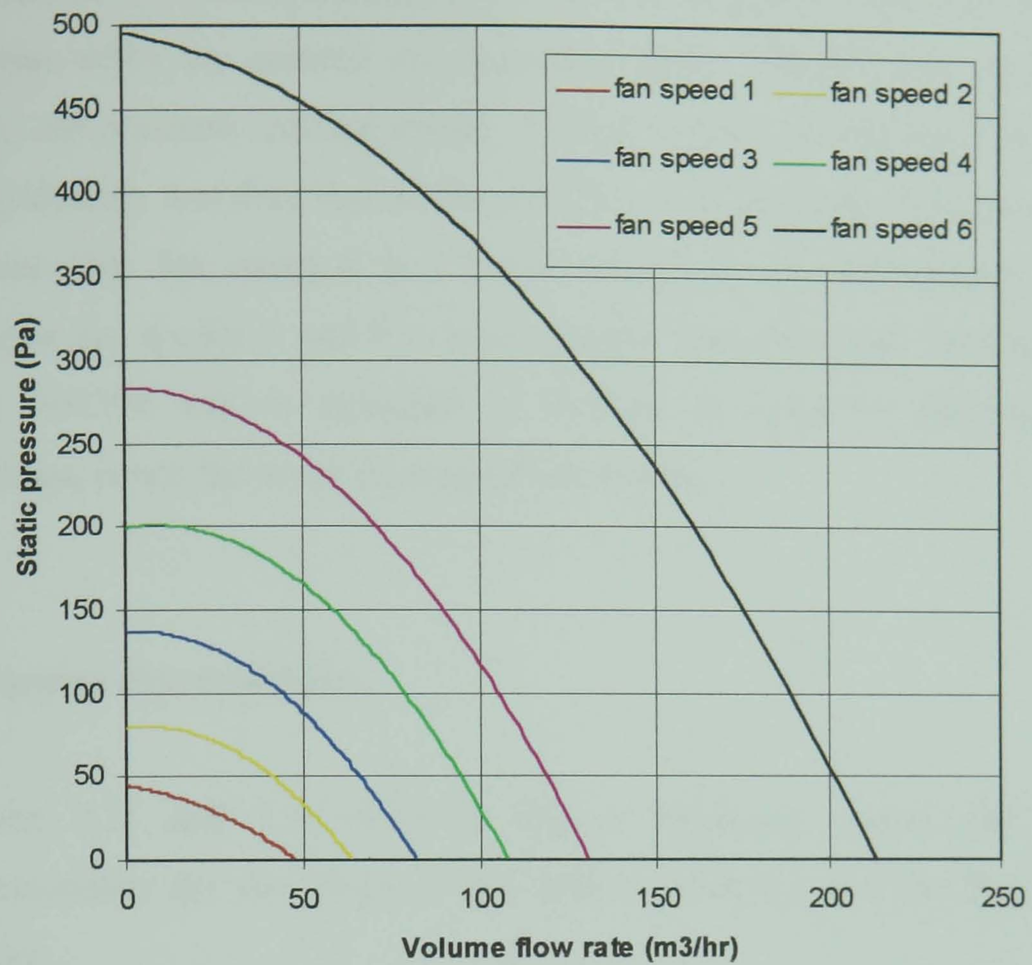


Figure 7.24: Fan characteristic: Baxi WH300

Figures 7.21 to 7.24 show that the four systems tested have similar pressure flow rate characteristics. When the systems are running with a closed outlet, no air is delivered, the static pressure is at a maximum and the velocity pressure is negligible. As the outlet opening is increased, the static pressure reduces and the velocity increases until the static pressure approaches atmospheric pressure. The flow of air at this point has reached a maximum.

The wire finned revolving heat pipes perform better than the plate finned heat pipes. For the same rpm and volume flow rate, the static pressures are approximately 15% higher for the wire finned system than the plate finned system. The maximum flow rates achievable at each rpm are also higher for the wire finned system, again by approximately 15%. The 9-pipe and 24-pipe wire finned systems have similar performances.

The fans on the Baxi system generate more static pressure than the prototypes generate when the systems are operating with a closed outlet. At each fan speed, the pressure reduces sharply with flow rate to a maximum flow which is significantly less than that of the prototypes. From Figure 7.24 it can clearly be seen that fan speed 6 is a boost setting, as the performance increase between fan speeds 5 and 6 is much greater than the other increments. The Baxi WH300 unit is designed to operate in small to mid-range sized dwellings, hence the lower operational flow rates.

7.3 System characteristics

Figures 7.25 and 7.26 illustrate typical efficiency, power and pressure characteristics for the 24-pipe wire finned prototype and the Baxi WH300 system.

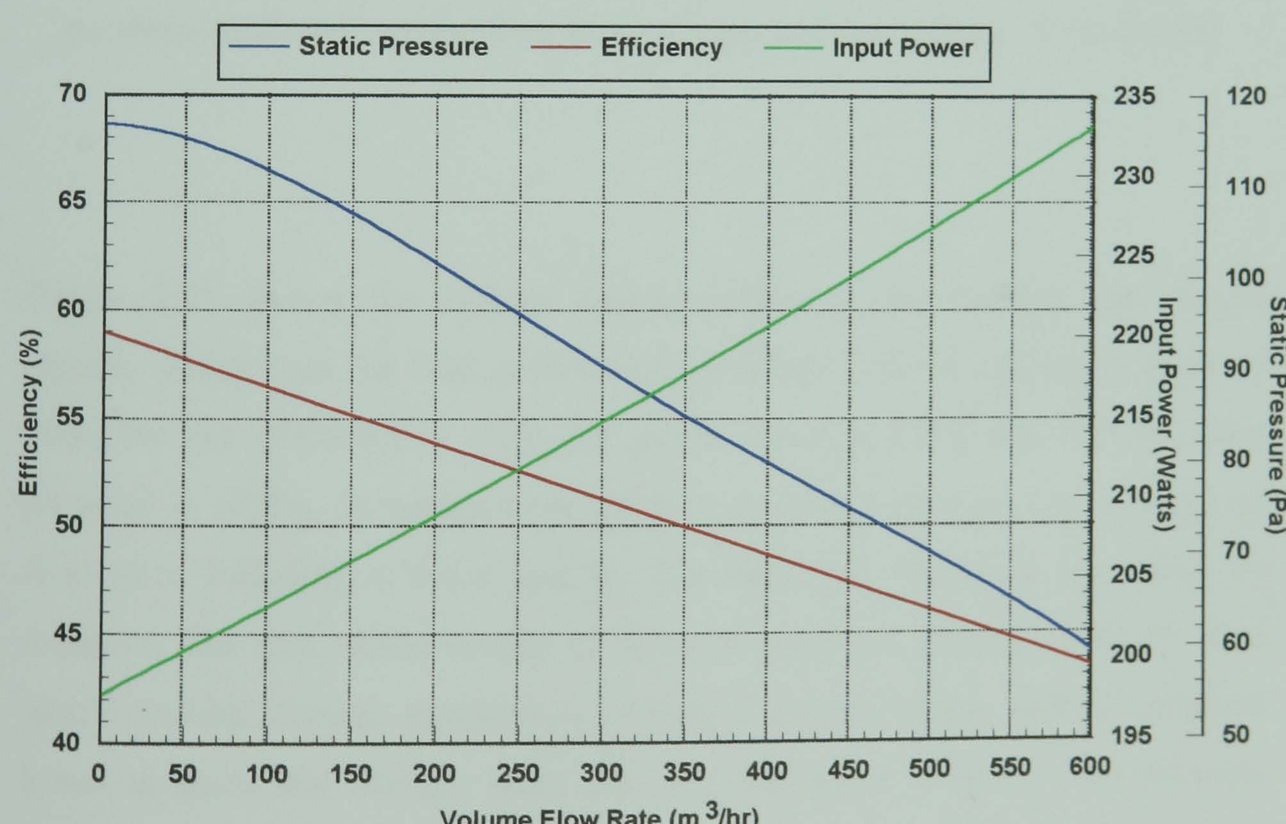


Figure 7.25: System Characteristics – 24-pipe, wire finned system

(Supply and exhaust inlet temperatures are 0°C and 20°C respectively. Efficiency and power are shown for 900rpm and static pressure is shown for 1000rpm).

Figure 7.26: System Characteristics – Baxi WH300 System

(Supply and exhaust inlet temperatures are 0°C and 20°C respectively. Efficiency and power are shown for the full range of fan speeds and static pressure is shown for fan speed 6).

Figure 7.25 shows the system characteristics of the 24-pipe wire finned system, which was the best performing prototype. At the operating point of 200m^3/hr the efficiency is 53%, the power input is 209Watts and the static pressure is 102Pa. In practice the prototypes would operate over a range of flow rates including a boost setting. The flow rate would be controlled by dampers. The rpm would remain constant at a level of around 1000rpm. This would provide enough pressure to overcome system losses, such as pressure losses in ducts and fittings. Also, the heat exchanger efficiency is somewhat dependent upon speed of rotation. The maximum heat exchanger efficiency occurs at higher speeds and, therefore, flow control by control dampers, and not fan speed, is more appropriate.

The performance of the prototype at 200m^3/hr is similar to the performance of the Baxi system shown in Figure 7.26. In Figure 7.26 the lines of efficiency and input power have been extrapolated to show values at flow rates higher than those achieved in testing. By comparing the efficiency characteristics of both systems it is clear to see that the maximum heat exchanger efficiency for the Baxi system is approximately 17% higher than the prototype at very low flow rates.

7.4 Other testing

7.4.1 High temperature heat recovery

A small number of heat recovery tests were undertaken using a propane gas heater in place of the electric duct heater. This enabled much higher exhaust air inlet temperatures to be produced. The testing was carried out on the 9-pipe wire finned system. The flow rate was kept high during testing to avoid damage to the system from overheating. The following values are an example of typical results from the limited number of tests carried out.

Volume flow rate = 400m^3/hr

Exhaust air inlet temperature = 90°C

Supply air inlet and outlet temperatures = 0°C and 33°C respectively.

Heat exchanger efficiency = 37%

The tests were an initial evaluation to establish whether the system could be used for warm air heating, using the heat recovered from high temperature heat sources such as boiler flue gases. The systems lend themselves to recovering heat from polluted air, such as that which might be found at high temperatures, due to their low fouling characteristics. The results show that, although the heat exchanger efficiency is low, the system could be used to provide warm air heating, without cross contamination, from recovered heat from high temperature polluted air.

7.4.2 Airflow through revolving heat pipes

The movement of air through the prototype systems was also examined to determine whether the entire finned lengths of the heat pipes revolving within the fan casings were being utilised. A perspex duct was positioned on the fan casing outlet and a smoke generator was used to supply smoke to the inlet duct. Photographs 7.7 and 7.8 show the duct before the smoke passes through and Photographs 7.9 and 7.10 show the duct with the smoke passing through. It can be seen from the photographs that the smoke fully fills the duct in both planes. There is no evidence of air short cutting through the fan and missing out part of the surfaces of the heat pipes. The smoke patterns through the duct were the same over a range of flow rates and rotational speeds.

Photograph 7.7: Top view – no smoke

Photograph 7.8: Side view – no smoke

Photograph 7.9: Top view – with smoke

Photograph 7.10: Side view – with smoke

7.4.3 Airtightness tests on the fan casings

The integrity of the rotating fan casing seals was checked by performing air leakage tests. Flow rates were measured in the inlet and outlet ducts over a range of flow rates and rotational speeds. For the supply and exhaust sides of the system, the inlet flow was compared to the outlet flow at each flow rate and speed. Over a range of conditions no significant variation in flow rates, before and after the fan casings was recorded. The results of these tests indicated that the rotating brush seals were operating satisfactorily.

Although the brush seals performed satisfactorily, an alternative sealing arrangement should be sought. The brush seals generate noise and create friction which increases the power required to rotate the pipes.

7.5 Economic Analysis

The pay back period for the revolving heat pipe MVHR system is calculated in the following section.

Capital Costs:

For 24-pipe, double row prototype:

Wire finned heat pipes (×24), £64.30 each:	£1543.20
Fan casings (×2), £128.40 each:	£256.80
Brush seals (×2), £8.90 each:	£17.80
Shaft, ∅25mm silver steel:	£37.38
Bearings, ∅25mm pillow block type (×2), £19.39 each:	£38.78
Motor, 1.5kW, 3-phase, 4-pole:	£116.50
Motor slide rails:	£27.30
Drive belt:	£3.91
Pulleys & bushes:	£25.44
Frame:	£10

Feet, anti-vibration mounts (×4), £14.24 each:	£56.96
Heat pipe clamps:	
Single cast ring clamp (×48), £1.60 each:	£76.80
Double cast ring clamp (×24), £1.80 each:	£43.20
M10 bar (3m), £1.53/m:	£4.59
Perspex revolving seals:	£3
Bushes (×5):	£4.50
Ductwork, diffusers & extract grilles:	£100
Additional fixings - nuts & bolts etc:	£5
Three-phase inverter:	£612.75
Three-phase inverter - key pad:	£54
Three-phase inverter - filter:	£154.57
Total (excluding labour)	£3192.48

The three-phase inverter and associated items are used for speed control to enable laboratory testing over a range of rotational speeds. In practice, they are not required and can therefore be deducted from the total capital cost:

Revised Total (excluding labour)	£2371.16
Labour (assume 10% total cost)	**£237.12**
Total Capital Cost of Prototype	**£2608.28**

After discussions with MVHR system manufactures it was assumed that mass production could reduce the total capital cost by a factor of 5:

Total Capital Cost of Commercial System	**£522**

(N.B. installation costs are not included)

Space Heating Costs

The average annual space heating demand in English houses has been estimated as 12000-14000kWh [Shorrock *et al*, 1992]. The cost of gas and electricity at present are typically 1.4p/kWh and 6.5p/kWh respectively.

The cost of 13000kWh of gas heating (assuming the efficiency of a gas boiler is 65%): $= (1.4 \times 13000) / 0.65$

= £280/year

The cost of 13000kWh of electric heating: $= 6.5 \times 13000$

= £845/year

These two heating costs show the cheapest and most expensive ways of heating. Heating costs using other fuel tariffs, such as Economy 7 off-peak electricity, will be somewhere in between.

Running Costs of System

The number of days on which heating is required is 116. This is based on the mean annual heating degree-day total for the 18 UK degree-day regions over 20 years (1976-1995) [CIBSE, 1999].

The power input to the revolving heat pipe MVHR system running at 900rpm and handling 200m^3/hr is 190Watts. The energy consumption is calculated, assuming the heat recovery system runs for the heating period i.e. 116 days/year (2784 hours/year).

Annual energy consumption for MVHR system $= 2784 \times 0.19$

= 529 KWh

Therefore the annual cost of running the system $= 6.5 \times 529$

= £34.38

Pay Back Period

For a well insulated dwelling without an MVHR system installed, it is assumed that 50% of the total heat loss is in the ventilation air. The MVHR system has been shown to be 52% efficient for typical operating conditions. The space heating cost is therefore reduced by 26%.

For a gas heated house:

Space heating saving $= 0.26 \times 280 - 34.38$

$= £38.42/\text{year}$

Therefore the pay back period for the system = 522/38.42

= **13.6 years**

For a house heated using electricity:

Space heating saving $= 0.26 \times 845 - 34.38$

$= £185.32/\text{year}$

Therefore the pay back period for the system = 522/185.32

= **2.8 years**

The pay back period for the system used in a house heated using gas, is fairly long. This is due to the very low cost of gas at present. However, the benefits gained in terms of improved air quality outweigh the long return period. Without the system installed, the building fabric of a house may succumb to mould and rot which will incur repair costs. Therefore, the occupants of the house may benefit from these hidden savings, as well as the health benefits associated with improved air quality. By incorporating the heat recovery system, the size of the heating system could be reduced. Economic savings could be made by reducing the size of boilers and radiators to match the decreased demand. If heat were recovered from the boiler flue gases, the economics would be even more favourable.

If a house is heated using electricity, the pay back period is much shorter due to the higher fuel cost of electricity. The pay back period is well within the expected lifetime of the system (assume 15 years). The occupiers of the house could recoup the capital cost of installing the system in a short period of time whilst experiencing the air quality benefits.

7.6 Conclusions

Five different variations of a prototype MVHR system were tested to establish their heat recovery efficiency and fan performance. A cross flow plate heat exchanger system currently marketed in the UK for use in dwellings was also tested to determine the performance of an established system.

As would be expected, flow rate has the greatest effect on the heat exchanger efficiency. The efficiency decreases with increased flow rate. The power consumption increases with rpm and flow rate. For the prototypes, the heat exchanger efficiencies tended to increase with increased rpm due to the increased air velocity over the finning. The heat pipes must also revolve at a high enough speed to provide enough pressure to overcome system losses. The results indicate that speeds of over 800rpm are therefore ideal, both for fan performance and heat exchanger efficiency.

The power curves for the revolving heat pipe systems are linear. The power is not proportional to the cube of the flow as would be expected for a standard fan. The relationship is probably different because the weight of the revolving heat pipes has more influence than the airflow on the power consumed.

The COPs were largely greater than 1 and, with the 24-pipe system, the COPs were very high reaching up to values of 26 (exhaust air inlet temperature = 30°C, supply air inlet temperature = 0°C, flow rate = 315m^3/hr, rpm = 500). The COPs increased with increased exhaust air inlet temperature, flow rate and decreased with increased rpm. Although a COP greater than 1 indicates a net economical benefit, one should bear in mind that space heating may be provided by cheap fuels such as oil and gas, whereas the MVHR systems are powered by electricity which is inherently expensive.

The wire finned heat pipes performed better than the plate finned heat pipes, both in terms of fan performance and heat exchanger efficiency. Tapering the heat pipe bundle marginally increased the efficiency however, in practice, the more complicated fixing arrangement makes it less viable. The tapered heat pipe bundle was affected by rotational speed more than the non-tapered bundle. At high rpms the tapered system performed worse than the non-tapered system. The 10% filled heat pipes performed slightly better than the 20% filled heat pipes. However, further testing using a wider range of working fluid charges, would be required to establish the optimum working fluid charge. Further work should also be undertaken to establish the effect of the internal structure of the heat pipes on the efficiency. Tests using different types of working fluids should also be undertaken. Where the systems are to operate in extreme cold climates, a working fluid with a lower boiling point than water would be more suitable. This would avoid freeze-up of the working fluid in conditions where the exhaust air could not provide enough heat to prevent the condensers from freezing.

The use of more heat pipes in a double row greatly increased the heat exchanger efficiency. The 24-pipe double row system had 15% more surface area than the 9-pipe system, which resulted in a similar increase in heat exchanger efficiency. The 24-pipe system compared favourably with the Baxi system at higher flow rates. An economic analysis of the 24-pipe system showed space heating savings of £38/year for a gas heated house and £185/year for an electrically heated house. In both cases the pay back periods are relatively short.

Further work is, however, required to reduce the energy consumption of the prototypes. Energy use could be lowered by reducing the size and weight of components. Freer running seals would also reduce energy use. The brush seals should be replaced by a configuration that has less friction and generates less noise. Many of these problems have been addressed in the latest prototype system shown in Appendix C.

8. MVHR HEAT PUMP SYSTEM

This chapter covers the design and construction of a prototype MVHR heat pump system. The prototype is a further development of the revolving heat pipe systems covered in the previous two chapters. Revolving pipes are once again used to impel air and exchange heat. The revolving pipes form the evaporator and condenser in a heat pump cycle, to provide heating or cooling. The system uses an alternative refrigerant with zero ozone depletion potential and very low global warming potential.

8.1 Theoretical analysis of refrigerants

Theoretical COPs were calculated for a number of different refrigerants to determine the most suitable choice for the operating conditions. The heat pump was designed to provide 4kW of supplementary space heating. For the theoretical calculations, the temperature of the evaporator and condenser are assumed to be 0°C and 40°C respectively. Table 8.1 contains the results of the theoretical analysis.

Refrigerant	P_{evap} (bar$_a$)	P_{cond} (bar$_a$)	W_{comp} (Watts)	Suction Volume (m3/hr)	COP_H
Methanol	0.04	0.353	687.8	201.03	5.81
R134a	2.928	10.166	644.8	6.19	6.19
R32	8.13	24.82	643.4	2.29	6.22
CARE 10	1.57	5.23	682.4	11	5.87
CARE 30	2.78	8.61	520	6.21	7.69
CARE 40	4.72	13.79	633.2	4.74	6.32
CARE 50	5.36	15.14	611.3	3.92	6.54

Table 8.1: Refrigerant Analysis

The following calculations show the procedure used to find the values shown in Table 8.1.

Refrigerant – CARE 10 (Isobutane, R600a)

Temperature of the evaporator = 0°C

Temperature of the condenser = 40°C

4kW rejected at the condenser.

(p-h chart is shown in Appendix F)

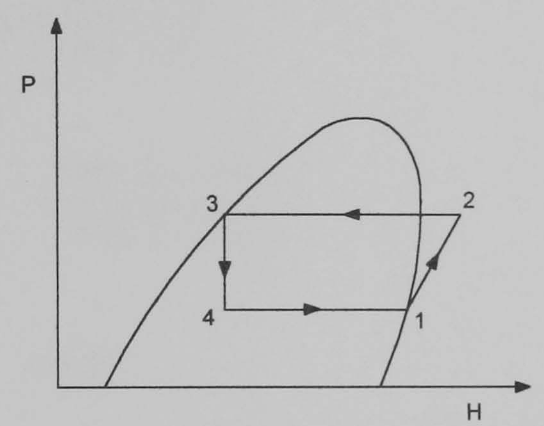


Figure 8.1: Vapour Compression Cycle

$H_1 = H_g$ at 0°C = 552.3kJ/kg

H_2 = 605.2kJ/kg (assuming isentropic compression)

$H_3 = H_4 = H_f$ at 40°C = 294.7kJ/kg

Calculation of suction volume flow rate:

For 4kW heating:

Mass flow rate = 4 / (H_2-H_3)

= 4 / (605.2-294.7)

= 0.0129kg/s

At 0°C the vapour specific volume = 0.2362m^3/kg

Therefore the suction volume flow rate = 0.2362 × 0.0129 (m^3/s)

= **11m^3/hr**

Calculation of compressor work:

W_{comp} = mass flow rate × (H_2-H_1)

= 0.0129 × (605.2 – 552.3)

= **682.4Watts**

Calculation of COP:

$$COP_H = \frac{H_2 - H_3}{H_2 - H_1}$$

$$= \frac{605.2 - 294.7}{605.2 - 552.3}$$

$$= \mathbf{5.87}$$

From the calculations it can be seen that the theoretical COPs are reasonably high in all cases. In practice, the COPs will not be as high due to system losses and compressor inefficiency.

The refrigerant chosen for the MVHR heat pump prototype was isobutane. Isobutane has a good theoretical COP and relatively low evaporator and condenser pressures. Because the prototype's evaporator and condenser will revolve, the pipe work is subject to stresses resulting from centrifugal force. It is therefore important to have low pressures in the pipes so that the overall stresses on the pipe work is minimised. Also, because the system has revolving heat exchangers, rotary seals are required. Again, the vapour pressure working on the seals should be low to avoid stressing them. The system could be designed to handle higher pressures, but it would necessitate stronger pipe work and rotary seals which would be larger, more expensive and have greater turning resistance.

Methanol was disregarded due to its high suction volume which would require a very large compressor. The evaporator and condenser pressures are sub atmospheric which necessitates extremely good sealing to avoid air/moisture ingress. All the other refrigerants screened were not chosen due to the resulting high condenser pressures. Theoretical COPs for isobutane over a range of condenser temperatures are shown in Table 8.2.

T_{cond} (°C)	P_{cond} (bar)	Enthalpy (kJ/kg)			COP
		H_1	$H_{3,4}$	H_2	
30	4.04	552.3	269.9	592.3	8.06
40	5.23	552.3	294.7	605.2	5.87
50	6.71	552.3	320.0	616.3	4.63
60	8.67	552.3	348.1	633.1	3.53
70	10.66	552.3	373.3	641.5	3.01
80	13.40	552.3	403.1	653.8	2.47

Table 8.2: COPs for CARE 10 With Varying Condenser Temperature (T_E = 0°C)

From Table 8.2 it is clear to see that as the condenser temperature increases the COP decreases. As the condenser temperature increases, the work required by the compressor increases to overcome the increased pressure differential between the evaporator and the condenser.

8.2 Compressor specification

One of the features of the compressor used in the vapour compression heat pump cycle of the prototype is that it is belt driven. This is so that it can be driven from the same motor that turns the revolving pipes. From the theoretical analysis the compressor should have a suction volume flow of $11m^3/hr$.

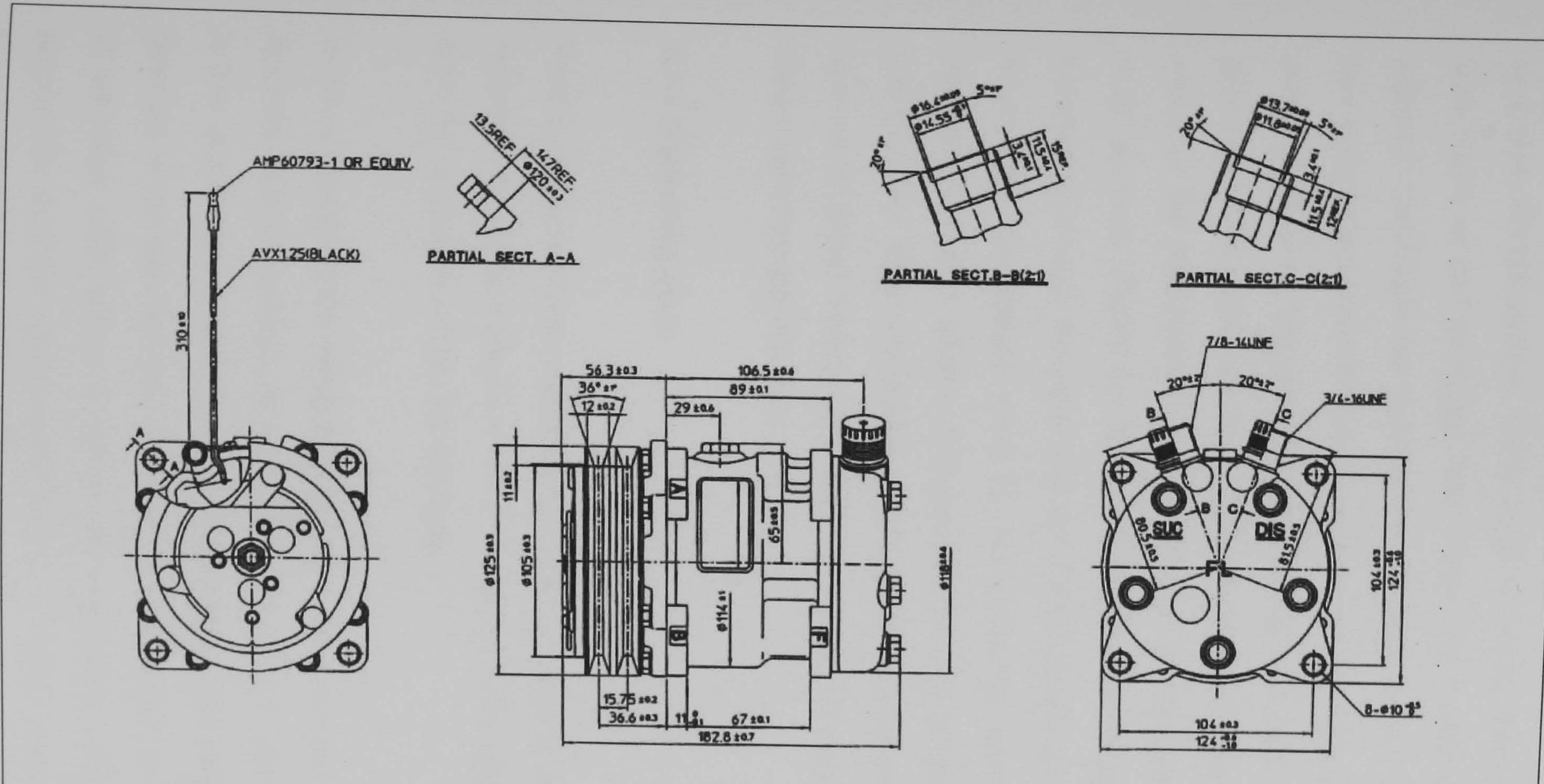
Compressor manufacturers around the world were contacted to supply a compressor to suit the application. Of those manufactures who produced isobutane compressors, the majority only manufactured them for the domestic refrigerator market. These compressors are for very small duties (100-200W). Only one supplier (Embraco in Brazil) produced an isobutane compressor which was close enough to the required specification, and this compressor was hermetic and not belt driven. This was a relatively new product in their range and they were unable to supply just one compressor. Their minimum order was 500 units which was non negotiable. Although isobutane is widely used in domestic refrigerators, at present it is virtually unused in domestic heat pump applications.

Unable to obtain an isobutane compressor other options were sought. The compressor chosen for the application is a R134a vehicle refrigeration compressor. The unit chosen is the smallest reciprocating compressor in Sanden International's range. The compressor specification is shown in Figure 8.2. The unit is belt driven with a clutch mechanism which operates from a 12 Volt supply. The clutch mechanism enables the compressor to be connected to a pressure cut-out switch. The switch will control the compressor only, leaving the motor free to revolve the pipes if the cut-out is activated.

From the compressor specification the displacement per revolution is 87cc/rev. The closest pulley ratio was specified to obtain an appropriate compressor speed giving the required suction volume flow rate. When driven by a 4-pole motor at maximum speed (1480rpm) the compressor runs at 2175rpm, giving the following compressor displacement.

$$\text{Compressor displacement} = 2175 \times 87$$
$$= 11.35\text{m}^3/\text{hr}$$

This value satisfies the theoretical suction volume flow rate of $11\text{m}^3/\text{hr}$.



COMPRESSOR SPECIFICATION

Bore	35mm (1.38ins)
Stroke	16.07mm (0.71ns)
Displacement per revolution	87cc/rev (5.3cu.in/rev)
Number of cylinders	5
Maximum allowable rpm	7500rpm
Maximum allowable continuous rpm	6500rpm
Refrigerant	R-134a
Oil (SP-20 or Equiv)	100cc (3.4fl.oz)

CLUTCH SPECIFICATION

Rated voltage	DC 12V
Breakaway torque	2.2 kg-m (15.9ft-lb)
Minimum engagement voltage	7.5V max
Power consumption	48W (AL)
Maximum allowable rpm	7500rpm
Maximum allowable continuous rpm	6500rpm
Belt type	A Type
Pulley diameter	125mm (4.9ins)

COMPRESSOR

SANDEN CORPORATION

Model: SD5H09FL5072

Figure 8.2: Compressor Specification [Sanden International]

8.3 MVHR heat pump design

The MVHR heat pump prototype is based on the 9-pipe wire fin revolving heat pipe MVHR system. Once again revolving wire finned pipes are used to both impel air and exchange heat. There are 9 wire finned heat pipes in two adjacent centrifugal fan casings. The 9 pipes in one fan casing comprise the heat pump evaporator and the 9 pipes in the other fan casing comprise the heat pump condenser. The same size fan casings are used as before and the pipe geometry is the same (see Figure 6.8). Because the geometry and sizes are the same, the fan performance in terms of static pressure and flow rate is assumed to be the same. Figure 8.3 shows a schematic of the system. Separate motors are used to revolve the pipes and drive the compressor. This makes it possible to measure the energy used by the revolving pipes and the compressor separately. It also allows independent control of speed without changing pulley ratios. This will enable optimum speeds for the compressor and the pipes to be found. Pulleys can then be specified for each speed so that a single motor can eventually be used.

8.3.1 Operating cycle

Heat is recovered from stale warm air passing over the evaporator. This heat is upgraded by work input from the compressor. The upgraded heat is rejected from the condenser to the cold fresh air.

In the evaporator the refrigerant forms a layer on the inside walls, furthest from the axis of rotation, of each of the nine pipes. The vapour pressure here is low due to the suction from the compressor. The refrigerant picks up heat from the warm stale air and it evaporates. The vapour in each pipe then travels to the pipe ends where it passes through tubes which provide structural support for the heat pipes as well as being vapour passageways. The vapour from each of the nine pipes combines and travels along a hollow section of the main shaft. At the end of the shaft the vapour passes through a rotary seal and into the suction line of the compressor. The vapour then passes through a filter drier on the way to the compressor inlet. Oil from the oil separator is mixed with the vapour before it enters the compressor, where it is compressed and exits as a compressed vapour/oil mixture. The oil is removed in the oil separator and the compressed vapour travels along the discharge line towards the revolving condenser pipes.

The vapour now passes through another rotary seal and along a hollow section of the main shaft, from which it travels into each of the condenser pipes via the spoke/tubes. It then condenses in each pipe giving out heat to the cold inlet air. Again, due to the centrifugal force, the liquid refrigerant forms a layer on the inside pipe walls furthest from the axis of rotation. The condenser pipes do not revolve parallel to the axis of rotation. They are slightly angled so that the radius of revolution at the vapour entry end is smaller than the radius of revolution at the liquid exit end. The angled pipe wall exerts a component of force, along the pipe length, on the liquid refrigerant propelling it to the exit. The exit points on the condenser pipes and the inlet points on the evaporator pipes are positioned at the same distance from the axis of revolution, so that the centrifugal forces at these points are equal.

From each of the condenser pipes the refrigerant enters a circular equalising header. The circular condenser header links the nine pipes together, thus ensuring equal quantities of refrigerant are in each pipe. The header is a ring of pipe with a ring diameter greater than the diameter of the condenser/evaporator pipe bundles. This ensures the circular header will remain full of liquid at all times during operation. From the condenser header the liquid refrigerant passes through three expansion valves. The valves are positioned equally around the circumference of the circular headers. From the valves, the expanded refrigerant enters the evaporator header. To allow room for expansion and flash gas, the diameter of pipe used for this header is slightly larger than the pipe used in the condenser header. The evaporator header then feeds equal quantities of refrigerant into the evaporator pipes, where the cycle repeats.

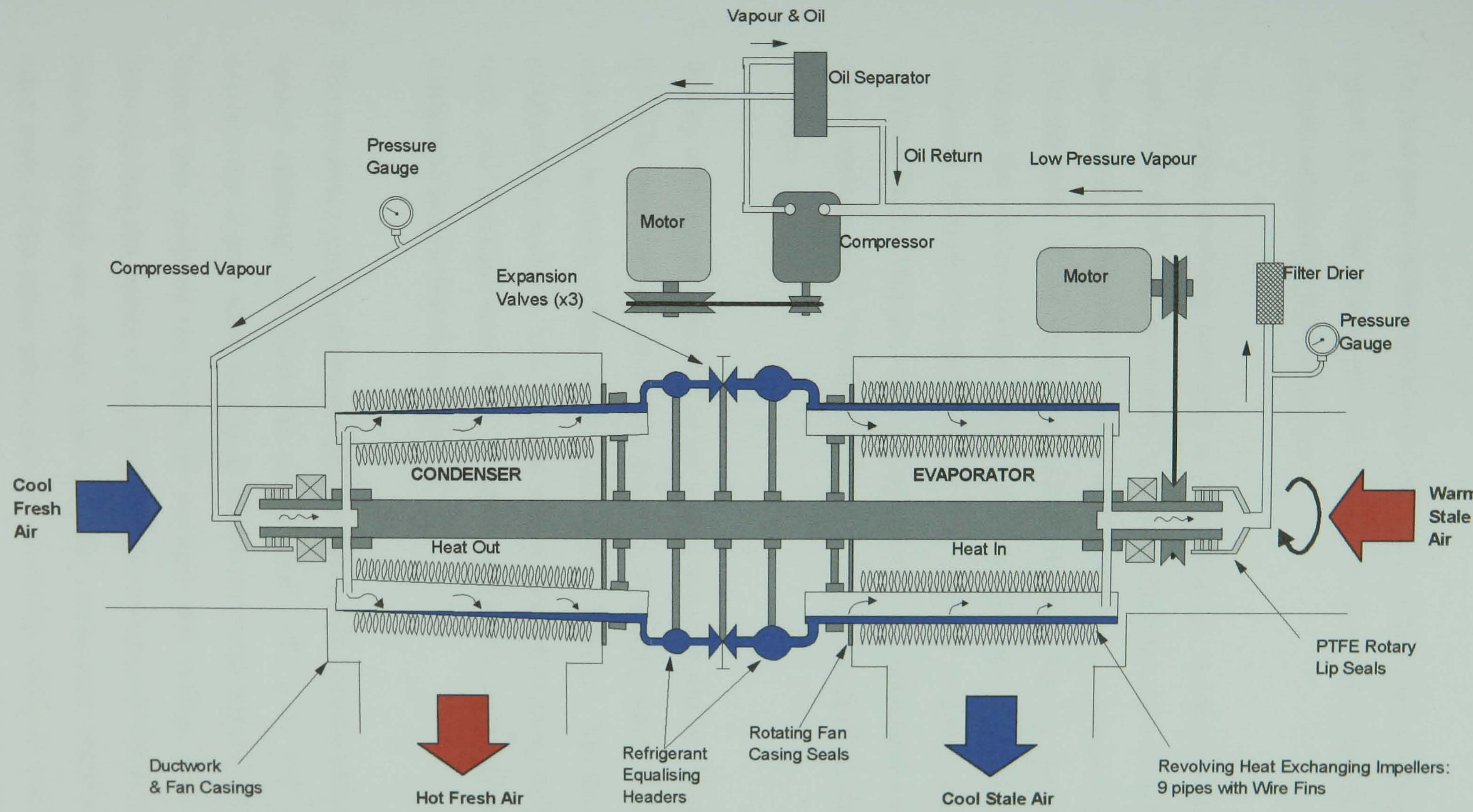


Figure 8.3: MVHR Heat Pump Schematic

8.3.2 Component specification

1. The heat pump system is mounted on the steel box-section frame shown in Figure 8.4. The frame sits on general purpose anti-vibration machine mounts supplied by RS Components.

2. The entire revolving assembly is held in place around the centre main shaft. The shaft was fabricated to the specification shown in Figure 8.5. The shaft ends were hollowed out and formed to the specification shown. The diameter of the shaft upon which the seals revolve was reduced to reduce the circumference which, in turn, reduces the seal length and rotational friction. The shaft surface was polished to a very smooth finish ($\leq 0.2\mu m$ Ra for vacuum sealing applications) over the length on which the rotary seals revolve. The shaft rotates in two 30mm shaft ∅ pillow block bearings.

3. Rotary seals were fabricated to the specification shown in Figures 8.6 and 8.7. The seals consist of a brass housing containing three PTFE lip seals and two bearings (17mm shaft ∅ and 20mm shaft ∅). The lip seals were produced by Seal Co. Ltd. to hold vacuum and pressure up to 15bar. The seals and bearings were held in place within the housing by retaining compound and circular brass spacers.

4. The revolving pipes, valves and headers were held in place with clamps on spokes radiating out from the centre shaft. 3/8" o.d., 1/4" i.d. tubes connect the centre shaft and each of the revolving pipes. The tubes are brazed into each of the pipes for strength and sealing. The tube ends protrude into the centre of each pipe. This prevents any liquid, at start-up, passing through the rotary seals into the compressor suction line. The other ends of the tubes are brazed into manifolds which sit over the radial holes in the centre shaft. The manifolds are sealed with 216 type O-rings. Both manifold specifications are shown in Figure 8.8.

5. The specification of the revolving wire finned pipes (9 in each fan casing) is shown in Figure 8.9. All the pipes had glass ends so that the refrigerant film could be observed, using a stroboscope, during operation. The pipes in the evaporator have wire mesh wicks to spread the refrigerant over the inner pipe surface and enhance heat transfer. The evaporator pipes revolve parallel to the main shaft. The axes of the evaporator pipes are 100mm from the axis of revolution. The condenser pipes are angled at 2.7° from the horizontal with the pipe centres at the ends closest to the evaporator pipes positioned 100mm from the centre of revolution.

6. The evaporator pipes are interconnected by the evaporator header which is a circular ring of 5/8" copper tube. The ring radius is 126mm. The evaporator pipes, header and valves are connected with 3/8" copper tube. The evaporator header and 3/8" tubes are insulated. The condenser pipes are interconnected by the condenser header which has a ring radius of 126mm and is fabricated from 1/2" copper tube. The condenser pipes, header and valves are connected with 1/4" copper tube. Three needle valves with vernier scale adjustment are used for vapour expansion.

7. Copper tubing is used for the suction and discharge lines and the diameters are 5/8" and 3/8" respectively. Flexible rubber hose is used between the rotary seals and the copper lines to absorb vibration movement. Site glasses are present in both the suction and discharge lines. A pressure gauge (-1 to 8bar) and filter drier are located in the suction line. A helical oil separator is positioned in the discharge line. Oil is fed into the supply line through 1/4" copper tubing and the rate of oil return is controlled by a valve. The filter reservoir is filled with high viscosity (ISO 100) PolyalkyleneGlycol (PAG) compressor oil. A 0bar to 30bar pressure gauge is located in the discharge line. The refrigerant lines and the oil separator are insulated with flexible synthetic insulation.

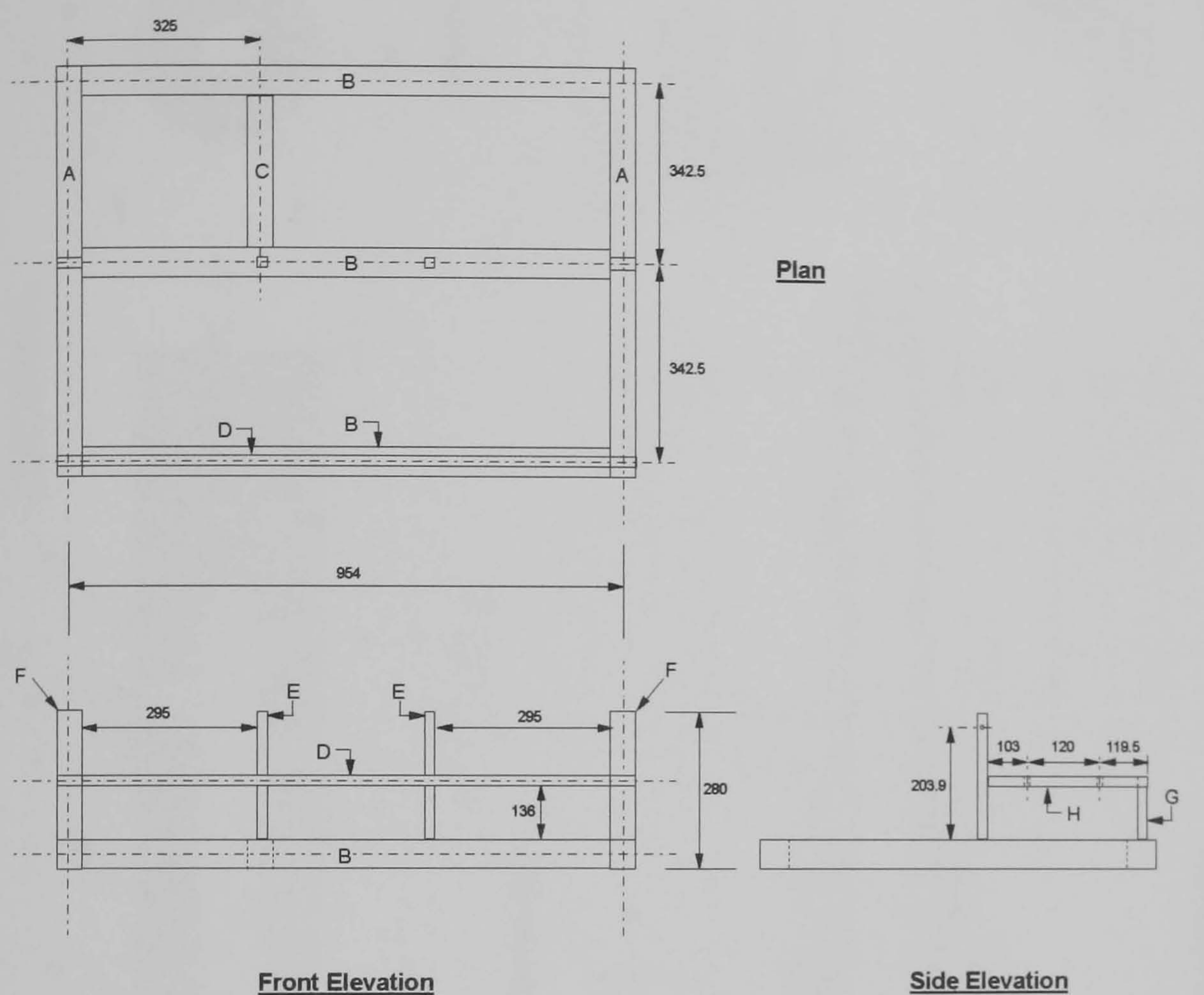
8. The compressor (Figure 8.2) is driven by a 1.5kW, 3-phase, 4-pole, foot mounted motor, supplied by Fenner Power Transmission, catalogue

number H1408. The motor sits on slide rails and has an SPA-100-1 pulley with a 1610 taper lock bush driving the motor with an A-type wedge belt. The revolving pipes are driven by another 1.5kW motor of the same specification.

9. A 12 Volt supply is used to power the compressor clutch mechanism. A pressure switch is connected to the clutch. If the pressure in the discharge line reaches 10 bar the switch will disengage the clutch.

10. The fan casings are the same as those used for the MVHR revolving heat pipe system. The same rotating brush seal arrangement is used. The rotating assembly between the two fan casings is covered with a steel mesh guard. A drain at the base of the evaporator fan casing was used to drain the condensed air moisture into a container below.

Photograph 8.1 shows the MVHR prototype heat pump rig with insulated refrigerant lines and ductwork attached to the fan casings. Photograph 8.2 illustrates the prototype components. The condenser and evaporator headers are shown in Photograph 8.3. Photographs 8.4 and 8.5 show the rotary shaft seals and Photograph 8.6 shows a side view of the prototype.



Component List:

section	Ref.	Length & Quantity
50x50x3	A	717.5 (x2)
	B	1004 (x3)
	C	275 (x1)
25x25x3	D	904 (x1)
	E	230 (x2)
50x25x3	F	230 (x2)
	G	136 (x2)
	H	342.5 (x2)

Notes:

Not drawn to scale.
All dimensions in mm, unless otherwise stated.
Steel box section cut to above specification.

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Figure 8.4: MVHR Heat Pump Frame

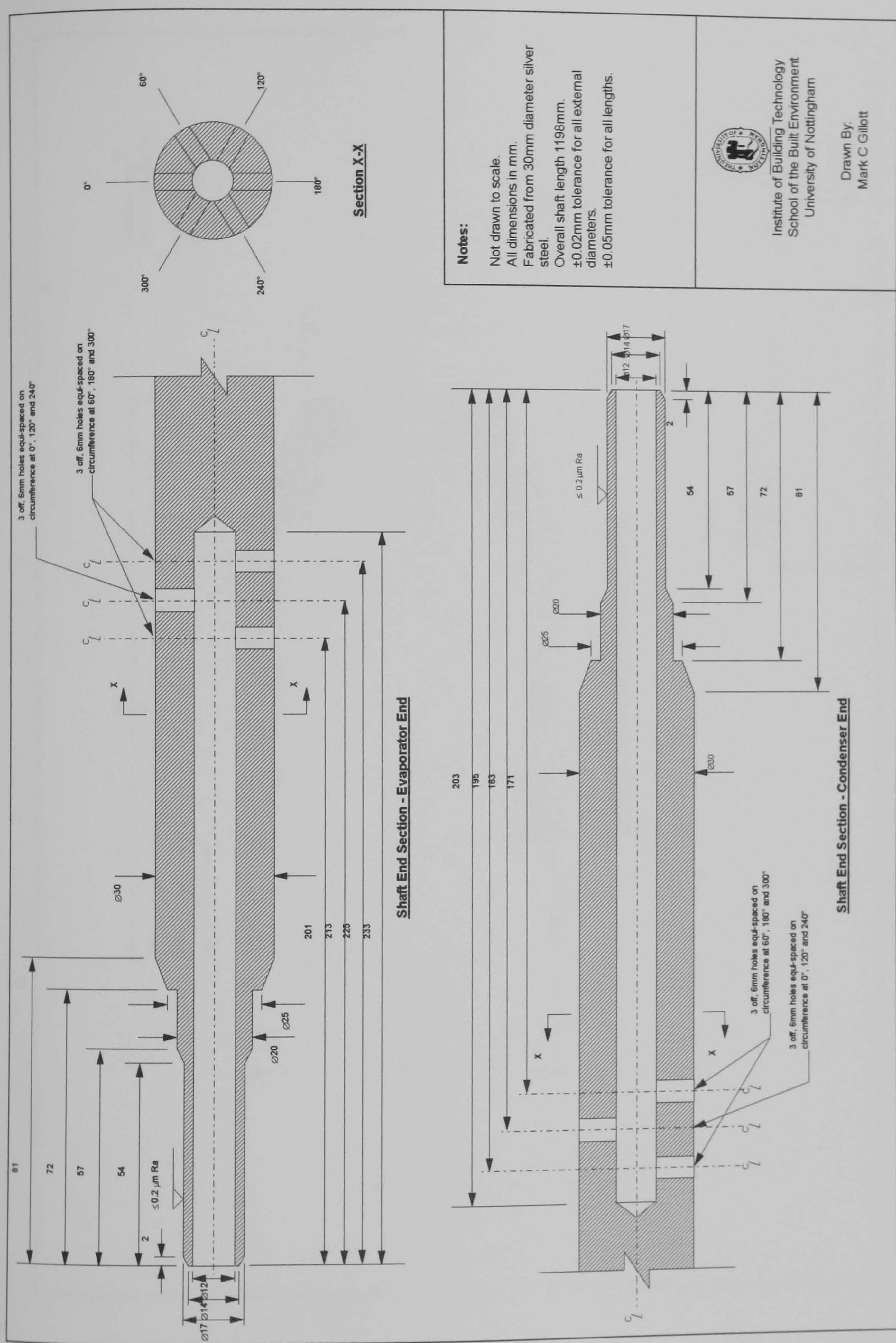


Figure 8.5: MVHR Heat Pump Main Shaft

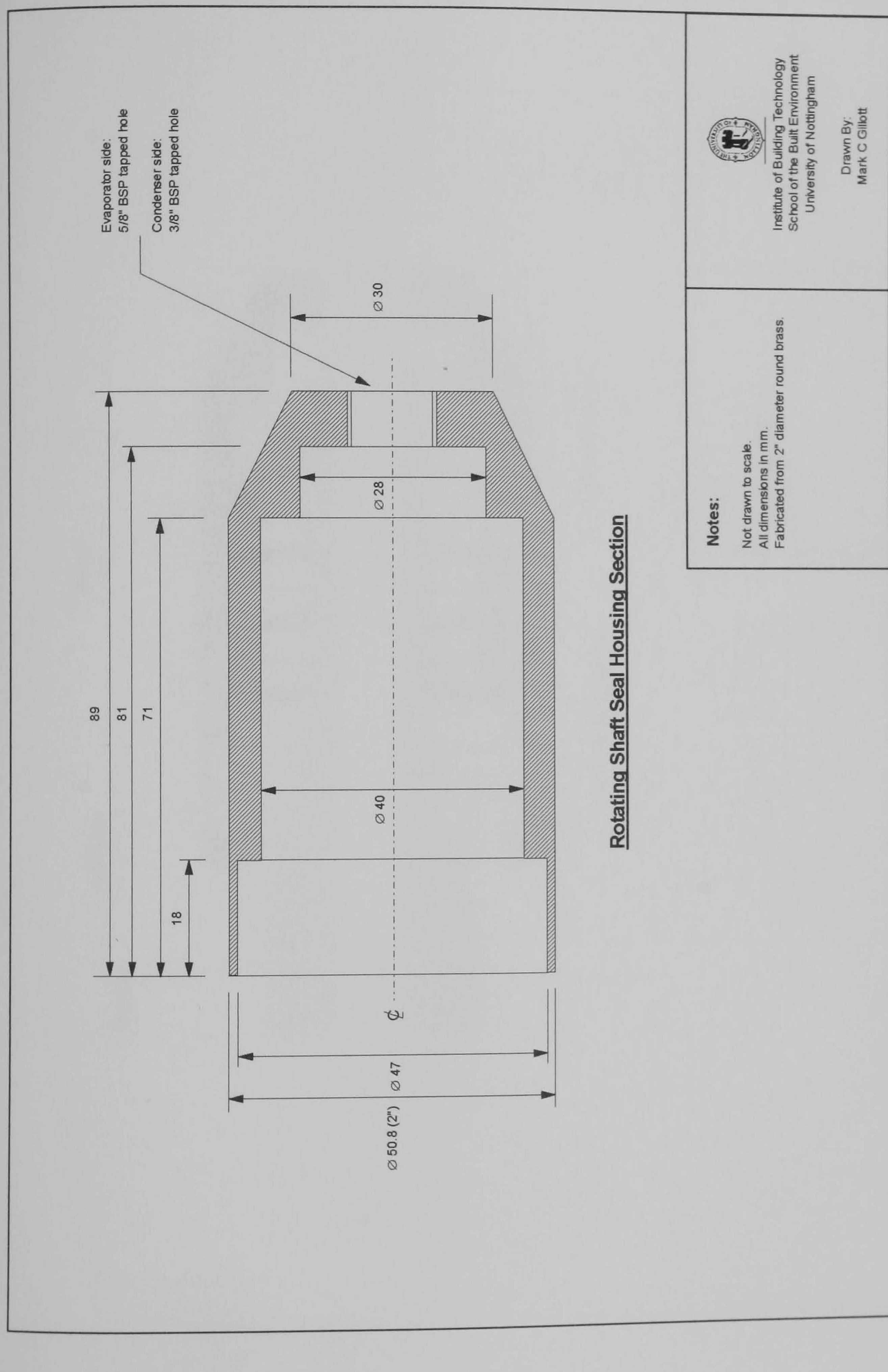


Figure 8.6: Rotary Shaft Seal Housing

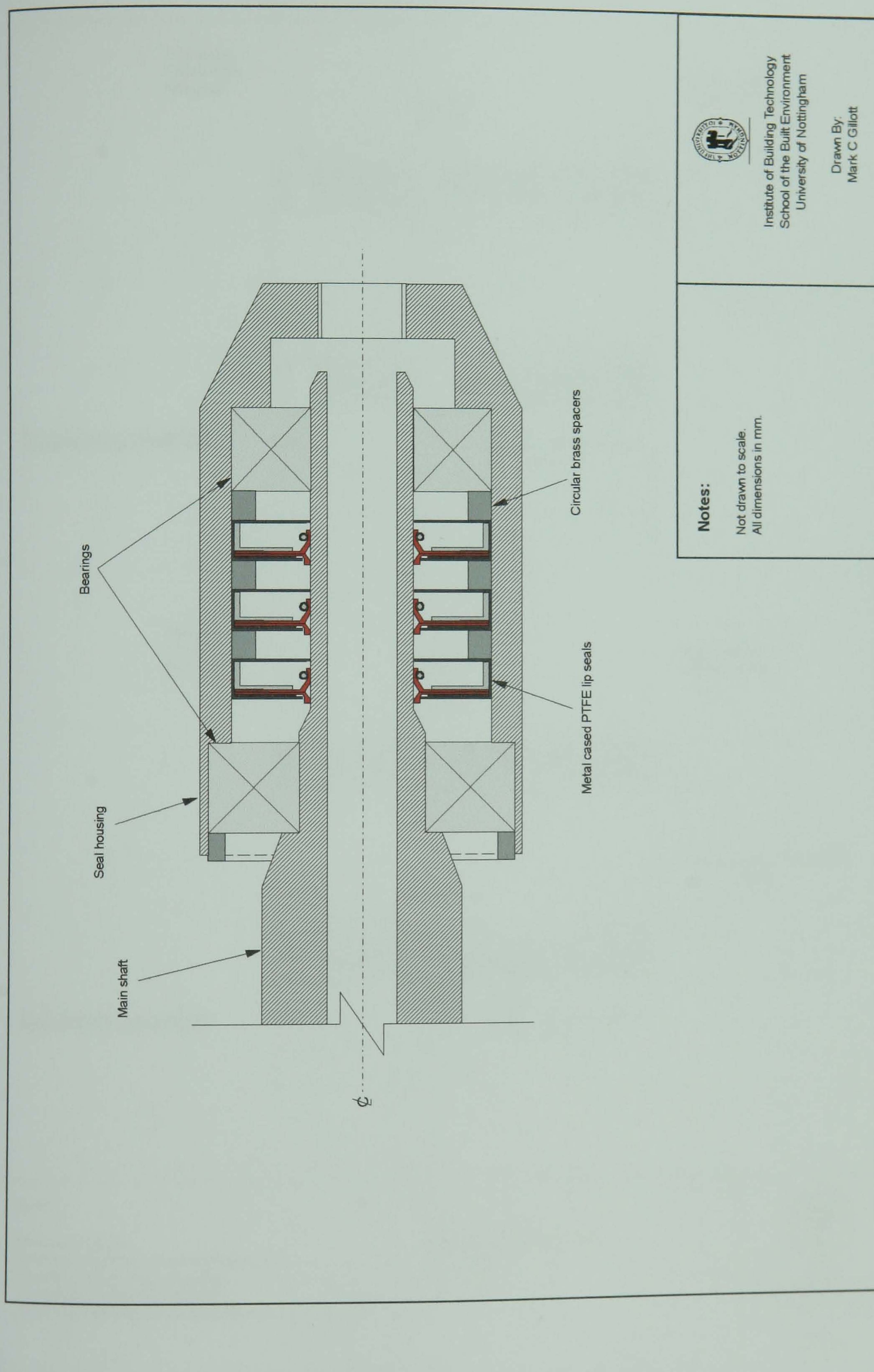


Figure 8.7: Rotary Shaft Seals

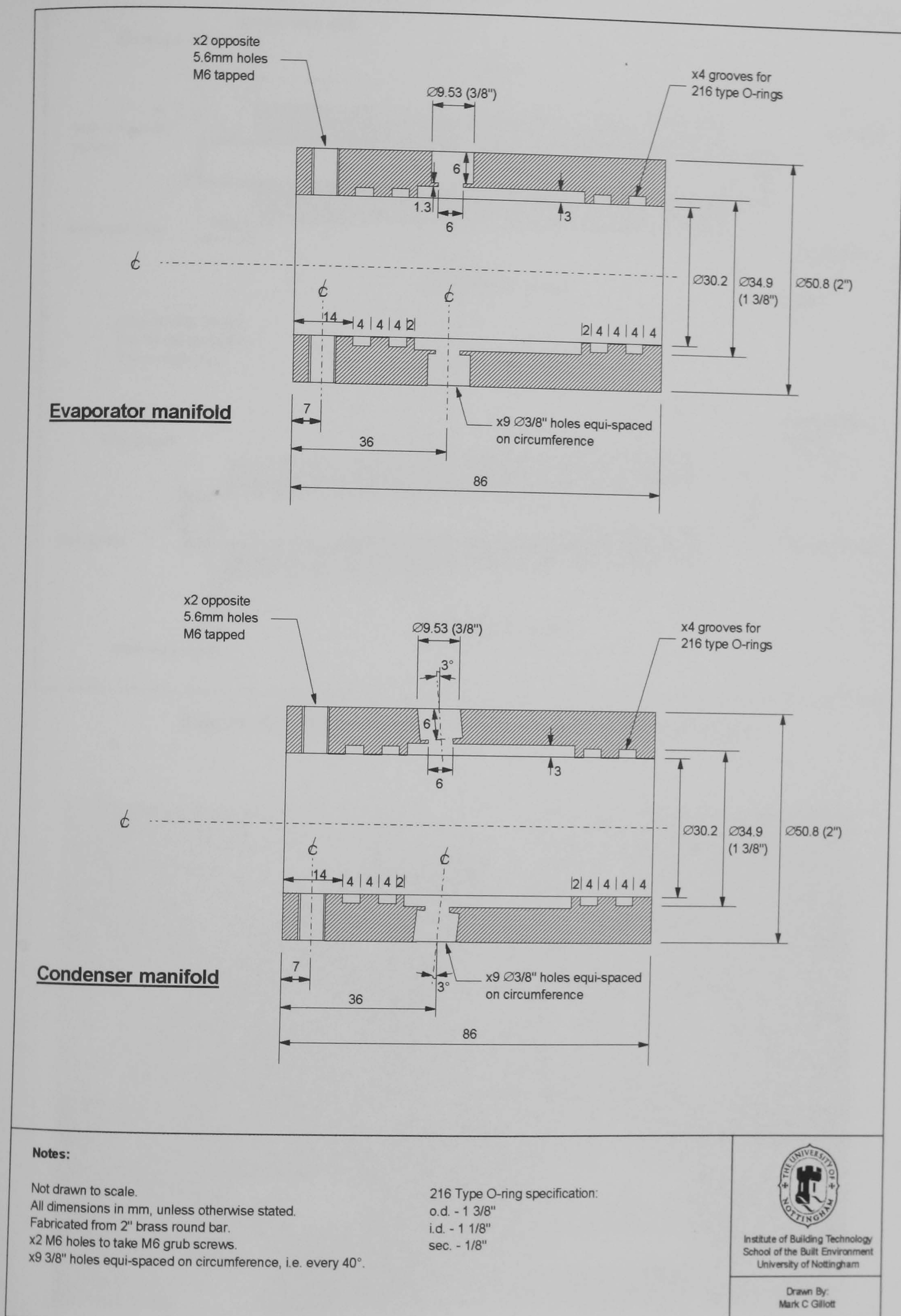


Figure 8.8: Evaporator and Condenser Manifolds

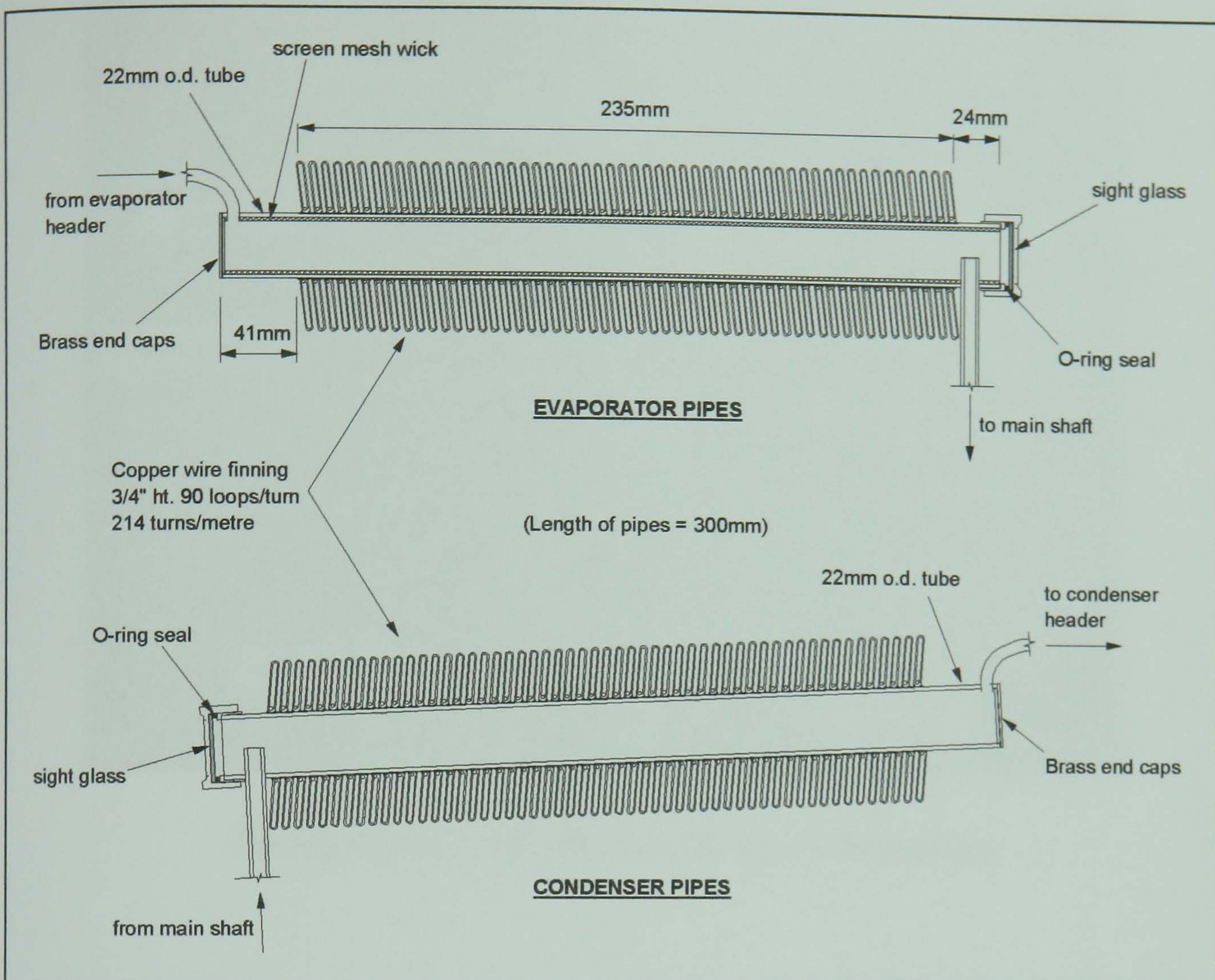
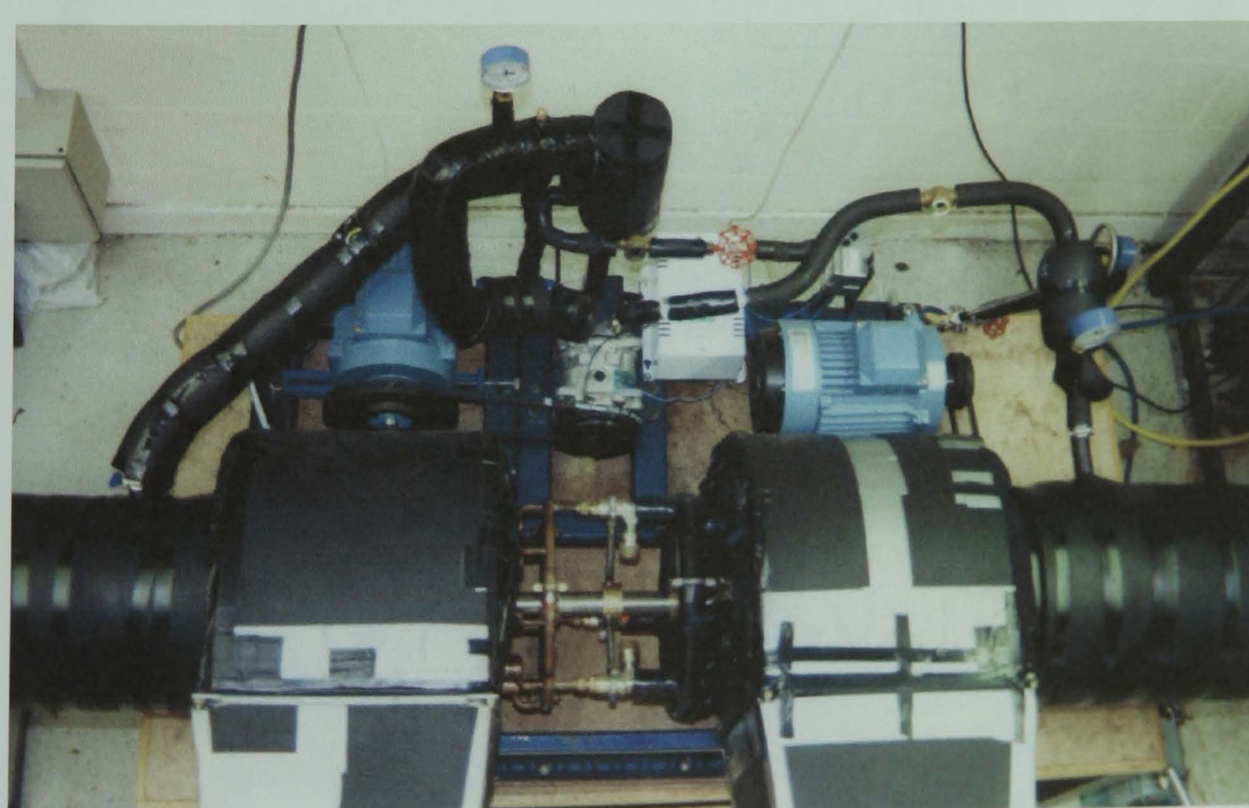
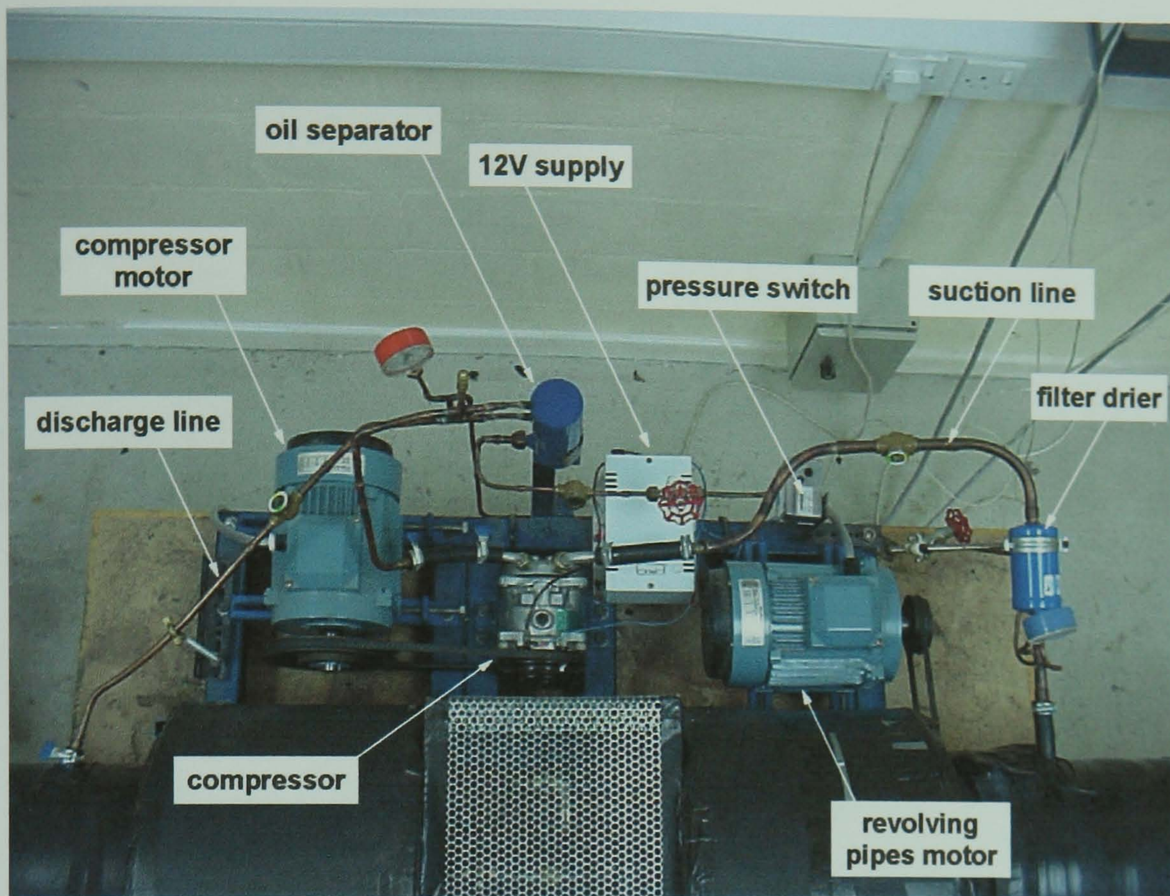


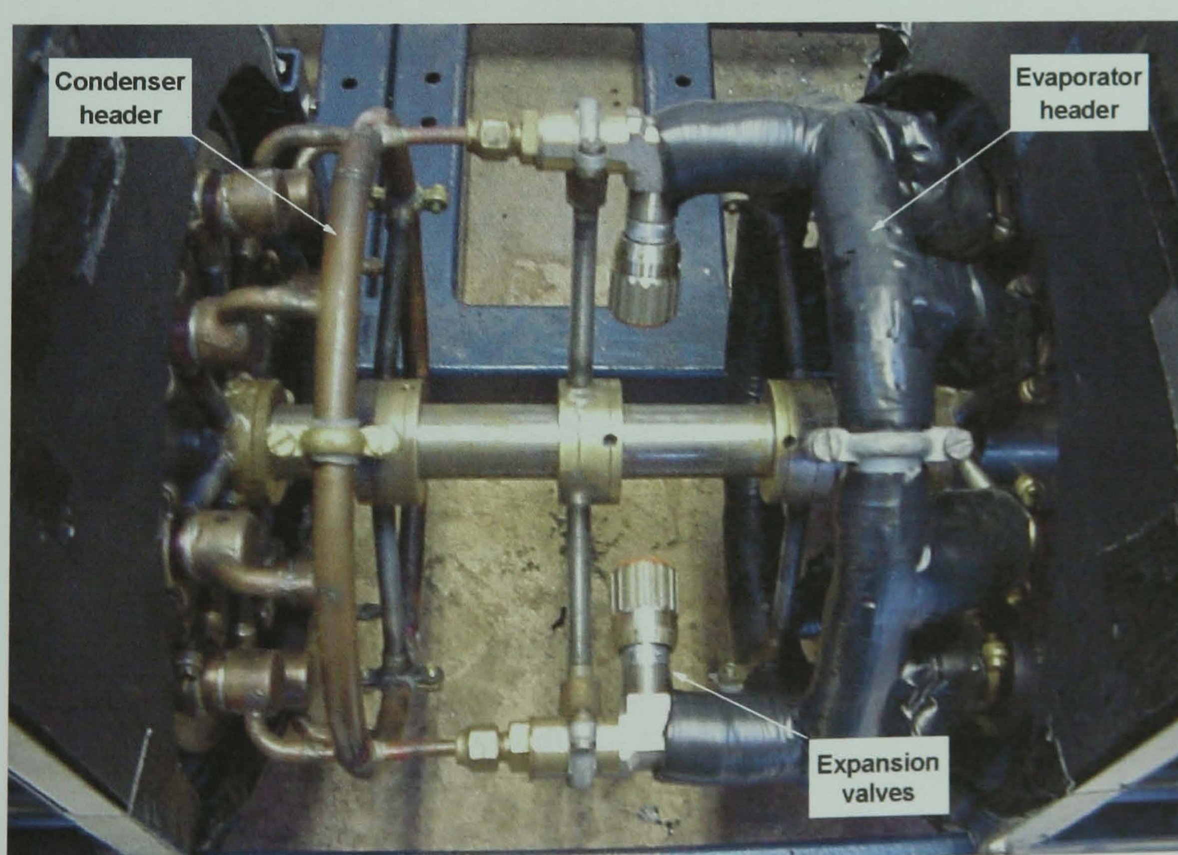
Figure 8.9: Wire Finned Evaporator/Condenser Pipes



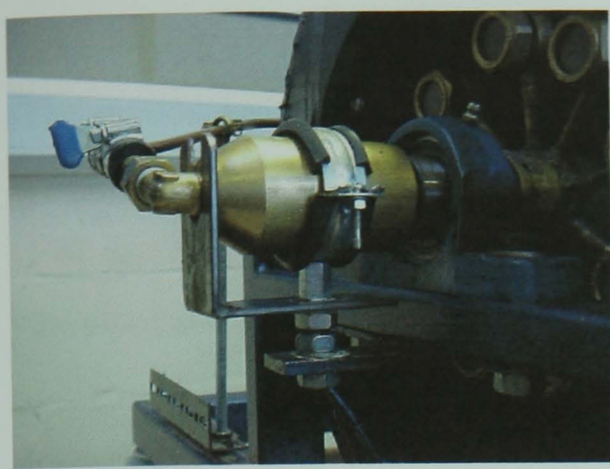
Photograph 8.1: MVHR Heat Pump Prototype Rig

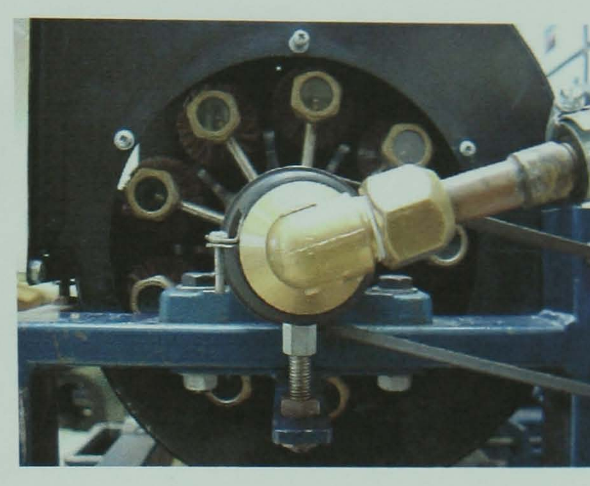


Photograph 8.2: MVHR Heat Pump - Components

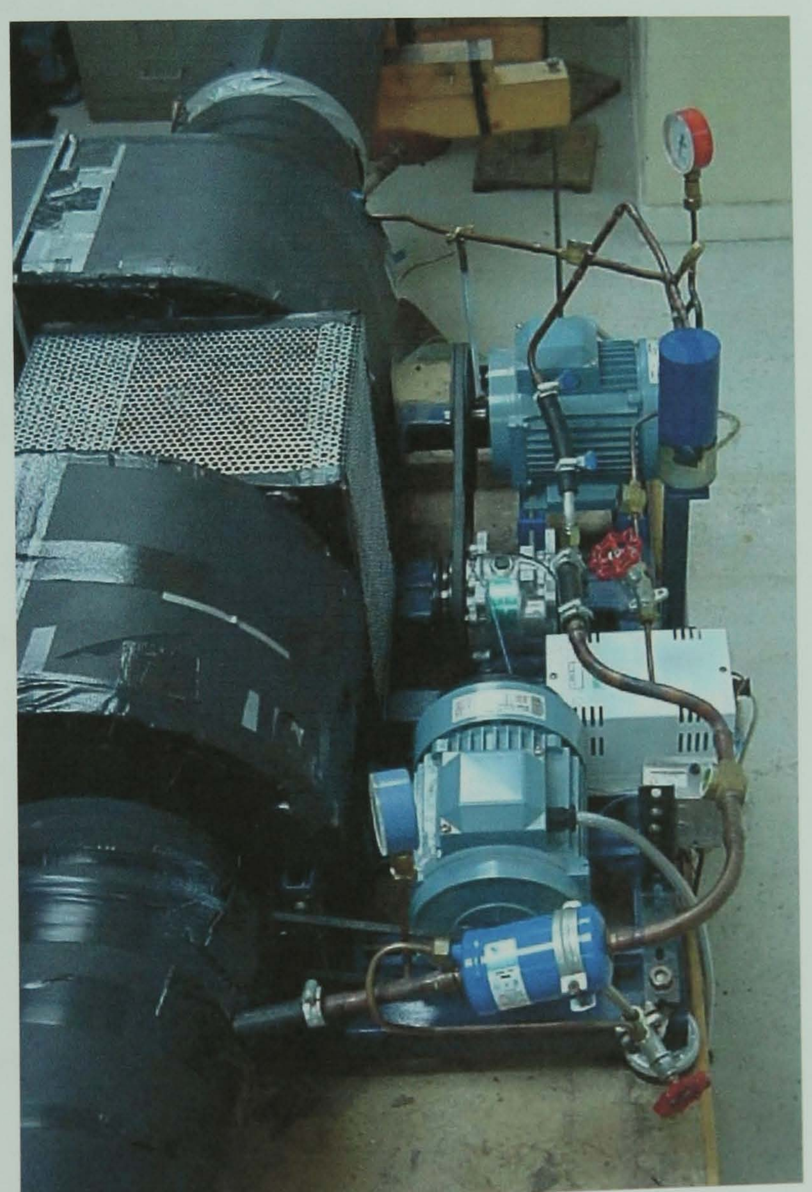


Photograph 8.3: Evaporator/Condenser Headers





Photographs 8.4 and 8.5: Condenser/Evaporator Rotary Shaft Seals



Photograph 8.6: Side view of MVHR Heat Pump Prototype

8.3.3 Pipe sizing

The suction and discharge lines were sized based on acceptable pressure drops along their lengths. Pipe sizing calculations are shown in Appendix F. For the 3/8" discharge line and the 5/8" suction line the calculated pressure drops in the pipes are 0.14148bar and 0.0219bar respectively.

Calculation of equivalent temperature drops:

Discharge:

temperature = 40°C, pressure ≈ 5.32bar

Discharge pressure – ΔP = 5.32 - 0.14148

= 5.17852bar (t_{sat} = 39.06°C)

Therefore drop in temperature in discharge line = 40 – 39.06

= **0.94°C**

The allowable drop in temperature in the discharge line should be lower than 1.1°C [Trane, 1965]. Therefore a temperature drop of 0.94°C satisfies this requirement. A high discharge velocity is a requirement of the helical oil separator. A small margin still remains to allow for any drops in temperature resulting from a pressure drop across the oil separator.

Suction:

temperature = 0°C, pressure ≈ 1.57bar

Suction pressure – ΔP = 1.57 - 0.0219

= 1.5481bar (t_{sat} = -0.29°C)

Therefore drop in temperature in suction line = 0 – (-0.29)

= **0.29°C**

The allowable drop in temperature in the suction line should not be greater than 1.1°C [Trane, 1965]. Therefore a temperature drop of 0.29°C satisfies this requirement. The drop is very low and therefore any drops in temperature resulting from a pressure drop across the filter drier should be allowable.

8.3.4 Refrigerant charge

The approximate mass of isobutane in each of the system components during operation was calculated so that an approximate refrigerant charge could be found for the whole system. Calculations are shown in Appendix F and the component masses are shown in Table 8.3.

Component	Mass (grams)
Discharge line	1.9
Oil separator	4.1
Condenser	92.6
Condenser header	53.3
Evaporator header	82.2
Evaporator	63.7
Suction Line	1
Filter Drier	1.1
Oil separator & pressure switch pipe work	0.6
Lines between headers	7.6
TOTAL	**308**

Table 8.3: Refrigerant Charge

Volume required in charging cylinder if the system is charged at lab room temperature (18°C, $\rho_{f18°C} = 559.356 kg/m^3$).

Isobutane volume in charging cylinder = 0.308/559.356

$= 5.506 \times 10^{-4} m^3$

= **551ml**

8.3.5 Condenser/Evaporator size

At 700 rpm the velocity of air over the fins is:
(R = distance between pipe axes and axis of revolution = 100mm)

$$v = 2\pi R\,(700/60)$$
$$= 7.33\text{m/s (24fps)}$$

To simplify the calculation, the theoretical heat transfer is calculated for the equivalent area of plate finning.

Substituting v into equation 6.4:

$$h = 0.5\,(24)^{0.8}$$
$$= 6.3\ \text{Btu/hr.ft}^2\text{.F}$$
$$= 36\ \text{W/m}^2\text{K}$$

Calculating the theoretical heat transfer:

Surface area of finning = 3.7m^2
Assume fin efficiency is 88%

$$Q = h\ \Delta t\ \eta_f A_f$$
$$= 36 \times 40 \times 0.88 \times 3.7$$
$$= 4.7\ \text{kW}$$

The calculated theoretical heat transfer rate meets the system requirement of 4kW.

8.4 Conclusions

The knowledge gained from testing the revolving heat pipe MVHR systems helped in developing a revolving heat pipe/heat pump system. The prototype heat pump uses revolving wire finned pipes to both impel air and exchange heat. The 'environmentally friendly' hydrocarbon, isobutane, is used as the refrigerant. The charge of isobutane (≈300 grams) is kept below the maximum quantity allowed for residential heat pump applications. The system is designed to be energy efficient with a high theoretical COP. Heat is recovered from a dwelling's exhaust air which is used as the heat pump's heat source. The system can be used for heating or cooling by altering the supply and exhaust air flows over the condenser or evaporator. Appendix G contains work on further development of the system, incorporating an ejector compressor.

9. TESTING OF MVHR HEAT PUMP SYSTEM

The testing and performance evaluation of the prototype MVHR heat pump system is covered in this chapter. The system was tested in terms of its heat pump performance, enabling the calculation of COPs.

9.1 Heat pump performance tests

The MVHR heat pump system was tested using the same experimental set-up used to test the revolving heat pipe systems. The set-up is shown in Figure 7.1 and Photograph 7.1. Over a range of air flows and inlet temperatures, measurements of outlet air temperatures were recorded. For all tests the supply and exhaust flow rates were kept the same. Other measurements taken were the vapour pressures in the suction and discharge lines, the rotational speeds of the revolving pipes and the compressor and the power input to the compressor and revolving pipe/fan motors. The relative humidity was also recorded in the outlet ducts using hygrometers, shown in Photograph 9.1, positioned within the ducts. The results of testing can be found in Appendix H.

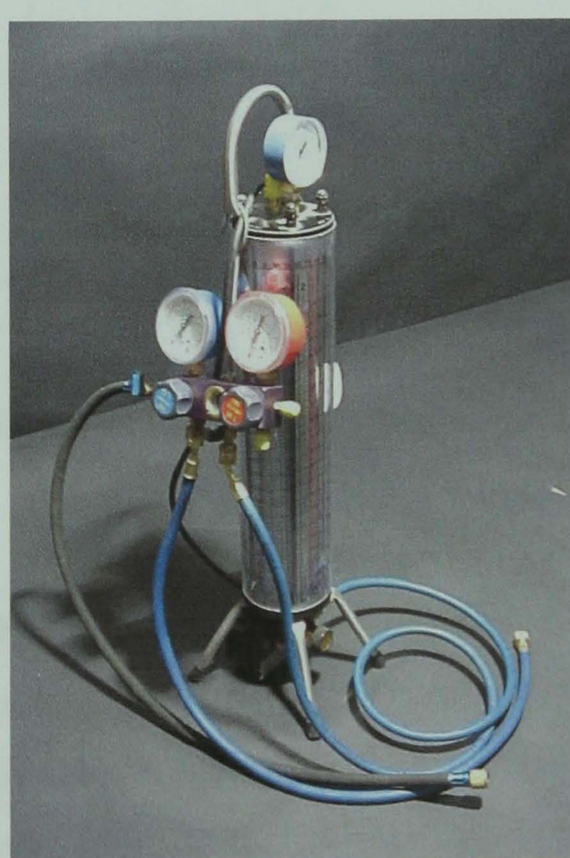
Photograph 9.1: Digital Hygrometer

Two COPs have been calculated, the heating/cooling COP and the system COP. The heating/cooling COP is the ratio of heating/cooling to the electrical power consumed by the compressor (see equations 5.2 and 5.3). The system COP is the ratio of the heating/cooling to the total electrical energy used by the system, i.e. the sum of the electrical energy used to drive the compressor (compressor power) and to rotate the pipes (fan power).

Because the specification of the revolving pipes is very similar to the 9-pipe wire fin system tested in Chapter 7, no fan performance tests were carried out. It is assumed that the air flow/static pressure characteristics are very similar.

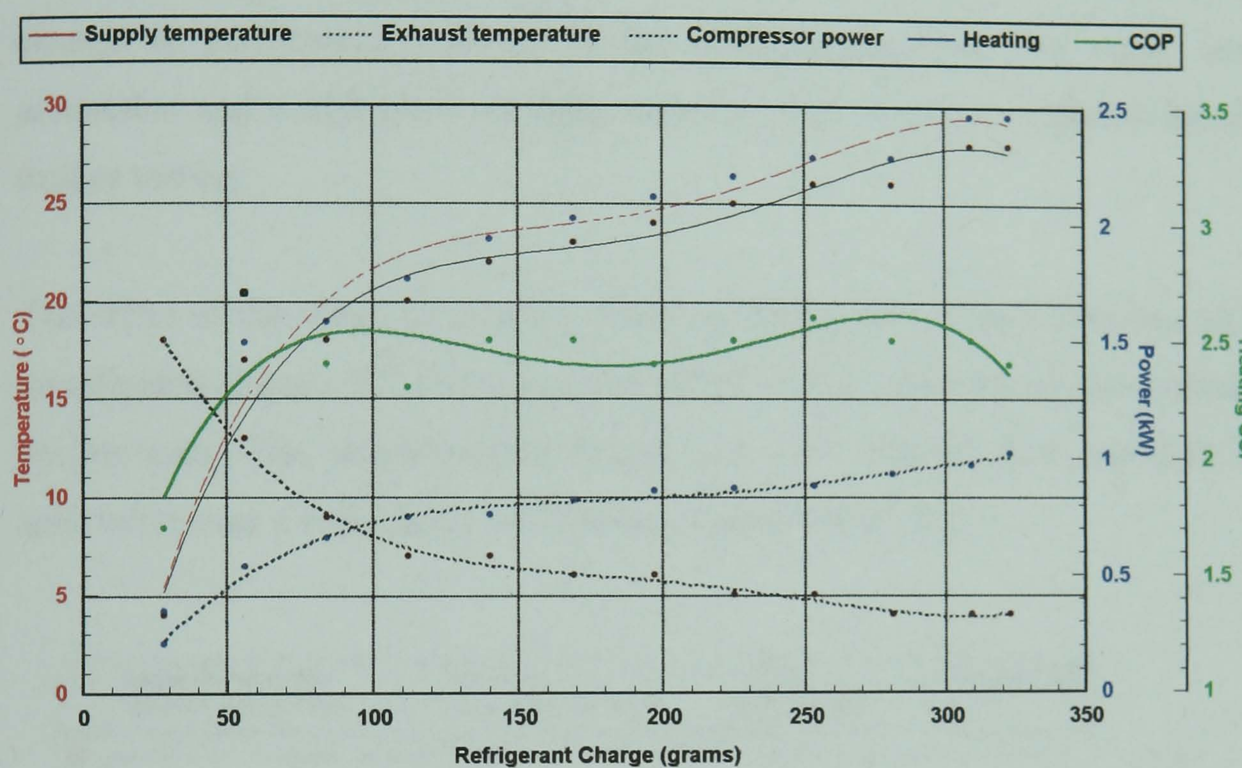
9.2 Commissioning tests

Before charging with isobutane, the system was leak tested using compressed CO_2 at 7.5bar. Once the system was free of leaks, the pipe work was degreased fully by flushing a solvent through the system. A vacuum pump was then used to continuously pull a vacuum and dry out the system. Once under vacuum the system was charged using the charging kit shown in Photograph 9.2.



Photograph 9.2: Refrigerant Charging Kit

The system was charged in 50ml increments of liquid isobutane whilst the compressor and revolving pipes were turning at 2175rpm and 700rpm respectively. The supply inlet and exhaust inlet temperatures were kept at 0°C and 20°C respectively and the supply and exhaust volume flow rates were kept at 257m^3/hr (2m/s air velocity in both outlet ducts). Measurements were taken at each 50ml charge increment. Figure 9.1 illustrates the effect that the refrigerant charge has on the system performance.



Supply Air Inlet Temperature = 0°C,
Fan Speed = 700rpm
Valve Settings = 0.6

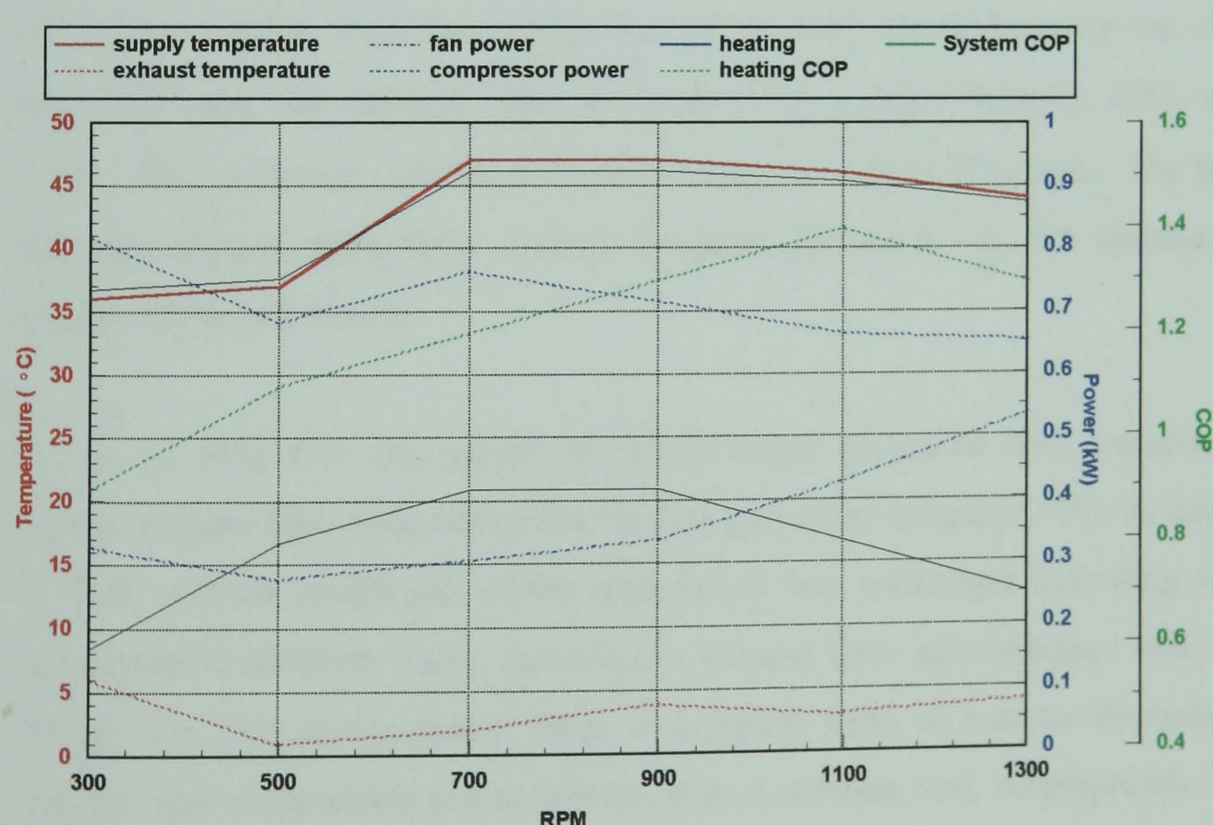
Exhaust Air Inlet Temperature = 20°C
Compressor Speed = 2175 rpm
Volume Flow Rate = 260m^3/hr

Figure 9.1: The Effect of Refrigerant Charge on System Performance

As the refrigerant charge is increased, the electrical power consumed by the compressor increases due to the demands of more refrigerant being pumped, and the resulting greater pressure differential between the evaporator and condenser. The exhaust temperature and heating output also increases with increased charge. The heating COP increases sharply between 30 and 90grams and then remains almost constant. This reflects the heating and compressor power consumption trends. The system operates satisfactorily near the design refrigerant charge of 308grams. All further testing was therefore undertaken using 550ml (≈308grams) of refrigerant. It was noted that during testing a steady flow of water came from the evaporator fan casing through the drain tube. This indicates that the exhaust air was being cooled below its dew point. The recorded values of temperature and relative humidity confirm this. The system is therefore recovering both sensible and latent heat.

A small number of tests were undertaken varying the openings on the expansion valves. Over the range of openings tested there was very little change in performance. However, the performance over this range was acceptable and a mid point opening value of 6/10 of a turn remained for all further testing.

The effect of the speed of rotation of the revolving pipes (fan speed) was also investigated. Figure 9.2 illustrates the effect of fan speed on the heat pump performance. The characteristics shown are for a volume flow rate that is achievable over a wide range of rotational speeds (64m^3/hr).



Supply air inlet temperature = 0°C

Compressor speed = 2175rpm

Refrigerant charge = 309grams (550ml)

Exhaust air inlet temperature = 20°C

Valve setting = 0.6

Volume flow rate = 64m^3/hr

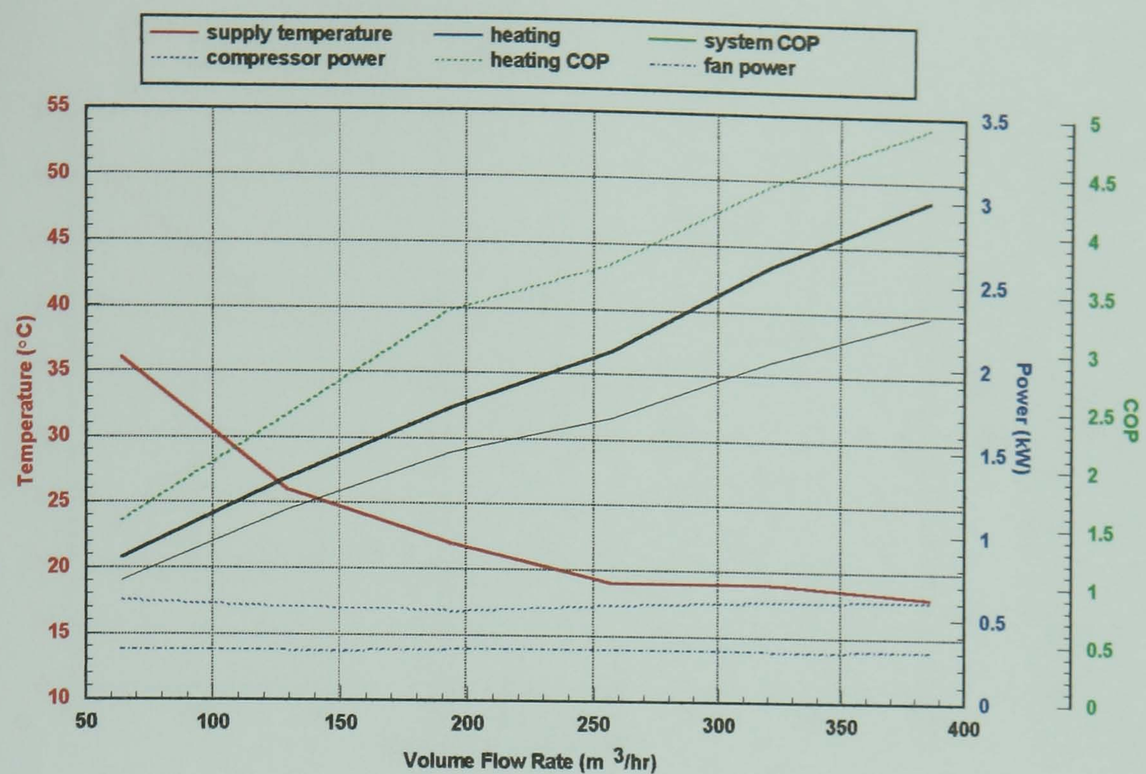
Figure 9.2: The Effect of Fan RPM on System Performance

It is clear to see from Figure 9.2 that the supply air temperature and heating increase between 300rpm and 700rpm. From 700rpm to 1300rpm these values remain almost constant at 46°C and 0.9kW. The heating COP increases with increasing rpm, peaking at 1100rpm. However, the system COP increases between 300rpm and 700rpm and decreases between 900rpm and 1300rpm. This is because the fan power consumption increases with fan speed. Once the fan speed has reached 900rpm, the heating benefit associated with a higher fan speed is outweighed by high values of fan power consumption at these speeds. On the basis of this result all further testing was undertaken with a fan speed of 700rpm. The commissioning tests provided a set of values that could remain constant for all the other testing.

9.3 Heating tests

Tests were carried out to simulate the MVHR heat pump system operating in the winter. For the tests, the supply inlet air temperature ranged between -5°C and 20°C and the exhaust inlet air temperature ranged between 20°C and 35°C. The tests were carried out over a range of volume flow rates. The flow rate was altered using flow control dampers. Figures 9.3 to 9.6 illustrate a sample of the test results.

It can be seen that the supply air temperature decreases with increase in supply volume flow rate, whilst the heating generated increases. This is due to the heat transfer coefficient of the wire finned heat exchanger increasing with the increased air flow. Also, the exhaust volume flow rate increases with the supply rate, and consequently, there is a greater input of heat to the system. The fan and compressor power remain almost constant and, consequently, the COPs increase with the rate of air flow. Typically at 200m^3/hr the system COPs are around 2. The outlet supply air temperature is dependent on the supply and exhaust air inlet temperatures.



Compressor speed = 2175rpm

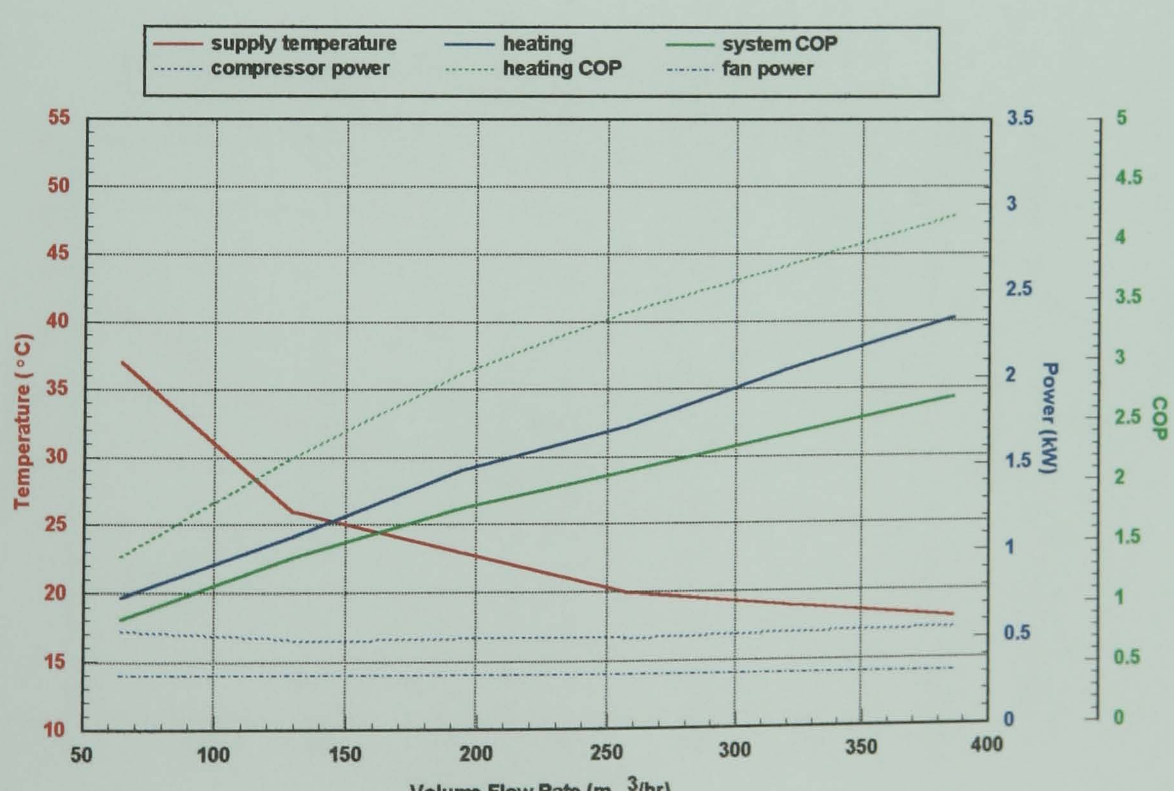
Valve setting = 0.6

Fan speed = 700rpm

Refrigerant charge = 309grams (550ml)

Figure 9.3: MVHR Heat Pump Characteristics:

Supply air inlet temperature = -5°C, Exhaust air inlet temperature = 25°C



Compressor speed = 2175rpm

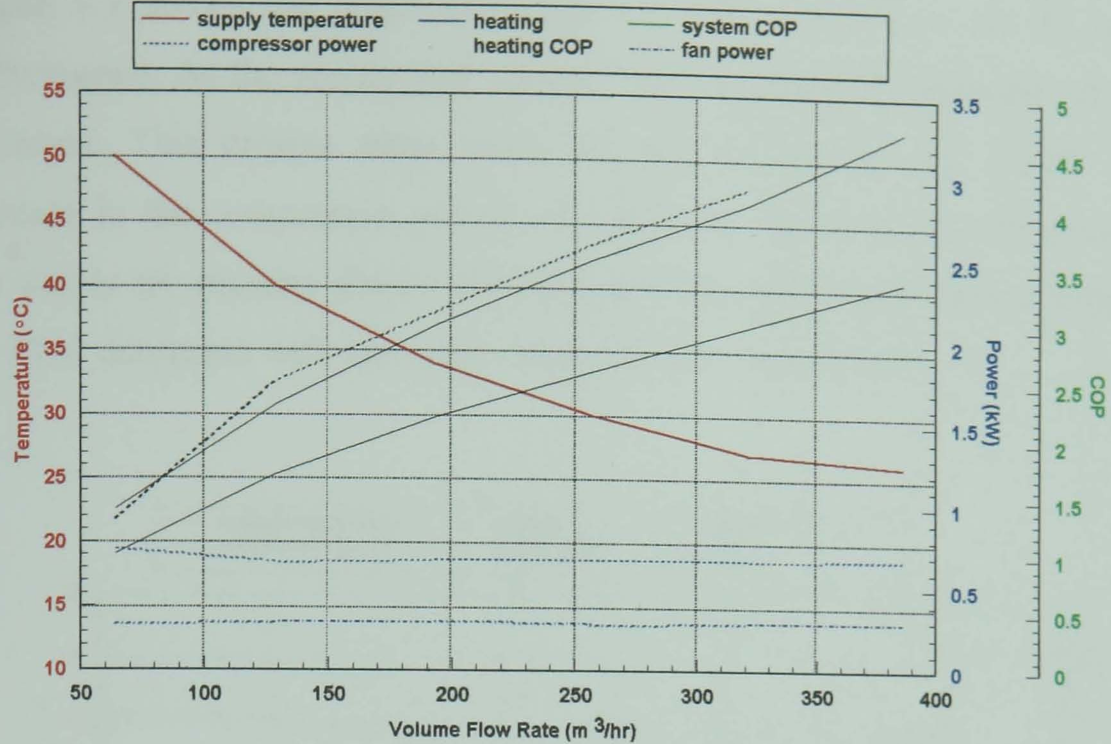
Valve setting = 0.6

Fan speed = 700rpm

Refrigerant charge = 309grams (550ml)

Figure 9.4: MVHR Heat Pump Characteristics:

Supply air inlet temperature = 0°C, Exhaust air inlet temperature = 20°C



Compressor speed = 2175rpm

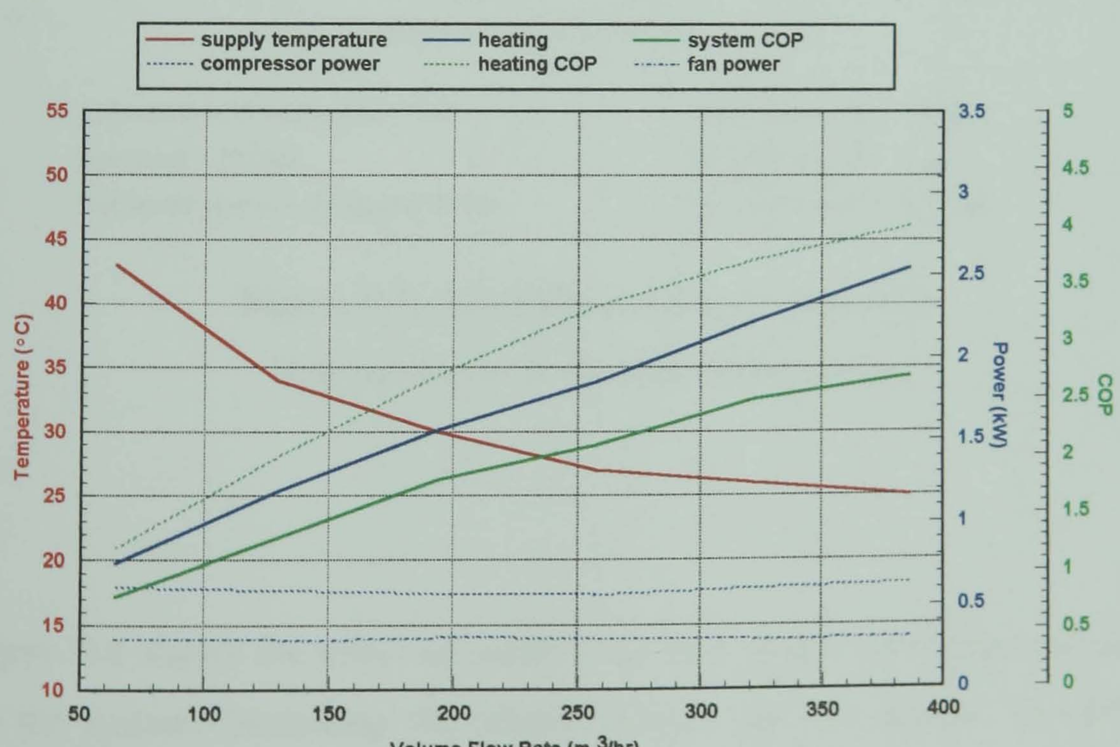
Valve setting = 0.6

Fan speed = 700rpm

Refrigerant charge = 309grams (550ml)

Figure 9.5: MVHR Heat Pump Characteristics:

Supply air inlet temperature = 0°C, Exhaust air inlet temperature = 35°C



Compressor speed = 2175rpm

Valve setting = 0.6

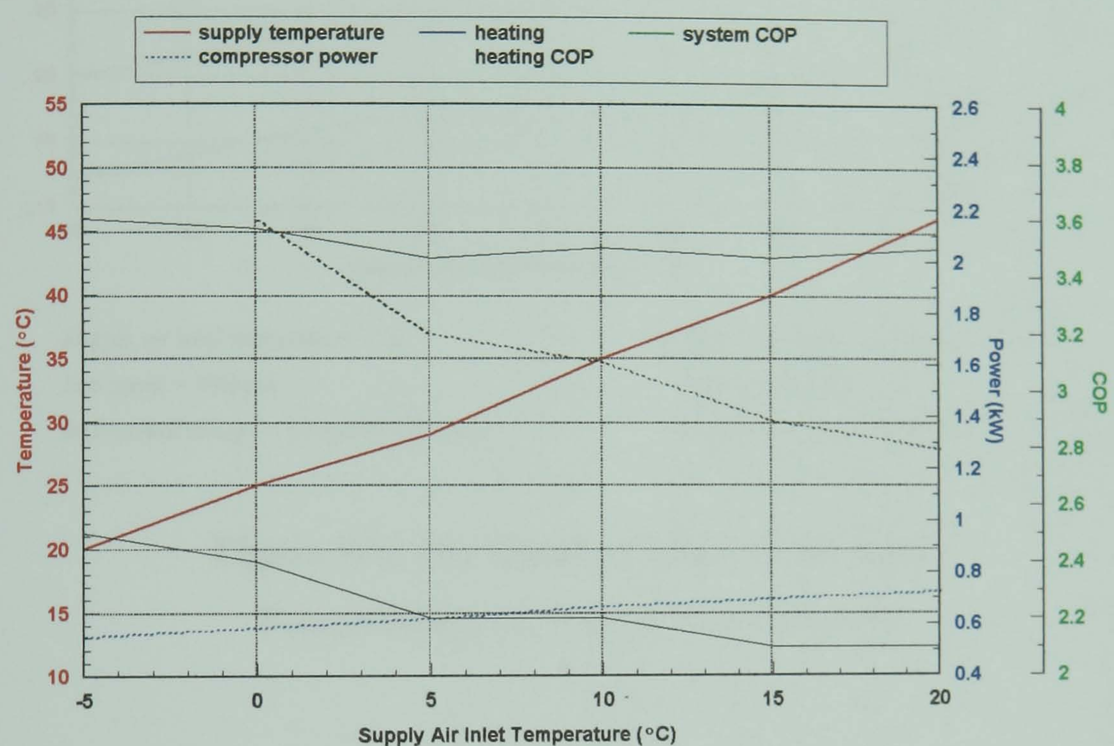
Fan speed = 700rpm

Refrigerant charge = 309grams (550ml)

Figure 9.6: MVHR Heat Pump Characteristics:

Supply air inlet temperature = 5°C, Exhaust air inlet temperature = 25°C

Figure 9.7 shows the effect of supply air inlet temperature on the system performance. As the supply inlet temperature increases the condenser pressure increases. This creates more work for the compressor and results in an increase in the compressor power consumption. Because the heat input into the supply air remains almost constant and the compressor power increases, the COP decreases with increased supply air inlet temperature.

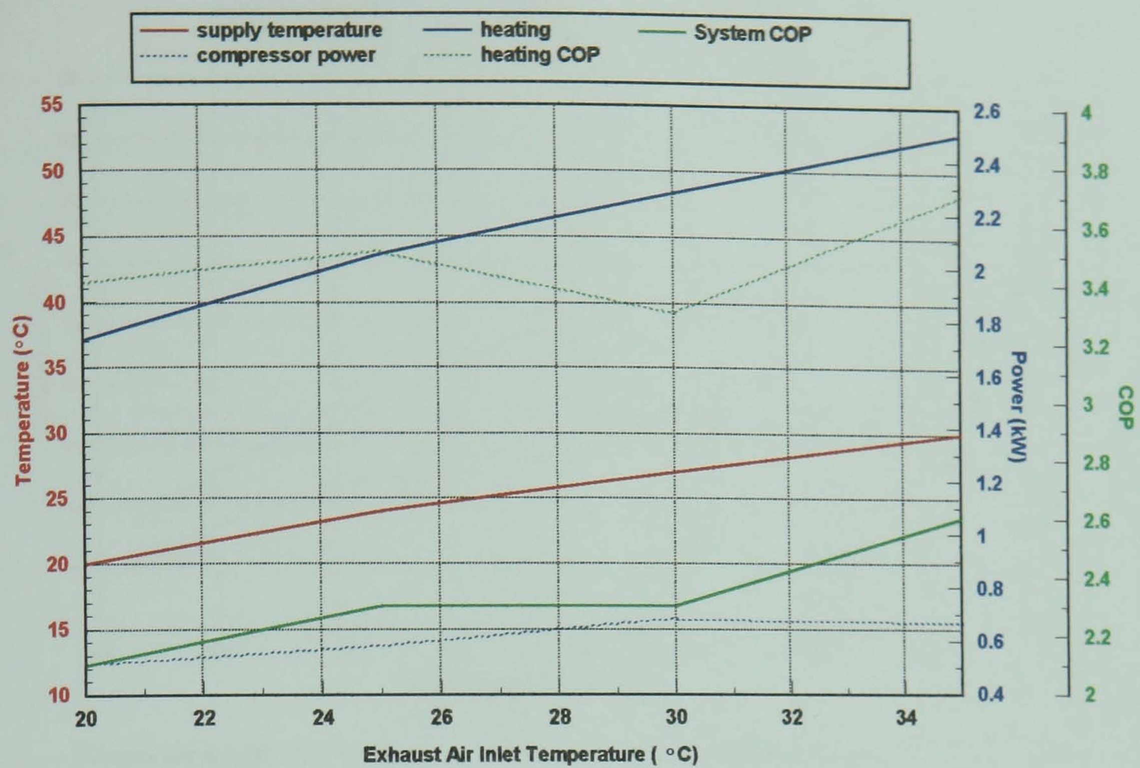


Exhaust air inlet temperature = 20°C
Fan speed = 700rpm
Refrigerant charge = 309grams (550ml)

Compressor speed = 2175rpm
Valve setting = 0.6
Volume flow rate = 260m3/hr

Figure 9.7: The Effect of Supply Air Inlet Temperature on System Performance

Figure 9.8 shows the effect of exhaust air inlet temperature (the heat source) on the system. Increasing the temperature of the heat source increases the heating and temperature of the supply air outlet. The power consumed by the compressor marginally increases, and the heating and system COPs both increase with increased exhaust air inlet temperature.



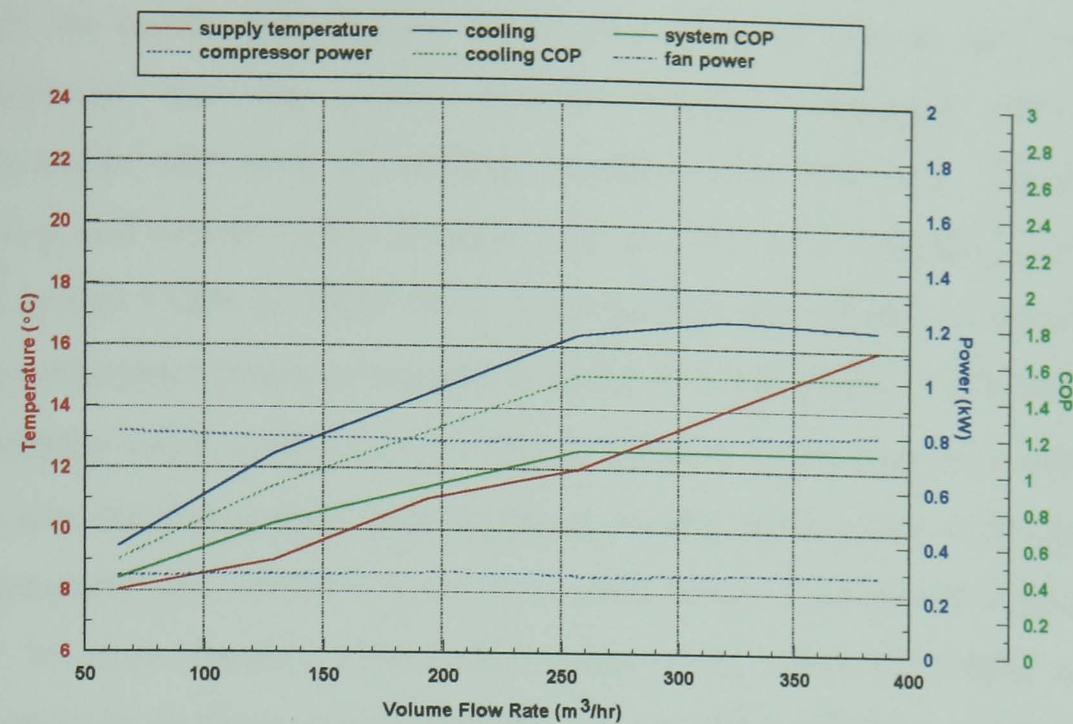
Supply air inlet temperature = 0°C
Fan speed = 700rpm
Refrigerant charge = 309grams (550ml)
Compressor speed = 2175rpm
Valve setting = 0.6
Volume flow rate = 260m3/hr

Figure 9.8: The Effect of Exhaust Air Inlet Temperature on System Performance

9.4 Cooling tests

A series of tests were carried out using the heat pump for summertime cooling. In this mode of operation, the supply air passes over the evaporator and the exhaust air passes over the condenser. In practice, the air flows operating between heating and cooling modes would be controlled by dampers.

The supply air inlet temperature, i.e. the temperature of the warm fresh outside air, was varied between 25°C and 40°C for the tests. The exhaust outlet air temperature was kept at 20°C, this figure representing the inside air temperature of the dwelling. Figures 9.9 and 9.10 illustrate a sample of the cooling mode test results.



Compressor speed = 2175rpm

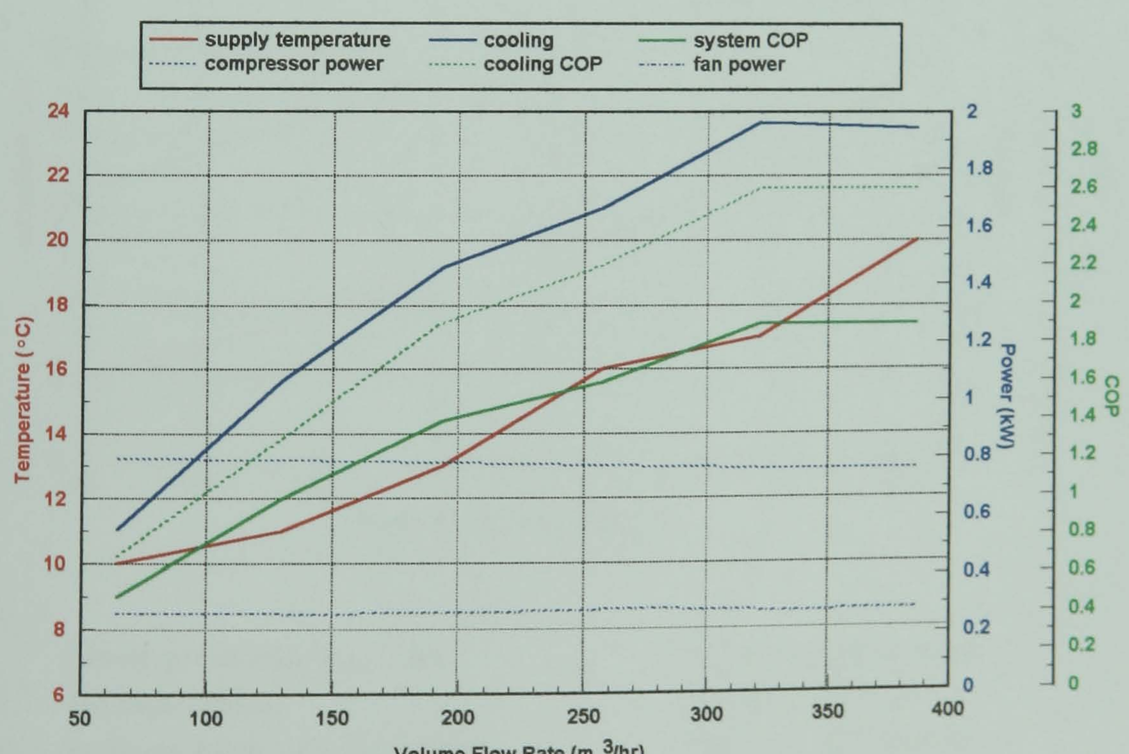
Valve setting = 0.6

Fan speed = 700rpm

Refrigerant charge = 309grams (550ml)

Figure 9.9: MVHR Heat Pump Characteristics, Cooling Mode:

Supply air inlet temperature = 25°C, Exhaust air inlet temperature = 20°C



Compressor speed = 2175rpm

Valve setting = 0.6

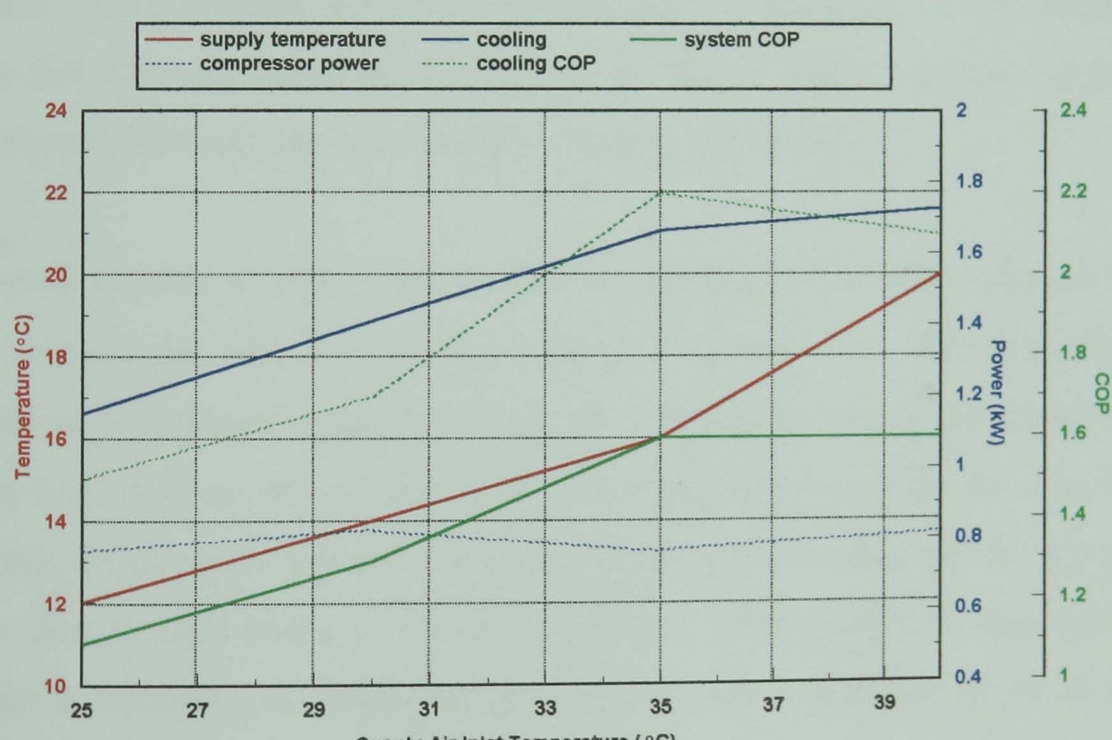
Fan speed = 700rpm

Refrigerant charge = 309grams (550ml)

Figure 9.10: MVHR Heat Pump Characteristics, Cooling Mode:

Supply air inlet temperature = 35°C, Exhaust air inlet temperature = 20°C

From the results one can see that over a range of volume flow rates, the compressor and fan power remains virtually unchanged. The supply temperature and level of cooling increases with flow rate. The resulting cooling and system COPs increase with air flow rate. Typically, the cooling and system COPs at 200m^3/hr in cooling mode are 1.9 and 1.4 respectively. The compressor power consumed is higher in cooling mode (≈800Watts) than in heating mode (≈600Watts). This is because a large pressure differential between the evaporator and condenser results from a very high condenser temperature and pressure when in cooling mode. Therefore the compressor has to work harder. Figure 9.11 shows the effect of supply air inlet temperature on the system performance in cooling mode for a supply flow rate of 260m^3/hr.



Exhaust air inlet temperature = 20°C

Fan speed = 700rpm

Refrigerant charge = 309grams (550ml)

Compressor speed = 2175rpm

Valve setting = 0.6

Volume flow rate = 260m3/hr

Figure 9.11: The Effect of Supply Air Inlet Temperature on System Performance in Cooling Mode

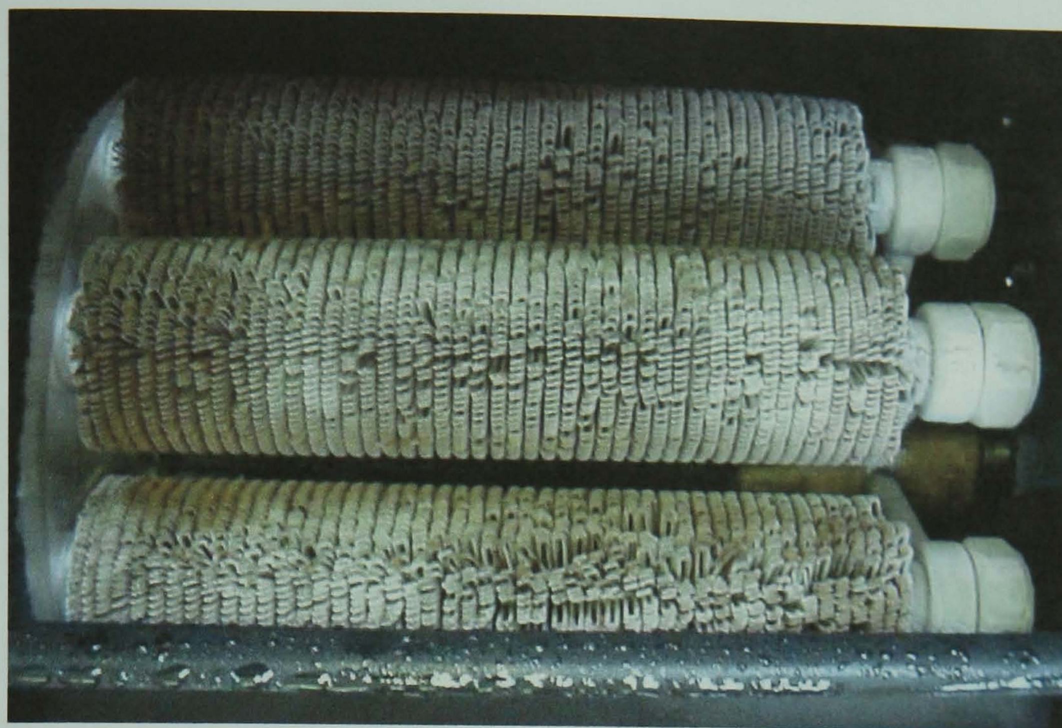
Figure 9.11 shows that as the inlet temperature of the supply air increases, the outlet temperature, cooling and COP also increase. The compressor power is relatively unaffected by the supply air inlet temperature.

9.5 Other testing

9.5.1 Frosting of evaporator

It was noted that over the range of conditions tested, the evaporator never suffered from frosting. The temperature of air passing over the evaporator was always high enough to prevent frosting occurring. Because the heat exchangers are revolving, the water vapour that condensed out of the air passing over the evaporator was thrown off due to centrifugal forces. This means that a thermal barrier of water never accumulates on the evaporator. The fact that water does not collect on the finning will also reduce the amount of frosting if conditions are such that frosting will occur.

A small number of tests were carried out varying the air flow rates, and with the exhaust air inlet temperature kept at 5°C and 10°C. This was done to simulate an extreme condition when the temperature inside the dwelling falls very low, such as after a long period of being unoccupied during a very cold winter. In practice, the internal temperature of a dwelling should not fall as low due to high levels of thermal insulation. The results of these tests are given in Appendix H. When the exhaust flow rate reduced to the point where the temperature of the exhaust outlet air reached 0°C, frosting on the evaporator pipes could be seen. Photograph 9.3 shows the frosted evaporator pipes. Further work is required to fully determine the effects of rotation on evaporator frosting.



Supply air inlet temperature = 0°C

Exhaust air inlet temperature = 5°C

Volume flow rate = 260m^3/hr

Photograph 9.3: Frosting of Evaporator Pipes:

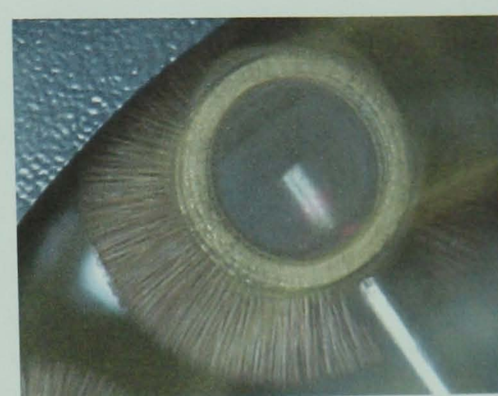
9.5.2 Refrigerant pool in evaporator pipes

At different fan speeds a stroboscope was used, in conjunction with a digital camera, to photograph the refrigerant pool through the glass end caps on the evaporator pipes during operation. Photographs 9.4 to 9.10 show the refrigerant pool at different rpms. The photographs clearly show the refrigerant forming a layer on the inner pipe wall furthest away from the axis of rotation. At 150rpm the centrifugal force is only just greater than the gravitational force and, as a result, the refrigerant layer does not fully form on the inner pipe wall. The photographs for all the other speeds are very similar, showing the refrigerant pooling on the inside pipe walls furthest from the axis of rotation.

9.4: 150rpm

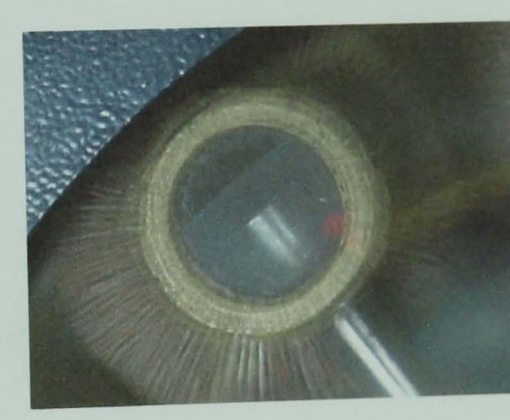
9.5: 300rpm

9.6: 450rpm



9.7: 600rpm

9.8: 750rpm



9.9: 900rpm

9.10: 1050rpm

Photographs 9.4 – 9.10: Refrigerant Pools In Evaporator Pipes

9.6 Economic Analysis

The pay back period for the MVHR heat pump system is calculated in the following section.

Capital Costs:

MVHR Heat Pump prototype:

Wire finned pipes without mesh wicks (×9), £27.50 each:	£247.50
Wire finned pipes with mesh wicks (×9), £54 each:	£486
Fan casings (×2), £128.40 each:	£256.80
Brush seals (×2), £8.90 each:	£17.80
Shaft, ∅30mm silver steel:	£45.22
Bearings, ∅30mm pillow block type, (×2), £23.01 each:	£46.02
Motor, 1.5kW, 3-phase, 4-pole (×2), £116.50 each:	£233
Motor slide rails (×2), £27.30 each:	£54.60
Drive belts:	£8
Pulleys & bushes:	£38.16
Frame:	£15
Feet, anti-vibration mounts (×4), £14.24 each:	£56.96
Rotary shaft seals (×2), £70 each:	£140
Compressor (Sanden SD5H09FL5072):	£148
Compressor fittings:	£20.35
Compressor oil:	£8.84
Pressure switch:	£25.35
Oil separator:	£64.69
Isobutane (300g):	£50
Filter drier:	£24.62
Pipe work:	£10
Expansion valves (×4), £23.80 each:	£71.40
Perspex revolving seals:	£3
Bushes (×7):	£6.30
Pipe clamps:	£50

Additional fixings - nuts & bolts etc.:	£10
Ductwork, diffusers & extract grilles:	£100
Three-phase inverter:	£612.75
Three-phase inverter - key pad:	£54
Three-phase inverter - filter:	£154.57

Total (excluding labour)	**£3058.93**

For a domestic installation, the three-phase inverter and associated items are not required and can therefore be deducted from the total capital cost. In practice, only one motor will be used to drive the compressor and rotate the pipes. Therefore, the cost of the second motor and associated items can be deducted.

Revised Total (excluding labour)	**£2088.88**
Labour (assume 10% total cost)	£208.89

Total Capital Cost of Prototype	**£2297.77**

Assuming mass production reduces the total capital cost by a factor of 5:

Total Capital Cost of Commercial System	**£460**

Space Heating Costs

From section 7.5 the annual space heating demand is 13000kWh and the space heating cost, per annum, using gas and electricity are £280 and £845 respectively. These are the cheapest and most expensive heating options.

MVHR Heat Pump System Running Costs and Output

The number of days on which heating is required is 116. This is based on the mean annual heating degree-day total for the 18 UK degree-day regions over 20 years (1976-1995) [CIBSE, 1999].

The power input to the MVHR Heat Pump system, running under typical operating conditions, is 900Watts. The energy consumption is calculated, assuming the system runs for the heating period i.e. 116 days/year (2784 hours/year).

Annual energy consumption for MVHR Heat Pump system $= 2784 \times 0.9$
$= 2506$ KWh

The annual cost of running the system $= 6.5 \times 2506$
$=$ **£162.86**

Typically the system output is 2kW. Therefore, over the heating period the system provides $2784 \times 2 =$ **5568KWh**

Pay Back Period

The MVHR Heat Pump is used to provide supplementary heating to the existing heating system. Using the MVHR Heat Pump, the heating provided by the existing system is reduced.

Reduction in heating $= 13000 - 5568$
$=$ **7432kWh**

Revised Gas heating cost $= (1.4 \times 7432)/ 0.65$
$=$ £160

New cost with gas and heat pump $= 160 + 162.86$
$=$ £322.86

The cost of space heating using gas alone is £280/annum. The cost using gas supplemented by the MVHR Heat Pump is £43/annum more expensive. There is therefore no pay back. This is because of the very low cost of gas compared to electricity. The running cost of the heat pump system not only takes into account the cost of space heating, but also the cost of ventilation. The additional benefits associated with the provision of controlled ventilation may outweigh the economic penalty. The provision of summer time cooling may also be attractive to potential users.

If the system is used to supplement electric heating the economics are much more favourable, as the following calculations show.

Revised Electricity heating cost = 6.5 × 7432
= £483

New electric heating cost = 483 + 162.86
= £645.86

The savings made using the MVHR heat pump are:

Cost of electricity saved = 845 – 645.86
= **£199.14**

Therefore the payback period = 460 / 199.14
= **2.3 years**

It is therefore attractive, both in terms of cost savings and improved indoor air quality, to use the system in conjunction with an electric heating system.

9.7 Conclusions

The testing of the prototype MVHR heat pump system has shown that the system performs satisfactorily for heat recovery, heating and cooling, whilst providing an adequate level of ventilation. Over a range of air inlet temperatures and air flow rates, when in heating mode, the heating COP ranged between 1.3 and 4.9 and the system COP ranged between 0.8 and 3.4. When in cooling mode, the cooling COP ranged between 0.5 and 2.6 and the system COP ranged between 0.4 and 1.9. Under typical operating conditions the COPs would be, approximately, at a mid point between these ranges. The heating COP is less than the calculated theoretical value due to system inefficiencies.

Over the range of inlet temperatures tested and, between $200m^3/hr$ and $250m^3/hr$ volume flow rate, the system typically provides 2kW of heating or 1.5kW of cooling. This value is below the design figure of 4kW rejected at the condenser. The heating figure could be improved by enlarging the condenser/evaporators or by increasing the supply and exhaust air flow rates. Also, the valve settings used may be restricting the refrigerant flow too much. The results show that the evaporator pressure is below the design condition. This would result in the mass flow rate being lower than that designed for and as a result the heating output would be less. The system requires optimising using expansion valves appropriately sized for isobutane. Although the R134a compressor performed satisfactorily an appropriately sized isobutane compressor should be used for system optimisation.

Although a flammable refrigerant has been used, the charge of refrigerant has been kept low. The charge is well below the maximum allowed for a domestic application (<1.5kg, BS4434, British Standards Institute, 1995).

The heating/cooling of the supply air and the COP are greatly affected by the volume flow rate and inlet air temperatures. The fan speed has a significant effect on the system COP. This is because as the fan speed increases the fan power consumption also increases. Test results show that a fan speed of between 700rpm and 900rpm is the most efficient.

Economic analysis shows the system compares well with an electric heating system but is less favourable than gas heating due to the low cost of gas at present. The additional benefits of improved indoor air quality and the provision of summer time cooling may outweigh the economic penalties when used in place of gas space heating.

The system could be improved by placing the condenser header inside the fan casing. At present, useful heat from the header is lost and not transferred to the supply air flow. Pipe work lengths and component sizes could also be optimised to improve the system performance and further reduce the mass of isobutane. Now that the optimum speeds for the compressor and revolving pipes are known a single motor could be used with appropriately sized pulleys. This would result in an improved system COP.

Because the rotary shaft seals are the most likely point from which a refrigerant leak could occur, they should not be placed in the air ducts. This could be achieved using 90° bends in the inlet ducts just before the fan casings. The main shaft could be extended at both ends through the duct wall allowing the seals to be positioned out of the airways.

10. CONCLUSIONS

The advent of energy efficient and more airtight buildings has clearly created a need for ventilation, through means other than natural infiltration. Since it is not energy efficient during winter months to rely on windows for whole house ventilation, the use of mechanical ventilation with heat recovery is a practical solution. Many studies have shown that using such systems dramatically improves indoor air quality. In particular, the systems have been shown to be very good at reducing levels of condensation and dust mite populations. When combined with heat pumps, these systems can provide a significant proportion of a dwelling's space heating requirement, as well as summer time cooling. The majority of systems employed in homes at present, use cross flow plate heat exchangers to transfer heat from the exhaust air to the supply air. These systems require regular maintenance, which includes cleaning the heat exchanger core and changing air filters. Without this maintenance work the system efficiency deteriorates. As the systems are often positioned out of sight in inaccessible places, such as lofts, maintenance may be overlooked by the dwellers. It is therefore important that such systems should be robust and maintenance free. The work outlined in this thesis has produced several prototypes that provide mechanical ventilation with heat recovery and, in one case, heat pumping. Revolving heat exchangers have been used to enhance heat transfer, reduce fouling and negate the requirement for separate fans and heat exchangers.

10.1 Prototype MVHR revolving heat pipe systems

The prototype systems use an annular array of revolving heat pipes to both impel air and transfer heat. The systems were designed to transfer up to 1.5kW of heat from exhaust air, with as high as possible heat exchanger efficiency. The systems are capable of providing 0.35ach for a typical four bedroom detached house (200m^3/hr continuous, 250m^3/hr boost). All testing was carried out under balanced flow conditions. Several different variations of the system were constructed to determine the best configuration. Tests showed that, both in terms of fan performance and heat exchanger efficiency, wire finning performed better than more conventional plate fins. The wire finned heat pipe system has approximately 10 percentage points greater heat exchanger efficiency than the equivalent system with plate finning, even though the plate finning has a greater surface area. The wire finned pipes have high heat transfer coefficients enabling good air side heat transfer. The static pressure generated from the revolving wire finned pipes is approximately 15% higher than that generated from the plate finned pipes.

Tests on the effect of working fluid charge on heat exchanger efficiency, showed that a 10% by volume charge is marginally better than a 20% by volume charge. The best performing system has a double row of 24 wire finned heat pipes. The double row system presents a greater number of heat pipes, with more surface area, to the air passing over them. The 24 pipe system has approximately 10 percentage points better heat exchanger efficiency than the 9 pipe single row system. The effect of tapering the heat pipe bundle was also investigated. Except at high speeds of rotation, the tapered system performs approximately 5 percentage points better, in terms of heat exchanger efficiency, than the non- tapered system with the same heat pipes.

The heat exchanger efficiency generally increases with speed of revolution up to approximately 900 rpm. At greater speeds the efficiency tends to reduce. Higher air velocity enhances the external heat transfer, and greater centrifugal forces increase the rate of working fluid return from the condensers. This was not the case for the 20% filled system and the tapered system. In both these cases the heat exchanger efficiency falls at high speeds. This is probably due to higher pressures in the thicker refrigerant pool, from increased centrifugal force at higher speed. As the pressure in the liquid increases with speed, nucleate boiling is inhibited.

The heat exchanger efficiency is largely affected by the volume flow rate due to the residence time of air in the fan casings. The efficiency is greatest at low flow rates and vice versa. The power consumption increases with rpm and slightly increases with volume flow rate. A fan speed of around 800rpm works well in terms of heat exchanger efficiency and fan performance. At this speed enough airflow is provided with acceptable static pressure. The benefits of running the system at a higher speed would be outweighed by the increased electrical power consumption. In order to have enough heat exchanger area the systems are larger than conventional fans and, consequently, are able to impel more air than is required. When the systems are operating the airflow should be controlled by dampers, with the speed of revolution kept high to maintain optimum heat transfer and enough fan pressure.

The electrical power used to rotate the pipes ranged between 60Watts and 400Watts, over a range of flow rates and rotational speeds for the 24 pipe system. At 900rpm the power consumed is approximately 200Watts. The power consumption is slightly less for the 9 pipe systems, as they have less mass and air resistance.

The 24 pipe prototype system has a heat exchanger efficiency of up to 65%. At 200m^3/hr and static pressure 100Pa, the efficiency averages 53%. These values are similar to a plate heat exchanger system which was also tested under the same operating conditions. The COP of the 24 pipe system at 900rpm and 200m^3/hr ranges between 2 and 6, depending on the inlet air temperatures. Economic analysis showed annual fuel savings would result if the system was used.

10.2 Prototype MVHR heat pump system

A hybrid version of the revolving heat pipe system was produced, which works as an air to air heat pump system. The system uses a small refrigerant charge ($\approx$300grams) of isobutane. This is a hydrocarbon with zero ozone depletion potential and very low global warming potential. The system was designed to provide 4kW of supplementary space heating at volume flow rates of 200m^3/hr to 250m^3/hr. Theoretical calculations were carried out on a number of refrigerants to determine the most suitable choice. Isobutane was chosen due to its high theoretical COPs and low values of condenser pressure. A high COP is important in terms of energy efficiency, and a low condenser pressure imposes less stress on seals and joints in pipe work. By altering the supply and exhaust air flows over the condenser and evaporator, the system can also be used for cooling.

The heating/cooling and the COPs are greatly affected by air flow rate and air inlet temperatures. Over a range of inlet temperatures, and air flows between 200m^3/hr and 250m^3/hr, the system typically provides 2kW of heating and 1.5kW of cooling. These values could be improved by increasing the mass flow rate of refrigerant around the system, increasing the airflow over the heat exchangers or by further optimising the evaporator and condenser. The heating COP over this range is typically 3 and the system COP, which includes fan power consumption, is typically 2. Cooling COPs are typically 1.6 with corresponding system COPs of 1.1. The fan speed has a large effect on the system COP due to increased power consumption with speed. A speed of rotation between 700rpm and 900rpm has been shown to be the most suitable.

Economic analysis showed that if the system is used to supplement electric heating a large saving in fuel costs could be achieved with a pay back period of 2.3 years. If the system is used to supplement gas heating the annual fuel bill to the consumer is increased by approximately 15% due to the very low cost of gas compared to electricity. However, the benefits to the dwellers in terms of improved indoor air quality may outweigh the economic penalty.

10.3 Contribution to knowledge

The development of a novel MVHR system and a novel MVHR heat pump system and the establishment of the performances of these systems has contributed to engineering and scientific knowledge. The originality of design is the use of revolving heat pipes and heat exchangers to both impel air and transfer heat. The research included the novel use of a hydrocarbon refrigerant in a domestic heat pump system. Further original ideas have been suggested.

10.4 Recommendations for further work

Improved fan casing seals should be developed with less rotational resistance than the brush seals which are used at present. By reducing this resistance the power consumption can be lowered. To further reduce power consumption, the systems should be made more compact and out of lighter components. The current work shown in Appendix C is addressing these points. Testing should be undertaken measuring the temperatures of the revolving pipes, to fully evaluate the system performance. This has not been done to date, due to the difficulty of taking surface temperature measurements on rotating bodies. Tests to establish the non-fouling characteristics of the systems should be undertaken using dust laden inlet air, over a long operating period. The systems require ducts of smaller diameter to make them more practical for domestic installation. A suitable diameter is between 100mm and 150mm. The latest prototype, shown in Appendix C, uses this size of ducting.

Tests using other working fluids in the revolving heat pipes should be carried out, as well as testing the effect of using heat pipes with internally grooved structures. Further work could be carried out using the systems for high temperature heat recovery, such as from boiler flue gases.

The heat pump system could be further improved by using pipes with internal structures, such as turbulator inserts or grooves on the inner pipe walls. A single motor should be used, both to drive the compressor and to rotate the pipes. This would reduce the number of components and increase the system COP by slightly reducing power consumption. The requirement for the tapered condenser pipe bundle should also be investigated. Without the tapered pipes, the refrigerant flow could be reversed using a reversing valve. This would be a more practical method of switching between heating and cooling modes. Further work is required to develop the proposed ejector compressor system shown in Appendix G. Other design ideas are contained in Appendix I. It is proposed that these systems use the revolving heat pipes to impel air, transfer heat and provide air cleaning, with significant modification of relative humidity, using desiccants and evaporative cooling.

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APPENDIX A

Residential MVHR System Manufacturers & Suppliers

Appendix A:
Residential MVHR System Manufacturers & Suppliers

- ABB (UK), Groveland Industrial Estate, Longford Road, Coventry. Tel:01203 368500

- Baxi Air Management Ltd., Unit 20, Roman Way, Ribbleton, Preston, PR2 5BB, UK. Tel: 01772 693700 Fax: 01772 693701

- Johnson & Starley, Rhosili Road, Brackmills, Northampton, NN4 7LZ, UK.
  Tel: 01604 762881 Fax: 01604 767408

- Mitsubishi Electric Europe, Environmental Control Systems Division, Travellers Lane, Hatfield, Hertfordshire, AL10 8XB, UK. Tel: 01707 276100 Fax: 01707 278674

- Venmar Ventilation Inc., Residential Products Group, 550 Blvd. Lemire, Drummondville, QC, Canada, J2B 8A9.
  UK Venmar Supplier: Advanced Ventilation Systems, 18 Snaith Crescent, Loughton, Milton Keynes, Bucks. MK5 8HG, UK. Tel: 01908 609777
  Fax: 01908 609778

- Vent-Axia Ltd. Fleming Way, Crawley, West Sussex, RH10 2NN.
  Tel: 01293 526062

- Villovent, Avenue 2, Station Lane Industrial Estate, Witney, Oxon, OX8 6YD, UK. Tel: 01993 778481 Fax: 01993 779962

APPENDIX B

Component Suppliers

Appendix B: Component Suppliers

- Airflow Developments Limited, Lancaster Road, Cressex Business Park, High Wycombe, Buckinghamshire, HP12 3QP. England. Tel. 01494 525252.

- Dawson & Son Limited, Claytonwood Rise, West Park Ring Road, Leeds LS16 6RH. UK. Tel. 0113 2759321

- Fenner Power Transmission, UK. Tel. 0115 9861886

- Isoterix Limited, Wooler Industrial Estate, Northumberland, NE71 6SL, England. Tel. 01668 281173

- RS Components, UK. Tel. 01536 201234

- Sanden International (Europe) Ltd. Hampshire International Business Park, Crockford Lane, Chineham, Basingstoke, Hants. RG24 8WH. UK Tel. 01256 708888

- Seal Co. Ltd. 66 Lichfield Road, Pelsall, Walsall, West Midlands. WS3 4HL. UK. Tel. 01922692447.

- Venturi Jet Pumps Ltd., Venturi House, Edensor Road, Longton, Stoke on Trent. ST3 2QE. UK Tel. 01782 599800

APPENDIX C

Modified MVHR System Prototype

Appendix C: Modified MVHR System Prototype

The following figures show the latest revolving heat pipe MVHR system prototype.

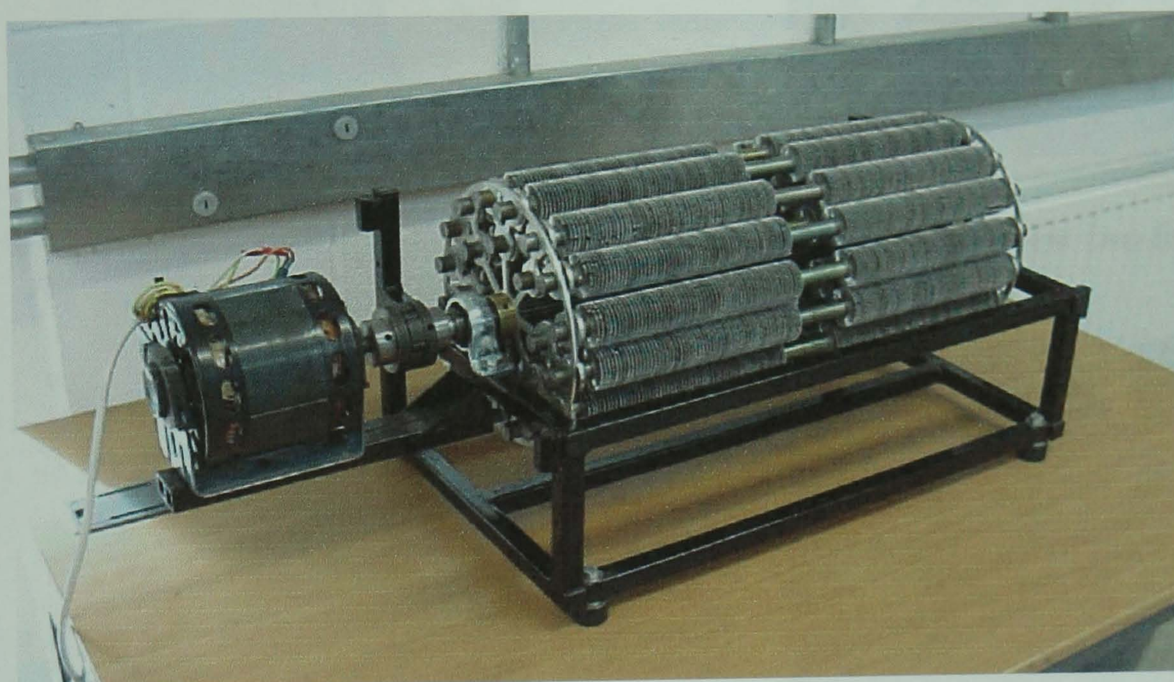


Figure C.1: System with fan casings removed

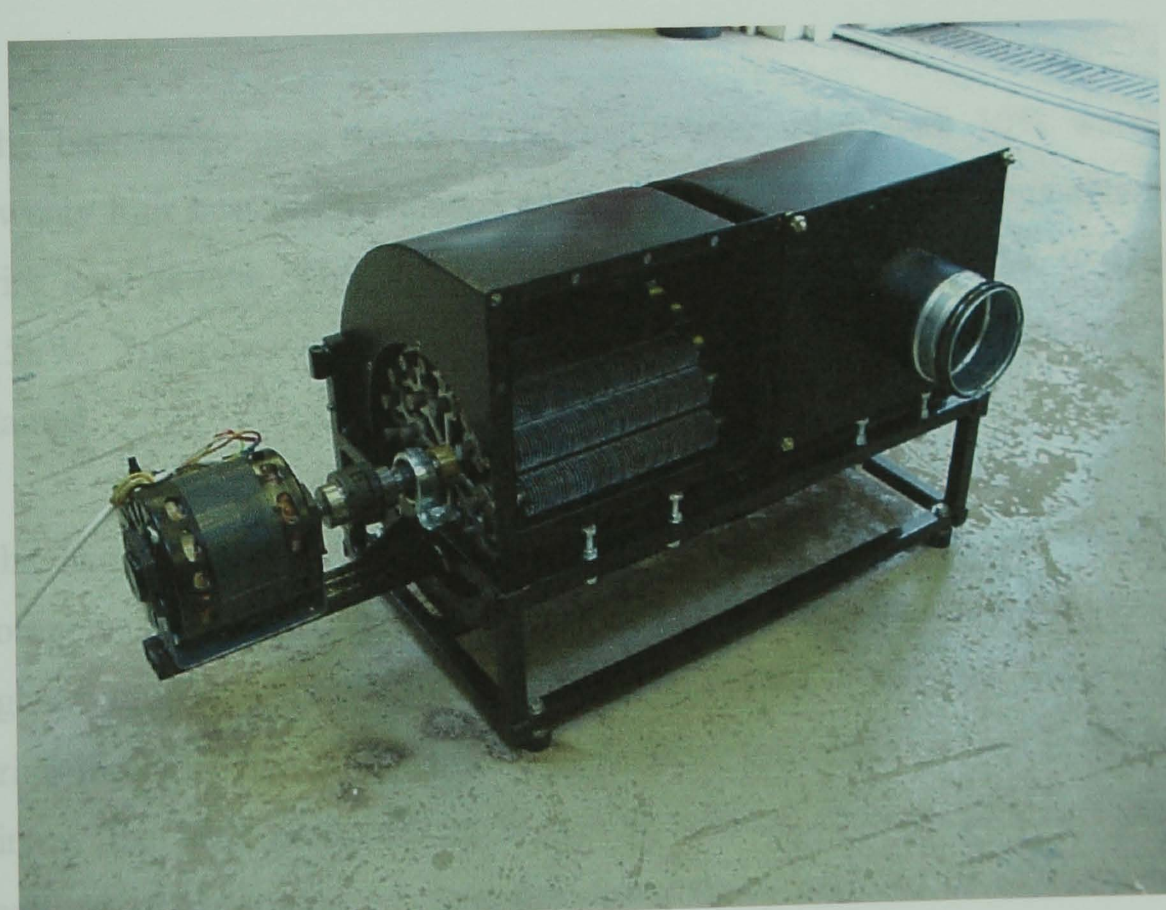


Figure C.2: Revolving Heat Pipe MVHR System (aluminium fins)

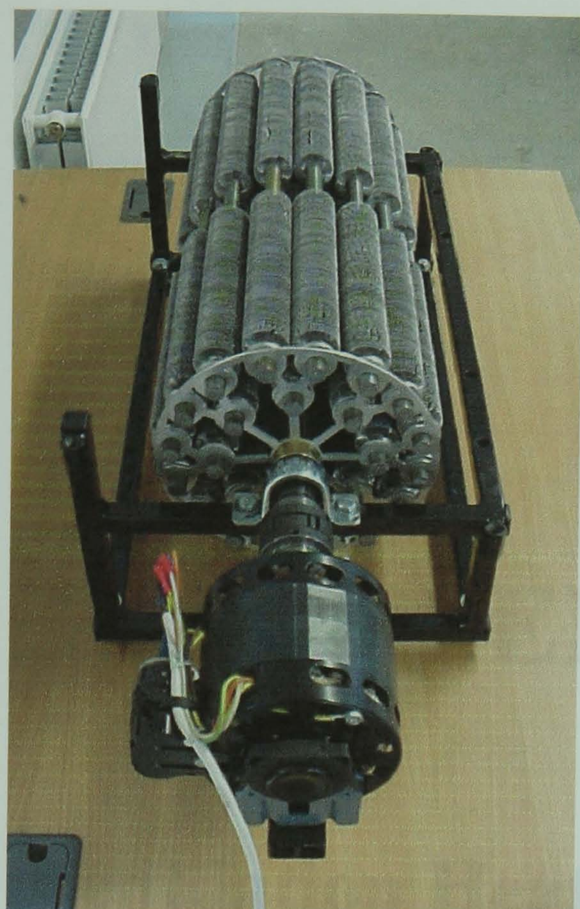


Figure C.3: Above end view

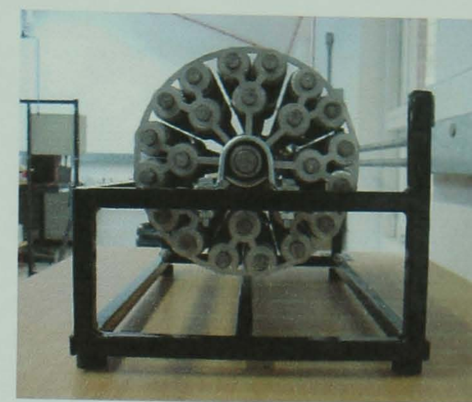


Figure C.4: End view (no fan cases)

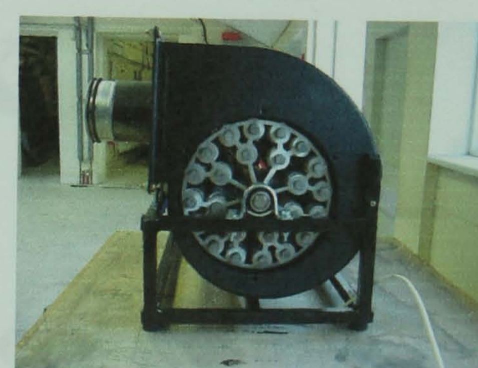


Figure C.5: End view

The heat pipes used are 530mm long copper pipes with 10% water fill. Twenty four heat pipes are used which are finned with 11mm high aluminium wire loop finning. The heat pipes are designed by Isoterix Ltd. to transfer 1.5kW. The pipes are held in place with two 3mm thick aluminium plate clamps.

The latest system has been designed to be more compact and use lighter components. By making the system smaller it is more practical and, by using lighter components, the power consumption should be reduced. Using a direct drive motor is more efficient. Also, because the motor is positioned in the airway the heat produced by it will be utilised. Testing of this system is required to determine whether the down sizing and use of aluminium wire fiining significantly affect the performance. Initial testing has shown that the power consumption is very low. The system uses approximately 90Watts to supply up to 800m^3/hr.

APPENDIX D

Ventilation Heat Recovery Results

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 16.3°C, 55%								
100	0.6	77.2	0	7	20	115.5	35	1.64
100	0.5	64.3	0	8	20	112.7	40	1.60
100	0.4	51.4	0	9	20	109.9	45	1.47
100	0.3	38.6	0	10	20	109.9	50	1.22
100	0.2	25.7	0	11	20	107.1	55	0.92
100	0.1	12.9	0	12	20	107.1	60	0.50
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 16.1°C, 64%								
300	1.5	192.9	0	8	20	119.5	40	4.53
300	1.3	167.2	0	8	20	110.5	40	4.24
300	1.1	141.5	0	9	20	107.6	45	4.14
300	0.9	115.7	0	9	20	101.6	45	3.58
300	0.7	90.0	0	10	20	101.6	50	3.08
300	0.5	64.3	0	10	20	98.6	50	2.27
300	0.3	38.6	0	11	20	98.6	55	1.49
300	0.1	12.9	0	12	20	92.6	60	0.58
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 19.1°C, 56%								
500	2.4	308.6	0	7	20	119.7	35	6.35
500	2.0	257.2	0	8	20	113.2	40	6.37
500	1.6	205.7	0	8	20	110.0	40	5.25
500	1.2	154.3	0	8	20	106.7	40	4.06
500	0.8	102.9	0	9	20	103.5	45	3.13
500	0.4	51.4	0	10	20	103.5	50	1.73
500	0.1	12.9	0	12	20	100.3	60	0.53
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 19.1°C, 56%								
700	3.2	411.5	0	6	20	164.9	30	5.29
700	2.7	347.2	0	7	20	161.0	35	5.31
700	2.2	282.9	0	9	20	157.1	45	5.66
700	1.7	218.6	0	9	20	153.2	45	4.49
700	1.2	154.3	0	10	20	153.2	50	3.51
700	0.7	90.0	0	11	20	149.2	55	2.30
700	0.1	12.9	0	12	20	149.2	60	0.36
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 19.1°C, 56%								
900	4.1	527.2	0	6	20	231.4	30	4.83
900	3.45	443.6	0	6	20	231.4	30	4.06
900	2.8	360.1	0	7	20	221.7	35	4.00
900	2.15	276.5	0	7	20	216.9	35	3.14
900	1.5	192.9	0	9	20	212.1	45	2.86
900	0.85	109.3	0	11	20	207.3	55	2.01
900	0.2	25.7	0	12	20	207.3	60	0.51
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 18.7°C, 51%								
1100	5.0	643.0	0	6	20	300.3	30	4.54
1100	4.3	552.9	0	6	20	294.5	30	3.98
1100	3.6	462.9	0	7	20	288.8	35	3.95
1100	2.9	372.9	0	7	20	288.8	35	3.18
1100	2.2	282.9	0	7	20	283.0	35	2.46
1100	1.5	192.9	0	8	20	277.2	40	1.95
1100	0.8	102.9	0	9	20	277.2	45	1.17
1100	0.3	38.6	0	11	20	271.4	55	0.54
1100	0.2	25.7	0	12	20	271.4	60	0.39
1100	0.1	12.9	0	13	20	265.7	65	0.22

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 18.8°C, 43%								
1300	5.6	720.1	0	5	20	399.8	25	3.19
1300	5.1	655.8	0	5	20	393.0	25	2.96
1300	4.6	591.5	0	5	20	393.0	25	2.67
1300	4.1	527.2	0	5	20	379.5	25	2.46
1300	3.6	462.9	0	5	20	379.5	25	2.16
1300	3.1	398.6	0	6	20	372.7	30	2.27
1300	2.6	334.3	0	7	20	365.9	35	2.25
1300	2.1	270.0	0	8	20	365.9	40	2.07
1300	1.6	205.7	0	8	20	359.1	40	1.61
1300	1.1	141.5	0	10	20	352.4	50	1.40
1300	0.6	77.2	0	11	20	345.6	55	0.85
1300	0.15	19.3	0	11	20	345.6	55	0.21
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 17.7°C, 46%								
1480	6.4	823.0	0	5	20	539.0	25	2.71
1480	5.4	694.4	0	5	20	515.9	25	2.39
1480	4.4	565.8	0	6	20	500.5	30	2.40
1480	3.4	437.2	0	6	20	485.1	30	1.91
1480	2.4	308.6	0	6	20	469.7	30	1.39
1480	1.4	180.0	0	8	20	462.0	40	1.09
1480	0.4	51.4	0	10	20	454.3	50	0.39
1480	0.15	19.3	0	12	20	446.6	60	0.18
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 17.8°C, 54%								
100	0.6	77.2	5	11	20	118.4	40	1.36
100	0.5	64.3	5	12	20	115.5	47	1.35
100	0.4	51.4	5	12	20	115.5	47	1.08
100	0.3	38.6	5	13	20	112.7	53	0.94
100	0.2	25.7	5	13	20	109.9	53	0.64
100	0.1	12.9	5	15	20	107.1	67	0.41
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 18.9°C, 56%								
500	2.4	308.6	5	10	20	116.4	33	4.61
500	2.0	257.2	5	11	20	113.2	40	4.73
500	1.6	205.7	5	12	20	110.0	47	4.53
500	1.2	154.3	5	12	20	103.5	47	3.61
500	0.8	102.9	5	13	20	103.5	53	2.74
500	0.4	51.4	5	13	20	100.3	53	1.41
500	0.1	12.9	5	13	20	100.3	53	0.35
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 18.9°C, 56%								
900	4.1	527.2	5	8	20	236.2	20	2.35
900	3.45	443.6	5	8	20	226.5	20	2.06
900	2.8	360.1	5	10	20	221.7	33	2.83
900	2.15	276.5	5	12	20	216.9	47	3.08
900	1.5	192.9	5	12	20	216.9	47	2.15
900	0.85	109.3	5	13	20	212.1	53	1.42
900	0.2	25.7	5	14	20	207.3	60	0.38

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 19.9°C, 58%								
1300	5.6	720.1	5	9	20	406.6	27	2.48
1300	5.1	655.8	5	9	20	393.0	27	2.33
1300	4.6	591.5	5	9	20	386.2	27	2.14
1300	4.1	527.2	5	9	20	372.7	27	1.98
1300	3.6	462.9	5	9	20	372.7	27	1.74
1300	3.1	398.6	5	10	20	365.9	33	1.90
1300	2.6	334.3	5	11	20	365.9	40	1.90
1300	2.1	270.0	5	12	20	359.1	47	1.82
1300	1.6	205.7	5	13	20	359.1	53	1.58
1300	1.1	141.5	5	13	20	345.6	53	1.13
1300	0.6	77.2	5	14	20	338.8	60	0.70
1300	0.15	19.3	5	14	20	338.8	60	0.18
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 18.8°C, 54%								
1480	6.4	823.0	5	9	20	531.3	27	2.16
1480	5.4	694.4	5	9	20	523.6	27	1.85
1480	4.4	565.8	5	9	20	508.2	27	1.56
1480	3.4	437.2	5	9	20	500.5	27	1.22
1480	2.4	308.6	5	11	20	485.1	40	1.32
1480	1.4	180.0	5	12	20	477.4	47	0.91
1480	0.4	51.4	5	13	20	462.0	53	0.31
1480	0.15	19.3	5	15	20	454.3	67	0.15
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 19.1°C, 55%								
500	2.4	308.6	0	8	25	103.5	32	8.36
500	2.0	257.2	0	9	25	100.3	36	8.07
500	1.6	205.7	0	10	25	97.0	40	7.38
500	1.2	154.3	0	11	25	93.8	44	6.28
500	0.8	102.9	0	12	25	93.8	48	4.55
500	0.4	51.4	0	13	25	90.6	52	2.54
500	0.1	12.9	0	14	25	90.6	56	0.68
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 18.4°C, 59%								
900	4.1	527.2	0	7	25	231.4	28	5.61
900	3.45	443.6	0	7	25	221.7	28	4.93
900	2.8	360.1	0	8	25	216.9	32	4.66
900	2.15	276.5	0	9	25	212.1	36	4.10
900	1.5	192.9	0	10	25	207.3	40	3.24
900	0.85	109.3	0	12	25	207.3	48	2.19
900	0.2	25.7	0	13	25	207.3	52	0.56
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 18.9°C, 63%								
1300	5.6	720.1	0	8	25	399.8	32	5.05
1300	4.6	591.5	0	8	25	393.0	32	4.22
1300	3.6	462.9	0	9	25	386.2	36	3.77
1300	2.6	334.3	0	10	25	372.7	40	3.12
1300	1.6	205.7	0	11	25	365.9	44	2.15
1300	0.6	77.2	0	12	25	352.4	48	0.91
1300	0.15	19.3	0	13	25	345.6	52	0.25

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 18.9°C, 63%								
500	2.4	308.6	5	11	25	97.0	30	6.62
500	2.0	257.2	5	12	25	93.8	35	6.63
500	1.6	205.7	5	12	25	90.6	35	5.50
500	1.2	154.3	5	13	25	90.6	40	4.69
500	0.8	102.9	5	14	25	87.3	45	3.64
500	0.4	51.4	5	15	25	87.3	50	2.01
500	0.1	12.9	5	15	25	84.1	50	0.52
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 19.7°C, 34%								
900	4.1	527.2	5	9	25	226.5	20	3.25
900	3.45	443.6	5	9	25	216.9	20	2.86
900	2.8	360.1	5	12	25	212.1	35	4.11
900	2.15	276.5	5	13	25	207.3	40	3.67
900	1.5	192.9	5	14	25	207.3	45	2.87
900	0.85	109.3	5	14	25	202.4	45	1.67
900	0.2	25.7	5	15	25	197.6	50	0.44
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 19.7°C, 34%								
1300	5.6	720.1	5	9	25	399.8	20	2.52
1300	4.6	591.5	5	9	25	393.0	20	2.10
1300	3.6	462.9	5	10	25	379.5	25	2.12
1300	2.6	334.3	5	12	25	372.7	35	2.17
1300	1.6	205.7	5	14	25	365.9	45	1.74
1300	0.6	77.2	5	15	25	352.4	50	0.75
1300	0.15	19.3	5	15	25	345.6	50	0.19
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 20°C, 36%								
500	2.4	308.6	0	9	30	103.5	30	9.38
500	2.0	257.2	0	11	30	100.3	37	9.79
500	1.6	205.7	0	12	30	97.0	40	8.79
500	1.2	154.3	0	13	30	93.8	43	7.37
500	0.8	102.9	0	15	30	93.8	50	5.63
500	0.4	51.4	0	15	30	90.6	50	2.91
500	0.1	12.9	0	15	30	90.6	50	0.73
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 19.3°C, 42%								
900	4.1	527.2	0	7	30	221.7	23	5.86
900	3.45	443.6	0	7	30	216.9	23	5.04
900	2.8	360.1	0	9	30	212.1	30	5.34
900	2.15	276.5	0	10	30	207.3	33	4.64
900	1.5	192.9	0	12	30	202.4	40	3.95
900	0.85	109.3	0	14	30	197.6	47	2.66
900	0.2	25.7	0	14	30	197.6	47	0.63
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 19.3°C, 42%								
1300	5.6	720.1	0	6	30	399.8	20	3.82
1300	4.6	591.5	0	6	30	393.0	20	3.19
1300	3.6	462.9	0	7	30	386.2	23	2.95
1300	2.6	334.3	0	9	30	372.7	30	2.82
1300	1.6	205.7	0	12	30	359.1	40	2.38
1300	0.6	77.2	0	15	30	359.1	50	1.10
1300	0.15	19.3	0	16	30	352.4	53	0.30

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 21.1°C, 33%								
500	2.4	308.6	5	14	30	100.3	36	9.51
500	2.0	257.2	5	14	30	97.0	36	8.19
500	1.6	205.7	5	15	30	97.0	40	7.25
500	1.2	154.3	5	15	30	97.0	40	5.44
500	0.8	102.9	5	15	30	93.8	40	3.75
500	0.4	51.4	5	16	30	93.8	44	2.06
500	0.1	12.9	5	17	30	93.8	48	0.56
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 21.1°C, 33%								
900	4.1	527.2	5	9	30	231.4	16	3.18
900	3.45	443.6	5	10	30	226.5	20	3.41
900	2.8	360.1	5	13	30	221.7	32	4.47
900	2.15	276.5	5	15	30	212.1	40	4.46
900	1.5	192.9	5	15	30	207.3	40	3.18
900	0.85	109.3	5	16	30	202.4	44	2.02
900	0.2	25.7	5	17	30	202.4	48	0.52
9-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 21.1°C, 33%								
1300	5.6	720.1	5	10	30	399.8	20	3.14
1300	4.6	591.5	5	10	30	393.0	20	2.62
1300	3.6	462.9	5	11	30	386.2	24	2.49
1300	2.6	334.3	5	14	30	372.7	36	2.77
1300	1.6	205.7	5	15	30	365.9	40	1.92
1300	0.6	77.2	5	17	30	352.4	48	0.89
1300	0.15	19.3	5	18	30	345.6	52	0.25
9-pipe, wire fin, 20% fill: Lab Temperature & Relative Humidity: 20°C, 52%								
500	2.0	257.2	0	8	20	142.3	40	5.07
500	1.6	205.7	0	8	20	142.3	40	4.06
500	1.2	154.3	0	10	20	139.1	50	3.86
500	0.8	102.9	0	10	20	135.8	50	2.64
500	0.4	51.4	0	11	20	132.6	55	1.48
500	0.1	12.9	0	11	20	132.6	55	0.37
900	4.1	527.2	0	7	20	250.7	35	5.18
900	3.45	443.6	0	7	20	245.8	35	4.45
900	2.8	360.1	0	7	20	241.0	35	3.68
900	2.15	276.5	0	8	20	236.2	40	3.28
900	1.5	192.9	0	8	20	231.4	40	2.34
900	0.85	109.3	0	9	20	231.4	45	1.49
900	0.2	25.7	0	11	20	231.4	55	0.42
1300	5.6	720.1	0	7	20	399.8	35	4.44
1300	4.6	591.5	0	7	20	393.0	35	3.71
1300	3.6	462.9	0	7	20	386.2	35	2.95
1300	2.6	334.3	0	7	20	379.5	35	2.17
1300	1.6	205.7	0	8	20	372.7	40	1.55
1300	0.6	77.2	0	10	20	365.9	50	0.73
1300	0.15	19.3	0	10	20	365.9	50	0.18

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, wire fin, 20% fill: Lab Temperature & Relative Humidity: 22.3°C, 60%								
500	2.0	257.2	0	9	25	129.4	36	6.25
500	1.6	205.7	0	10	25	129.4	40	5.54
500	1.2	154.3	0	10	25	126.1	40	4.26
500	0.8	102.9	0	11	25	126.1	44	3.11
500	0.4	51.4	0	12	25	122.9	48	1.74
500	0.1	12.9	0	12	25	122.9	48	0.43
900	4.1	527.2	0	8	25	236.2	32	6.26
900	3.45	443.6	0	9	25	231.4	36	6.03
900	2.8	360.1	0	9	25	231.4	36	4.89
900	2.15	276.5	0	9	25	226.5	36	3.84
900	1.5	192.9	0	10	25	221.7	40	3.03
900	0.85	109.3	0	11	25	221.7	44	1.88
900	0.2	25.7	0	12	25	216.9	48	0.49
1300	5.6	720.1	0	8	25	393.0	32	5.14
1300	4.6	591.5	0	8	25	379.5	32	4.37
1300	3.6	462.9	0	9	25	372.7	36	3.91
1300	2.6	334.3	0	9	25	365.9	36	2.87
1300	1.6	205.7	0	10	25	359.1	40	1.99
1300	0.6	77.2	0	11	25	359.1	44	0.82
1300	0.15	19.3	0	12	25	352.4	48	0.23
9-pipe, wire fin, 20% fill: Lab Temperature & Relative Humidity: 22°C, 58%								
500	2.0	257.2	0	11	30	122.9	37	7.98
500	1.6	205.7	0	11	30	122.9	37	6.39
500	1.2	154.3	0	12	30	119.7	40	5.35
500	0.8	102.9	0	13	30	116.4	43	3.96
500	0.4	51.4	0	15	30	116.4	50	2.27
500	0.1	12.9	0	15	30	116.4	50	0.57
900	4.1	527.2	0	11	30	231.4	37	8.69
900	3.45	443.6	0	11	30	231.4	37	7.32
900	2.8	360.1	0	12	30	226.5	40	6.59
900	2.15	276.5	0	12	30	221.7	40	5.17
900	1.5	192.9	0	12	30	221.7	40	3.61
900	0.85	109.3	0	13	30	216.9	43	2.26
900	0.2	25.7	0	13	30	216.9	43	0.53
1300	5.6	720.1	0	9	30	393.0	30	5.76
1300	4.6	591.5	0	10	30	379.5	33	5.43
1300	3.6	462.9	0	11	30	372.7	37	4.74
1300	2.6	334.3	0	11	30	365.9	37	3.49
1300	1.6	205.7	0	12	30	359.1	40	2.38
1300	0.6	77.2	0	14	30	352.4	47	1.05
1300	0.15	19.3	0	14	30	352.4	47	0.26

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, wire fin, 20% fill: Lab Temperature & Relative Humidity: 20°C, 42%								
500	2.0	257.2	5	10	20	132.6	33	3.38
500	1.6	205.7	5	11	20	129.4	40	3.31
500	1.2	154.3	5	11	20	126.1	40	2.55
500	0.8	102.9	5	12	20	122.9	47	2.03
500	0.4	51.4	5	12	20	122.9	47	1.01
500	0.1	12.9	5	13	20	119.7	53	0.30
900	4.1	527.2	5	9	20	241.0	27	3.06
900	3.45	443.6	5	10	20	236.2	33	3.27
900	2.8	360.1	5	11	20	231.4	40	3.24
900	2.15	276.5	5	11	20	226.5	40	2.54
900	1.5	192.9	5	12	20	221.7	47	2.10
900	0.85	109.3	5	12	20	216.9	47	1.22
900	0.2	25.7	5	12	20	216.9	47	0.29
1300	5.6	720.1	5	9	20	386.2	27	2.61
1300	4.6	591.5	5	9	20	379.5	27	2.18
1300	3.6	462.9	5	9	20	372.7	27	1.74
1300	2.6	334.3	5	10	20	359.1	33	1.62
1300	1.6	205.7	5	11	20	359.1	40	1.19
1300	0.6	77.2	5	12	20	352.4	47	0.53
1300	0.15	19.3	5	13	20	352.4	53	0.15
9-pipe, wire fin, 20% fill: Lab Temperature & Relative Humidity: 22.7°C, 53%								
500	2.0	257.2	5	11	25	132.6	30	4.04
500	1.6	205.7	5	12	25	129.4	35	3.85
500	1.2	154.3	5	12	25	129.4	35	2.89
500	0.8	102.9	5	13	25	126.1	40	2.25
500	0.4	51.4	5	14	25	122.9	45	1.29
500	0.1	12.9	5	14	25	122.9	45	0.32
9-pipe, wire fin, 20% fill: Lab Temperature & Relative Humidity: 22.4°C, 57%								
900	4.1	527.2	5	10	25	236.2	25	3.89
900	3.45	443.6	5	11	25	231.4	30	3.99
900	2.8	360.1	5	13	25	226.5	40	4.38
900	2.15	276.5	5	13	25	221.7	40	3.43
900	1.5	192.9	5	13	25	221.7	40	2.40
900	0.85	109.3	5	15	25	216.9	50	1.72
900	0.2	25.7	5	15	25	216.9	50	0.41
9-pipe, wire fin, 20% fill: Lab Temperature & Relative Humidity: 20°C, 44%								
1300	5.6	720.1	5	10	25	386.2	25	3.25
1300	4.6	591.5	5	10	25	379.5	25	2.71
1300	3.6	462.9	5	11	25	372.7	30	2.58
1300	2.6	334.3	5	12	25	365.9	35	2.21
1300	1.6	205.7	5	14	25	365.9	45	1.74
1300	0.6	77.2	5	14	25	359.1	45	0.66
1300	0.15	19.3	5	15	25	352.4	50	0.19

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, wire fin, 20% fill: Lab Temperature & Relative Humidity: 20.6°C, 45%								
500	2.0	257.2	5	14	30	132.6	36	5.99
500	1.6	205.7	5	15	30	129.4	40	5.44
500	1.2	154.3	5	16	30	126.1	44	4.58
500	0.8	102.9	5	17	30	122.9	48	3.41
500	0.4	51.4	5	17	30	119.7	48	1.75
500	0.1	12.9	5	18	30	119.7	52	0.47
9-pipe, wire fin, 20% fill: Lab Temperature & Relative Humidity: 21.7°C, 41%								
900	4.1	527.2	5	10	30	241.0	20	3.81
900	3.45	443.6	5	12	30	236.2	28	4.54
900	2.8	360.1	5	13	30	231.4	32	4.29
900	2.15	276.5	5	15	30	226.5	40	4.17
900	1.5	192.9	5	15	30	216.9	40	3.04
900	0.85	109.3	5	17	30	212.1	48	2.10
900	0.2	25.7	5	18	30	212.1	52	0.53
1300	5.6	720.1	5	10	30	386.2	20	3.25
1300	4.6	591.5	5	11	30	379.5	24	3.24
1300	3.6	462.9	5	12	30	372.7	28	3.00
1300	2.6	334.3	5	13	30	365.9	32	2.52
1300	1.6	205.7	5	15	30	359.1	40	1.96
1300	0.6	77.2	5	16	30	352.4	44	0.82
1300	0.15	19.3	5	16	30	345.6	44	0.21
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 18.5°C, 68%								
500	2.0	257.2	0	6	20	139.1	30	3.92
500	1.7	218.6	0	7	20	135.8	35	3.96
500	1.4	180.0	0	7	20	132.6	35	3.34
500	1.1	141.5	0	8	20	129.4	40	3.07
500	0.8	102.9	0	8	20	126.1	40	2.29
500	0.5	64.3	0	9	20	126.1	45	1.60
500	0.2	25.7	0	10	20	122.9	50	0.73
900	3.5	450.1	0	5	20	236.2	25	3.38
900	2.8	360.1	0	5	20	231.4	25	2.76
900	2.1	270.0	0	7	20	226.5	35	2.94
900	1.4	180.0	0	7	20	226.5	35	1.96
900	0.7	90.0	0	8	20	221.7	40	1.14
900	0.2	25.7	0	9	20	221.7	45	0.36
1300	5.5	707.3	0	6	20	372.7	30	4.02
1300	4.4	565.8	0	6	20	359.1	30	3.34
1300	3.3	424.4	0	6	20	352.4	30	2.55
1300	2.2	282.9	0	7	20	345.6	35	2.02
1300	1.1	141.5	0	8	20	338.8	40	1.17
1300	0.2	25.7	0	9	20	338.8	45	0.24

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 20.3°C, 71%								
500	1.7	218.6	0	8	25	119.7	32	5.12
500	1.4	180.0	0	9	25	119.7	36	4.73
500	1.1	141.5	0	9	25	119.7	36	3.72
500	0.8	102.9	0	10	25	116.4	40	3.08
500	0.5	64.3	0	10	25	116.4	40	1.92
500	0.2	25.7	0	11	25	116.4	44	0.84
900	3.5	450.1	0	7	25	231.4	28	4.79
900	2.8	360.1	0	7	25	226.5	28	3.91
900	2.1	270.0	0	8	25	221.7	32	3.42
900	1.4	180.0	0	8	25	221.7	32	2.28
900	0.7	90.0	0	10	25	216.9	40	1.44
900	0.2	25.7	0	11	25	216.9	44	0.45
1300	5.0	643.0	0	6	25	365.9	24	3.72
1300	4.4	565.8	0	6	25	359.1	24	3.34
1300	3.3	424.4	0	7	25	352.4	28	2.97
1300	2.2	282.9	0	7	25	345.6	28	2.02
1300	1.1	141.5	0	9	25	345.6	36	1.29
1300	0.2	25.7	0	10	25	338.8	40	0.26
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 21.2°C, 56%								
500	1.7	218.6	0	9	30	116.4	30	5.90
500	1.4	180.0	0	10	30	113.2	33	5.54
500	1.1	141.5	0	11	30	113.2	37	4.77
500	0.8	102.9	0	12	30	110.0	40	3.88
500	0.5	64.3	0	12	30	110.0	40	2.42
500	0.2	25.7	0	13	30	110.0	43	1.05
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 21.2°C, 59%								
900	3.2	411.5	0	8	30	221.7	27	5.21
900	2.8	360.1	0	9	30	216.9	30	5.22
900	2.1	270.0	0	9	30	216.9	30	3.91
900	1.4	180.0	0	11	30	207.3	37	3.31
900	0.7	90.0	0	11	30	207.3	37	1.66
900	0.2	25.7	0	13	30	207.3	43	0.56
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 20.6°C, 65%								
1300	5.0	643.0	0	9	30	365.9	30	5.52
1300	4.4	565.8	0	9	30	359.1	30	4.95
1300	3.3	424.4	0	9	30	352.4	30	3.79
1300	2.2	282.9	0	10	30	352.4	33	2.79
1300	1.1	141.5	0	11	30	345.6	37	1.56
1300	0.2	25.7	0	11	30	338.8	37	0.29

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 17°C, 52%								
500	1.7	218.6	5	8	20	122.9	20	1.87
500	1.4	180.0	5	10	20	119.7	33	2.62
500	1.1	141.5	5	10	20	116.4	33	2.11
500	0.8	102.9	5	11	20	116.4	40	1.84
500	0.5	64.3	5	11	20	113.2	40	1.18
500	0.2	25.7	5	11	20	113.2	40	0.47
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 17.1°C, 52%								
900	3.5	450.1	5	8	20	221.7	20	2.14
900	2.8	360.1	5	9	20	216.9	27	2.32
900	2.1	270.0	5	9	20	216.9	27	1.74
900	1.4	180.0	5	10	20	212.1	33	1.48
900	0.7	90.0	5	11	20	207.3	40	0.90
900	0.2	25.7	5	12	20	207.3	47	0.30
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 15.8°C, 56%								
1300	5.5	707.3	5	8	20	379.5	20	1.96
1300	4.4	565.8	5	8	20	372.7	20	1.60
1300	3.3	424.4	5	9	20	359.1	27	1.65
1300	2.2	282.9	5	9	20	352.4	27	1.12
1300	1.1	141.5	5	11	20	345.6	40	0.85
1300	0.2	25.7	5	12	20	338.8	47	0.18
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 21°C, 43%								
500	1.7	218.6	5	12	25	113.2	35	4.67
500	1.4	180.0	5	12	25	113.2	35	3.85
500	1.1	141.5	5	12	25	110.0	35	3.11
500	0.8	102.9	5	13	25	110.0	40	2.58
500	0.5	64.3	5	14	25	110.0	45	1.81
500	0.2	25.7	5	14	25	110.0	45	0.72
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 19.7°C, 48%								
900	3.2	411.5	5	10	25	221.7	25	3.23
900	2.8	360.1	5	11	25	216.9	30	3.45
900	2.1	270.0	5	11	25	212.1	30	2.65
900	1.4	180.0	5	12	25	212.1	35	2.05
900	0.7	90.0	5	13	25	207.3	40	1.20
900	0.2	25.7	5	14	25	207.3	45	0.38
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 20.5°C, 43%								
1300	5.5	707.3	5	9	25	372.7	20	2.65
1300	4.4	565.8	5	9	25	365.9	20	2.16
1300	3.3	424.4	5	10	25	359.1	25	2.06
1300	2.2	282.9	5	11	25	352.4	30	1.67
1300	1.1	141.5	5	13	25	345.6	40	1.13
1300	0.2	25.7	5	13	25	338.8	40	0.21

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 21.2°C, 56%								
500	1.7	218.6	5	11	30	116.4	24	3.91
500	1.4	180.0	5	13	30	116.4	32	4.26
500	1.1	141.5	5	13	30	113.2	32	3.44
500	0.8	102.9	5	14	30	113.2	36	2.81
500	0.5	64.3	5	14	30	110.0	36	1.81
500	0.2	25.7	5	15	30	110.0	40	0.80
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 20°C, 54%								
900	3.2	411.5	5	11	30	221.7	24	3.86
900	2.8	360.1	5	12	30	216.9	28	4.02
900	2.1	270.0	5	12	30	216.9	28	3.01
900	1.4	180.0	5	13	30	212.1	32	2.34
900	0.7	90.0	5	14	30	207.3	36	1.34
900	0.2	25.7	5	14	30	207.3	36	0.38
9-pipe, plate fin, 10% fill, no taper: Lab Temperature & Relative Humidity: 19.5°C, 62%								
1300	5.5	707.3	5	10	30	365.9	20	3.36
1300	4.4	565.8	5	10	30	365.9	20	2.69
1300	3.3	424.4	5	11	30	352.4	24	2.51
1300	2.2	282.9	5	12	30	352.4	28	1.94
1300	1.1	141.5	5	14	30	338.8	36	1.29
1300	0.2	25.7	5	14	30	338.8	36	0.23
9-pipe, plate fin, 10% fill, 1.2° taper: Lab Temperature & Relative Humidity: 13.5°C, 40%								
500	2.0	257.2	0	4	20	177.9	20	2.06
500	1.7	218.6	0	6	20	161.7	30	2.86
500	1.4	180.0	0	8	20	145.5	40	3.47
500	1.1	141.5	0	8	20	155.2	40	2.56
500	0.8	102.9	0	10	20	152.0	50	2.36
500	0.5	64.3	0	10	20	145.5	50	1.54
500	0.2	25.7	0	11	20	145.5	55	0.67
900	3.5	450.1	0	5	20	178.3	25	4.47
900	2.8	360.1	0	6	20	173.5	30	4.40
900	2.1	270.0	0	6	20	168.7	30	3.39
900	1.4	180.0	0	7	20	168.7	35	2.63
900	0.7	90.0	0	9	20	168.7	45	1.68
900	0.2	25.7	0	10	20	163.9	50	0.55
9-pipe, plate fin, 10% fill, 1.2° taper: Lab Temperature & Relative Humidity: 16.2°C, 45%								
1300	5.5	707.3	0	3	20	237.2	15	3.19
1300	4.4	565.8	0	4	20	237.2	20	3.39
1300	3.3	424.4	0	5	20	230.4	25	3.26
1300	2.2	282.9	0	6	20	223.6	30	2.68
1300	1.1	141.5	0	7	20	223.6	35	1.56
1300	0.2	25.7	0	9	20	216.8	45	0.37

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, plate fin, 10% fill, 1.2° taper: Lab Temperature & Relative Humidity: 16°C, 48%								
500	2.0	257.2	0	8	25	122.9	32	5.87
500	1.7	218.6	0	10	25	113.2	40	6.72
500	1.4	180.0	0	10	25	110.0	40	5.70
500	1.1	141.5	0	10	25	110.0	40	4.48
500	0.8	102.9	0	11	25	106.7	44	3.68
500	0.5	64.3	0	12	25	106.7	48	2.50
500	0.2	25.7	0	12	25	103.5	48	1.03
900	3.5	450.1	0	8	25	173.5	32	7.28
900	2.8	360.1	0	8	25	168.7	32	5.99
900	2.1	270.0	0	9	25	168.7	36	5.03
900	1.4	180.0	0	10	25	163.9	40	3.82
900	0.7	90.0	0	11	25	163.9	44	2.10
900	0.2	25.7	0	13	25	159.1	52	0.72
1300	5.0	643.0	0	4	25	243.9	16	3.75
1300	4.4	565.8	0	5	25	237.2	20	4.23
1300	3.3	424.4	0	6	25	230.4	24	3.90
1300	2.2	282.9	0	8	25	223.6	32	3.55
1300	1.1	141.5	0	10	25	216.8	40	2.27
1300	0.2	25.7	0	11	25	216.8	44	0.45
500	2.0	257.2	0	11	30	110.0	37	8.92
500	1.7	218.6	0	11	30	110.0	37	7.59
500	1.4	180.0	0	11	30	106.7	37	6.44
500	1.1	141.5	0	12	30	110.0	40	5.33
500	0.8	102.9	0	13	30	106.7	43	4.32
500	0.5	64.3	0	15	30	106.7	50	3.09
500	0.2	25.7	0	15	30	106.7	50	1.24
900	3.2	411.5	0	8	30	178.3	27	6.47
900	2.8	360.1	0	9	30	173.5	30	6.52
900	2.1	270.0	0	10	30	173.5	33	5.42
900	1.4	180.0	0	11	30	168.7	37	4.07
900	0.7	90.0	0	13	30	163.9	43	2.46
900	0.2	25.7	0	14	30	159.1	47	0.78
1300	5.0	643.0	0	7	30	237.2	23	6.68
1300	4.4	565.8	0	7	30	230.4	23	6.05
1300	3.3	424.4	0	9	30	223.6	30	5.97
1300	2.2	282.9	0	10	30	223.6	33	4.40
1300	1.1	141.5	0	11	30	216.8	37	2.49
1300	0.2	25.7	0	12	30	216.8	40	0.49
9-pipe, plate fin, 10% fill, 1.2° taper: Lab Temperature & Relative Humidity: 16°C, 63%								
500	1.7	218.6	5	9	20	168.2	27	1.82
500	1.4	180.0	5	10	20	161.7	33	1.94
500	1.1	141.5	5	11	20	158.5	40	1.86
500	0.8	102.9	5	12	20	158.5	47	1.57
500	0.5	64.3	5	13	20	158.5	53	1.12
500	0.2	25.7	5	14	20	158.5	60	0.50

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, plate fin, 10% fill, 1.2° taper: Lab Temperature & Relative Humidity: 14°C, 65%								
900	3.5	450.1	5	8	20	173.5	20	2.73
900	2.8	360.1	5	9	20	178.3	27	2.82
900	2.1	270.0	5	10	20	173.5	33	2.71
900	1.4	180.0	5	11	20	173.5	40	2.16
900	0.7	90.0	5	12	20	168.7	47	1.29
900	0.2	25.7	5	12	20	163.9	47	0.38
1300	5.5	707.3	5	9	20	237.2	27	4.17
1300	4.4	565.8	5	10	20	230.4	33	4.27
1300	3.3	424.4	5	10	20	223.6	33	3.30
1300	2.2	282.9	5	10	20	223.6	33	2.20
1300	1.1	141.5	5	11	20	216.8	40	1.36
1300	0.2	25.7	5	11	20	216.8	40	0.25
9-pipe, plate fin, 10% fill, 1.2° taper: Lab Temperature & Relative Humidity: 16°C, 63%								
500	1.7	218.6	5	11	25	110.0	30	4.14
500	1.4	180.0	5	13	25	106.7	40	4.65
500	1.1	141.5	5	14	25	100.3	45	4.36
500	0.8	102.9	5	13	25	100.3	40	2.83
500	0.5	64.3	5	14	25	100.3	45	1.98
500	0.2	25.7	5	15	25	100.3	50	0.88
900	3.2	411.5	5	10	25	173.5	25	4.13
900	2.8	360.1	5	10	25	173.5	25	3.61
900	2.1	270.0	5	11	25	168.7	30	3.33
900	1.4	180.0	5	12	25	168.7	35	2.58
900	0.7	90.0	5	13	25	163.9	40	1.51
900	0.2	25.7	5	14	25	159.1	45	0.50
9-pipe, plate fin, 10% fill, 1.2° taper: Lab Temperature & Relative Humidity: 17.9°C, 40%								
1300	5.0	643.0	5	9	25	230.4	20	3.90
1300	4.4	565.8	5	10	25	237.2	25	4.15
1300	3.3	424.4	5	10	25	230.4	25	3.21
1300	2.2	282.9	5	11	25	216.8	30	2.72
1300	1.1	141.5	5	13	25	223.6	40	1.74
1300	0.2	25.7	5	13	25	223.6	40	0.32
9-pipe, plate fin, 10% fill, 1.2° taper: Lab Temperature & Relative Humidity: 14°C, 41%								
500	1.7	218.6	5	15	30	84.1	40	8.89
500	1.4	180.0	5	15	30	84.1	40	7.32
500	1.1	141.5	5	15	30	87.3	40	5.54
500	0.8	102.9	5	16	30	87.3	44	4.41
500	0.5	64.3	5	17	30	87.3	48	3.00
500	0.2	25.7	5	19	30	87.3	56	1.39
900	3.2	411.5	5	12	30	168.7	28	5.90
900	2.8	360.1	5	12	30	173.5	28	5.02
900	2.1	270.0	5	13	30	173.5	32	4.29
900	1.4	180.0	5	14	30	168.7	36	3.30
900	0.7	90.0	5	16	30	168.7	44	2.00
900	0.2	25.7	5	17	30	163.9	48	0.64

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
9-pipe, plate fin, 10% fill, 1.2° taper: Lab Temperature & Relative Humidity: 14°C, 41%								
1300	5.5	707.3	5	10	30	230.4	20	5.34
1300	4.4	565.8	5	10	30	230.4	20	4.27
1300	3.3	424.4	5	11	30	223.6	24	3.95
1300	2.2	282.9	5	12	30	216.8	28	3.16
1300	1.1	141.5	5	13	30	216.8	32	1.80
1300	0.2	25.7	5	14	30	210.1	36	0.38
24-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 18.6°C, 66%								
500	2.7	347.2	0	9	20	93.8	45	11.64
500	2.2	282.9	0	9	20	90.6	45	9.82
500	1.7	218.6	0	9	20	84.1	45	8.17
500	1.2	154.3	0	10	20	84.1	50	6.39
500	0.7	90.0	0	10	20	80.9	50	3.88
500	0.2	25.7	0	11	20	80.9	55	1.21
500	0.1	12.9	0	11	20	80.9	55	0.61
900	4.7	604.4	0	9	20	236.2	45	8.05
900	3.8	488.6	0	9	20	226.5	45	6.78
900	2.9	372.9	0	10	20	216.9	50	5.98
900	2.0	257.2	0	10	20	207.3	50	4.32
900	1.1	141.5	0	11	20	207.3	55	2.60
900	0.2	25.7	0	12	20	202.4	60	0.53
24-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 19.3°C, 68%								
1300	6.2	797.3	0	9	20	447.2	45	5.61
1300	5.2	668.7	0	9	20	440.4	45	4.77
1300	4.2	540.1	0	9	20	433.7	45	3.92
1300	3.2	411.5	0	9	20	420.1	45	3.08
1300	2.2	282.9	0	10	20	406.6	50	2.42
1300	1.2	154.3	0	11	20	399.8	55	1.47
1300	0.2	25.7	0	11	20	393.0	55	0.25
24-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 21°C, 55%								
500	2.7	347.2	5	10	20	74.4	33	8.12
500	2.2	282.9	5	12	20	71.1	47	9.62
500	1.7	218.6	5	12	20	67.9	47	7.79
500	1.2	154.3	5	13	20	67.9	53	6.26
500	0.7	90.0	5	13	20	64.7	53	3.83
500	0.2	25.7	5	14	20	61.4	60	1.29
900	4.7	604.4	5	10	20	221.7	33	4.74
900	3.8	488.6	5	10	20	212.1	33	4.01
900	2.9	372.9	5	11	20	202.4	40	3.83
900	2.0	257.2	5	12	20	197.6	47	3.15
900	1.1	141.5	5	13	20	197.6	53	1.97
900	0.2	25.7	5	14	20	192.8	60	0.41

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
24-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 21°C, 55%								
1300	6.2	797.3	5	11	20	433.7	40	3.83
1300	5.2	668.7	5	12	20	420.1	47	3.85
1300	4.2	540.1	5	12	20	413.3	47	3.16
1300	3.2	411.5	5	12	20	399.8	47	2.49
1300	2.2	282.9	5	12	20	393.0	47	1.74
1300	1.2	154.3	5	13	20	386.2	53	1.10
1300	0.2	25.7	5	14	20	379.5	60	0.21
24-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 20.6°C, 72%								
500	2.7	347.2	0	11	25	74.4	44	17.81
500	2.2	282.9	0	11	25	71.1	44	15.17
500	1.7	218.6	0	12	25	67.9	48	13.35
500	1.2	154.3	0	13	25	64.7	52	10.68
500	0.7	90.0	0	13	25	64.7	52	6.23
500	0.2	25.7	0	14	25	61.4	56	2.01
24-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 23°C, 33%								
900	4.7	604.4	0	10	25	207.3	40	10.15
900	3.8	488.6	0	11	25	202.4	44	9.21
900	2.9	372.9	0	11	25	197.6	44	7.20
900	2.0	257.2	0	12	25	192.8	48	5.53
900	1.1	141.5	0	13	25	188.0	52	3.37
900	0.2	25.7	0	14	25	183.2	56	0.67
24-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 21°C, 44%								
1300	6.2	797.3	0	10	25	426.9	40	6.50
1300	5.2	668.7	0	10	25	413.3	40	5.63
1300	4.2	540.1	0	10	25	406.6	40	4.62
1300	3.2	411.5	0	10	25	406.6	40	3.52
1300	2.2	282.9	0	11	25	399.8	44	2.70
1300	1.2	154.3	0	12	25	393.0	48	1.63
1300	0.2	25.7	0	14	25	386.2	56	0.32
24-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 22.4°C, 65%								
500	2.7	347.2	5	13	25	67.9	40	14.08
500	2.2	282.9	5	14	25	64.7	45	13.51
500	1.7	218.6	5	14	25	61.4	45	10.99
500	1.2	154.3	5	15	25	61.4	50	8.59
500	0.7	90.0	5	16	25	58.2	55	5.79
500	0.2	25.7	5	16	25	58.2	55	1.66
900	4.7	604.4	5	12	25	207.3	35	7.05
900	3.8	488.6	5	13	25	202.4	40	6.65
900	2.9	372.9	5	14	25	197.6	45	5.83
900	2.0	257.2	5	15	25	192.8	50	4.56
900	1.1	141.5	5	16	25	188.0	55	2.82
900	0.2	25.7	5	16	25	188.0	55	0.51

RPM	Outlet Velocity, (m/s)	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)	COP
24-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 22.4°C, 65%								
1300	6.2	797.3	5	13	25	426.9	40	5.14
1300	5.2	668.7	5	13	25	420.1	40	4.38
1300	4.2	540.1	5	14	25	413.3	45	4.03
1300	3.2	411.5	5	14	25	406.6	45	3.13
1300	2.2	282.9	5	15	25	399.8	50	2.42
1300	1.2	154.3	5	16	25	393.0	55	1.47
1300	0.2	25.7	5	17	25	386.2	60	0.27
24-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 23.3°C, 32%								
500	2.45	315.1	0	15	30	61.4	50	26.29
500	2.1	270.0	0	15	30	55.0	50	25.19
500	1.7	218.6	0	15	30	48.5	50	23.11
500	1.2	154.3	0	15	30	48.5	50	16.31
500	0.7	90.0	0	16	30	45.3	53	10.84
500	0.2	25.7	0	17	30	42.0	57	3.53
900	4.7	604.4	0	14	30	197.6	47	14.69
900	3.8	488.6	0	14	30	192.8	47	12.17
900	2.9	372.9	0	14	30	192.8	47	9.29
900	2.0	257.2	0	15	30	188.0	50	7.02
900	1.1	141.5	0	16	30	183.2	53	4.21
900	0.2	25.7	0	17	30	178.3	57	0.83
24-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 21°C, 46%								
1300	6.2	797.3	0	11	30	420.1	37	7.24
1300	5.2	668.7	0	12	30	413.3	40	6.71
1300	4.2	540.1	0	12	30	406.6	40	5.51
1300	3.2	411.5	0	12	30	399.8	40	4.27
1300	2.2	282.9	0	13	30	393.0	43	3.22
1300	1.2	154.3	0	15	30	393.0	50	2.01
1300	0.2	25.7	0	16	30	386.2	53	0.36

24-pipe, wire fin, 10% fill: Lab Temperature & Relative Humidity: 23°C, 65%								
500	2.45	315.1	5	16	30	61.4	44	19.21
500	2.1	270.0	5	16	30	58.2	44	17.38
500	1.7	218.6	5	17	30	58.2	48	15.30
500	1.2	154.3	5	17	30	55.0	48	11.43
500	0.7	90.0	5	18	30	55.0	52	7.20
500	0.2	25.7	5	19	30	51.7	56	2.35
900	4.7	604.4	5	15	30	207.3	40	9.97
900	3.8	488.6	5	16	30	202.4	44	9.04
900	2.9	372.9	5	17	30	192.8	48	7.88
900	2.0	257.2	5	17	30	192.8	48	5.43
900	1.1	141.5	5	17	30	188.0	48	3.06
900	0.2	25.7	5	18	30	183.2	52	0.62
1300	6.2	797.3	5	14	30	433.7	36	5.68
1300	5.2	668.7	5	15	30	426.9	40	5.36
1300	4.2	540.1	5	15	30	413.3	40	4.47
1300	3.2	411.5	5	15	30	406.6	40	3.46
1300	2.2	282.9	5	17	30	399.8	48	2.88
1300	1.2	154.3	5	18	30	393.0	52	1.73
1300	0.2	25.7	5	19	30	386.2	56	0.31

Fan Speed	Volume Flow Rate, (m3/hr)	Supply Air Inlet Temperature, T1, (°C)	Supply Air Outlet Temperature, T2, (°C)	Exhaust Air Inlet Temperature, T3, (°C)	Input Power (Watts)	Efficiency, (%)
Baxl WH300: Lab temperature & relative humidity: 18°C & 62%						
6	171	0	11	20	187	55
5	118	0	12	20	96	60
4	99	0	13	20	79	65
3	80	0	13	20	60	65
2	65	0	14	20	48	70
Baxl WH300: Lab temperature & relative humidity: 18°C & 53%						
6	171	0	14	25	187	56
5	118	0	15	25	96	60
4	99	0	16	25	79	64
3	80	0	16	25	60	64
2	65	0	16	25	48	64
Baxl WH300: Lab temperature & relative humidity: 18°C & 53%						
6	171	0	16	30	187	53
5	118	0	17	30	96	57
4	99	0	17	30	79	57
3	80	0	18	30	60	60
2	65	0	18	30	48	60
Baxl WH300: Lab temperature & relative humidity: 18.2°C & 62%						
6	171	5	13	20	187	53
5	118	5	14	20	96	60
4	99	5	14	20	79	60
3	80	5	15	20	60	67
2	65	5	15	20	48	67
Baxl WH300: Lab temperature & relative humidity: 17.2°C & 55%						
6	171	5	16	25	187	55
5	118	5	16	25	96	55
4	99	5	17	25	79	60
3	80	5	18	25	60	65
2	65	5	18	25	48	65
Baxl WH300: Lab temperature & relative humidity: 17.2°C & 55%						
6	171	5	18	30	187	52
5	118	5	19	30	96	56
4	99	5	19	30	79	56
3	80	5	20	30	60	60
2	65	5	20	30	48	60
Baxl WH300: Lab temperature & relative humidity: 18.2°C & 62%						
6	171	10	15	20	187	50
5	118	10	15	20	96	50
4	99	10	15	20	79	50
3	80	10	16	20	60	60
2	65	10	16	20	48	60
Baxl WH300: Lab temperature & relative humidity: 17.2°C & 55%						
6	171	10	18	25	187	53
5	118	10	18	25	96	53
4	99	10	19	25	79	60
3	80	10	19	25	60	60
2	65	10	19	25	48	60
Baxl WH300: Lab temperature & relative humidity: 17.2°C & 55%						
6	171	10	18	30	187	40
5	118	10	19	30	96	45
4	99	10	20	30	79	50
3	80	10	20	30	60	50
2	65	10	20	30	48	50

APPENDIX E

Fan Performance Test Results

RPM	Orifice Plate	Flow Rate (m3/hr)	Static Pressure (Pa)
9-pipe, wire fin			
200	OPEN	160	0.00
200	1	160	0.75
200	2	144	1.25
200	3	128	2.00
200	4	128	2.50
200	5	112	3.50
200	6	112	4.00
200	CLOSED	128	5.00
9-pipe, wire fin			
400	OPEN	320	1.00
400	1	288	4.50
400	2	240	7.50
400	3	192	10.50
400	4	144	12.00
400	5	128	15.00
400	6	176	16.00
400	CLOSED	160	16.50
9-pipe, wire fin			
600	OPEN	496	-1.00
600	1	416	6.50
600	2	352	13.50
600	3	272	19.00
600	4	192	23.50
600	5	176	30.00
600	6	224	33.00
600	CLOSED	45	34.50
9-pipe, wire fin			
800	OPEN	625	-2.50
800	1	529	12.00
800	2	464	24.50
800	3	336	35.00
800	4	256	43.50
800	5	208	52.50
800	6	288	58.00
800	CLOSED	256	60.50
9-pipe, wire fin			
1000	OPEN	721	-3.5
1000	1	657	18.5
1000	2	561	38.5
1000	3	416	57
1000	4	304	71.5
1000	5	256	86
1000	6	352	94
1000	CLOSED	304	99

RPM	Orifice Plate	Flow Rate (m3/hr)	Static Pressure (Pa)
9-pipe, wire fin			
1200	OPEN	977	-4.50
1200	1	817	29.00
1200	2	641	57.00
1200	3	480	81.00
1200	4	352	102.00
1200	5	288	126.00
1200	6	416	136.00
1200	CLOSED	368	142.00
9-pipe, wire fin			
1480	OPEN	1217	-6
1480	1	1041	40
1480	2	833	82
1480	3	641	122
1480	4	448	153
1480	5	288	188
1480	6	416	208
1480	CLOSED	272	217
9-pipe, plate fin			
200	OPEN	128	-0.1
200	1	112	0.8
200	2	96	1.7
200	3	72	2.3
200	4	48	2.9
200	5	48	3.8
200	6	64	4
200	CLOSED	64	4.2
9-pipe, plate fin			
400	OPEN	288	-0.1
400	1	256	2.8
400	2	224	5.6
400	3	176	8.5
400	4	128	10.7
400	5	128	13.1
400	6	128	14.4
400	CLOSED	80	14.8
9-pipe, plate fin			
600	OPEN	416	-0.5
600	1	352	5.4
600	2	272	11.3
600	3	224	17.1
600	4	176	22.2
600	5	160	26.6
600	6	144	29.8
600	CLOSED	0	31.4

RPM	Orifice Plate	Flow Rate (m3/hr)	Static Pressure (Pa)
	9-pipe, plate fin		
800	OPEN	544	-1.1
800	1	464	9.4
800	2	400	20.3
800	3	288	29.7
800	4	208	38.8
800	5	224	47.2
800	6	176	53.4
800	CLOSED	0	56.1
	9-pipe, plate fin		
1000	OPEN	720	-2
1000	1	608	14.2
1000	2	512	29.3
1000	3	368	43.9
1000	4	288	58.7
1000	5	272	72.8
1000	6	208	82.6
1000	CLOSED	0	86.5
	9-pipe, plate fin		
1200	OPEN	880	-2.7
1200	1	752	20.1
1200	2	624	40.9
1200	3	464	61.8
1200	4	384	83.5
1200	5	368	104.3
1200	6	256	118.3
1200	CLOSED	0	124.1
	9-pipe, plate fin		
1480	OPEN	1057	-4.2
1480	1	896	30.1
1480	2	704	61.9
1480	3	576	92.6
1480	4	400	125.7
1480	5	480	158.8
1480	6	304	181.7
1480	CLOSED	0	189.9

RPM	Orifice Plate	Flow Rate (m3/hr)	Static Pressure (Pa)
24-pipe, wire fin			
200	OPEN	160	0.0
200	1	160	0.7
200	2	128	1.7
200	3	96	2.5
200	4	96	3.2
200	5	96	3.8
200	6	96	4.1
200	CLOSED	80	3.9
24-pipe, wire fin			
400	OPEN	320	-0.1
400	1	288	3.6
400	2	240	6.7
400	3	208	9.6
400	4	160	11.6
400	5	128	13.5
400	6	160	14.5
400	CLOSED	144	15
24-pipe, wire fin			
600	OPEN	512	-0.7
600	1	400	7.4
600	2	352	14.2
600	3	240	20.4
600	4	208	25
600	5	176	29.8
600	6	208	32.2
600	CLOSED	192	32.4
24-pipe, wire fin			
800	OPEN	672	-1.6
800	1	544	13.2
800	2	480	25.3
800	3	352	37.1
800	4	256	45.5
800	5	224	53.6
800	6	256	58.4
800	CLOSED	240	61.1
24-pipe, wire fin			
1000	OPEN	816	-2.9
1000	1	688	20.6
1000	2	592	39.9
1000	3	464	59.3
1000	4	320	73.8
1000	5	256	84.8
1000	6	320	92
1000	CLOSED	272	95.7

RPM	Orifice Plate	Flow Rate (m3/hr)	Static Pressure (Pa)
24-pipe, wire fin			
1200	OPEN	992	-4.4
1200	1	848	29.9
1200	2	672	66.2
1200	3	528	97.1
1200	4	384	114.2
1200	5	320	123.7
1200	6	368	134.7
1200	CLOSED	304	139.9
24-pipe, wire fin			
1480	OPEN	1169	-8.2
1480	1	1057	50
1480	2	864	93
1480	3	640	135
1480	4	480	170
1480	5	304	195
1480	6	512	210
1480	CLOSED	352	210

Fan Speed	Orifice Plate	Flow Rate (m3/hr)	Static Pressure (Pa)
Baxl WH300			
6	OPEN	219	0
6	1	216	4
6	2	208	17
6	3	187	49
6	4	168	165
6	5	112	380
6	6	42	465
6	CLOSED	25	471
5	OPEN	133	0
5	1	131	2
5	2	128	7
5	3	119	23
5	4	114	81
5	5	70	208
5	6	32	265
5	CLOSED	24	273
4	OPEN	109	0
4	1	109	0
4	2	106	5
4	3	99	16
4	4	93	59
4	5	61	150
4	6	24	195
4	CLOSED	11	201
3	OPEN	81.6	0
3	1	81.6	0
3	2	80	3
3	3	76.8	10
3	4	73.6	38
3	5	43.2	100
3	6	20.8	130
3	CLOSED	5	138
2	OPEN	64	0
2	1	64	0
2	2	62.4	1.9
2	3	60.8	6
2	4	54.4	22
2	5	36.8	59
2	6	11.2	78
2	CLOSED	3	81
1	OPEN	48	0
1	1	48	0
1	2	48	0
1	3	46.4	3
1	4	40	11
1	5	25.6	29
1	6	9.6	40
1	CLOSED	0	45

APPENDIX F

Design Calculations and Data

Appendix F: Design Calculations and Data

Pipe Sizing:

Pressure drops were calculated using the Darcy-Weisbach equation:

$$\frac{\Delta P}{\rho} = \frac{2flv^2}{D} \qquad \textbf{(F.1)}$$

Where: ΔP = pressure drop (N/m^2)

ρ = density (kg/m^3)

f = friction factor

l = pipe length (m)

v = velocity (m/s)

D = pipe diameter (m)

$$f_{laminar} = \frac{16}{Re} \qquad \textbf{(F.2)}$$

$$f_{turbulent} = \frac{0.08}{(Re)^{1/4}} \qquad \textbf{(F.3)}$$

$$Re = \frac{\rho v D}{\mu} \qquad \textbf{(F.4)}$$

Where: μ = viscosity (Ns/m^2)

Mass flow rate for isobutane for 4kW heating = 0.0129kg/s

Suction line data:

$\rho_s = 4.2 kg/m^3$ (ρ_g at 0°C)

$\mu_s \approx 0.008$ centipoise

$\approx 8 \times 10^{-6}$ Ns/m^2

Discharge line data:

$\rho_d = 13.6 kg/m^3$ (ρ_g at 40°C)

$\mu_s \approx 8 \times 10^{-6}$ Ns/m^2

Calculation of cross-sectional area of tubes:

3/8" tube (20 gauge wall) for discharge line:

Bore = $d_{3/8} = 7.7 \times 10^{-3}$ m

$CSA_{3/8} = 4.7 \times 10^{-5}$ m^2

5/8" tube (18 gauge wall) for suction line:

Bore = $d_{5/8} = 14.05 \times 10^{-3}$ m

$CSA_{5/8} = 15.5 \times 10^{-5}$ m^2

Calculation of vapour velocities:

$V_{3/8}$ = mass flow rate / ($CSA_{3/8} \times \rho_d$)

$= 0.0129 / (4.7 \times 10^{-5} \times 13.6)$

$= 20.18$ m/s

$V_{5/8}$ = mass flow rate / ($CSA_{5/8} \times \rho_s$)

$= 0.0129 / (15.5 \times 10^{-5} \times 4.2)$

$= 19.8$ m/s

Calculation of Reynolds numbers from equation F.4:

$$Re_{3/8} = (13.6 \times 20.18 \times 7.7 \times 10^{-3}) / 8 \times 10^{-6}$$
$$= 264156$$

$$Re_{5/8} = (4.2 \times 19.8 \times 14.05 \times 10^{-3}) / 8 \times 10^{-6}$$
$$= 146050$$

Calculation of friction factors:

The Reynolds numbers are turbulent therefore friction factors are found using equation F.3.

$$f_{3/8} = 0.08 / (264156)^{1/4}$$
$$= 0.0035$$

$$f_{5/8} = 0.08 / (146050)^{1/4}$$
$$= 0.0041$$

Calculation of pressure drops from equation F.1:

Discharge:

$$\Delta P_{3/8} = \frac{2fv^2\rho_d}{D_{1/2}} \quad \text{(per metre run)}$$
$$= (2 \times 0.0035 \times 20.18^2 \times 13.6) / 7.7 \times 10^{-3}$$
$$= 5034.9 \text{ N/m}^2 \text{ per metre}$$

Suction:

$$\Delta P_{5/8} = \frac{2fv^2\rho_s}{D_{5/8}} \quad \text{(per metre run)}$$
$$= (2 \times 0.0041 \times 19.8^2 \times 4.2) / 14.05 \times 10^{-3}$$
$$= 961 \text{ N/m}^2 \text{ per metre}$$

Discharge line length = 2m straight pipe work plus 3 × 90° long radius bends.

Suction line length = 1.2m straight pipe work plus 3 × 90° long radius bends.

3/8" bend ≡ 0.27m straight pipe length.

5/8" bend ≡ 0.36m straight pipe length. [Trane,1965]

Therefore equivalent discharge line length $= 2 + 3 \times 0.27$

$= 2.81\text{m}$

Therefore equivalent suction line length $= 1.2 + 3 \times 0.36$

$= 2.28\text{m}$

Calculation of actual pressure drops in pipes:

Discharge (3/8"): $5034.9 \times 2.81 = 14148\text{N/m}^2$

$= \mathbf{0.14148bar}$

Suction (5/8"): $961 \times 2.28 = 2191\text{N/m}^2$

$= \mathbf{0.0219bar}$

Approximate Calculation of Refrigerant Charge:

Discharge line:

3/8" pipe, 2m long, CSA = $7.13 \times 10^{-5}\text{m}^2$, $\rho_{g40°C} = 13.55\text{kg/m}^3$

Volume = $2 \times 7.13 \times 10^{-5}$

$= 1.43 \times 10^{-4}\text{m}^3$

Mass $= 13.55 \times 1.43 \times 10^{-4}$

$= \mathbf{1.93 \times 10^{-3}kg}$

Oil separator:

300ml internal volume = 0.0003m^3, $\rho_{g40°C} = 13.55\text{kg/m}^3$

Mass $= 13.55 \times 0.0003$

$= \mathbf{4.065 \times 10^{-3}kg}$

Condenser:

9 pipes - Ø22mm, 300mm long, CSA = $3.8 \times 10^{-4}m^2$

Volume of condenser = $9 \times 3.8 \times 10^{-4} \times 0.3$

$= 0.001026m^3$

$\rho_{g40°C} = 13.55kg/m^3$

$\rho_{f0°C} = 525.2kg/m^3$

Assume in the condenser there is 85% vapour and 15% liquid.

Vapour mass = $0.85 \times 0.001026 \times 13.55$

$= 11.8 \times 10^{-3}kg$

Liquid mass = $0.15 \times 0.001026 \times 525.2$

$= 0.0808kg$

Combined mass in condenser = **0.0926kg**

Condenser header:

Diameter of pipe ring (pipe centre to pipe centre) = 0.252m

Length of pipe = $\pi \times 0.252$

$= 0.792m$

1/2" pipe, 0.792m long, CSA = $1.26 \times 10^{-4}m^2$

Volume = $0.792 \times 1.26 \times 10^{-4}$

$= 1.003 \times 10^{-4}m^3$

Assume a drop in temperature of 5°C, $\rho_{f35°C} = 531.35kg/m^3$

Mass = $1.003 \times 10^{-4} \times 531.35$

= 0.05329kg

Evaporator header:

Diameter of pipe ring (pipe centre to pipe centre) = 0.252m

Length of pipe = $\pi \times 0.252$

$= 0.792m$

5/8" pipe, 0.792m long, CSA = $1.979 \times 10^{-4}m^2$

Volume = $0.792 \times 1.979 \times 10^{-4}$

$= 1.567 \times 10^{-4}m^3$

Assume due to flash gas there is 10% vapour and 90% liquid in the evaporator header.

$\rho_{g0^\circ C} = 4.2337 kg/m^3$, $\rho_{f0^\circ C} = 582.4 kg/m^3$

Vapour mass $= 0.1 \times 1.567 \times 10^{-4} \times 4.2337$

$= 6.634 \times 10^{-5} kg$

Liquid mass $= 0.9 \times 1.567 \times 10^{-4} \times 582.4$

$= 0.08214 kg$

Combined mass in evaporator header = **0.0822kg**

Evaporator:

9 pipes - Ø22mm, 300mm long,

Volume of condenser = 0.001026m^3

$\rho_{g0^\circ C} = 4.2337 kg/m^3$, $\rho_{f0^\circ C} = 582.4 kg/m^3$

Assume in the evaporator there is 90% vapour and 10% liquid.

Vapour mass $= 0.9 \times 0.001026 \times 4.2337$

$= 3.9 \times 10^{-3} kg$

Liquid mass $= 0.1 \times 0.001026 \times 582.4$

$= 0.05975 kg$

Combined mass in evaporator = **0.0637kg**

Suction line:

5/8" pipe, 1.2m long, CSA = $1.979 \times 10^{-4} m^2$, $\rho_{g0^\circ C} = 4.2337 kg/m^3$

Volume = $1.2 \times 1.979 \times 10^{-4}$

$= 2.375 \times 10^{-4} m^3$

Mass $= 4.2337 \times 2.375 \times 10^{-4}$

$= 1.0 \times 10^{-3}$kg

Filter drier:

250ml internal volume = $0.00025\ m^3$, $\rho_{g0^\circ C} = 4.2337 kg/m^3$

Mass $= 4.2337 \times 0.00025$

$= 1.058 \times 10^{-3}$kg

Oil separator and pressure switch pipe work:

Discharge side – 1.1m, Ø1/4", CSA = $3.167 \times 10^{-5}m^2$, $\rho_{g40°C} = 13.55kg/m^3$

Mass = $1.1 \times 3.167 \times 10^{-5} \times 13.55$

= $4.72 \times 10^{-4}kg$

Suction side – 0.65m, Ø1/4", CSA = $3.167 \times 10^{-5}m^2$, $\rho_{g0°C} = 4.2337kg/m^3$

Mass = $0.65 \times 3.167 \times 10^{-5} \times 4.2337$

= $8.715 \times 10^{-5}kg$

Combined mass = $\mathbf{5.59 \times 10^{-4}kg}$

Lines between headers:

Discharge side – 0.13m, Ø1/4", CSA = $3.167 \times 10^{-5}m^2$,

$\rho_{f35°C} = 531.35kg/m^3$

Mass = $0.13 \times 3.167 \times 10^{-5} \times 531.35$

= $2.188 \times 10^{-3}kg$

Evaporator side – 0.13m, Ø3/8", CSA = $7.126 \times 10^{-5}m^2$,

$\rho_{f0°C} = 582.4kg/m^3$

Mass = $0.13 \times 7.126 \times 10^{-5} \times 582.4$

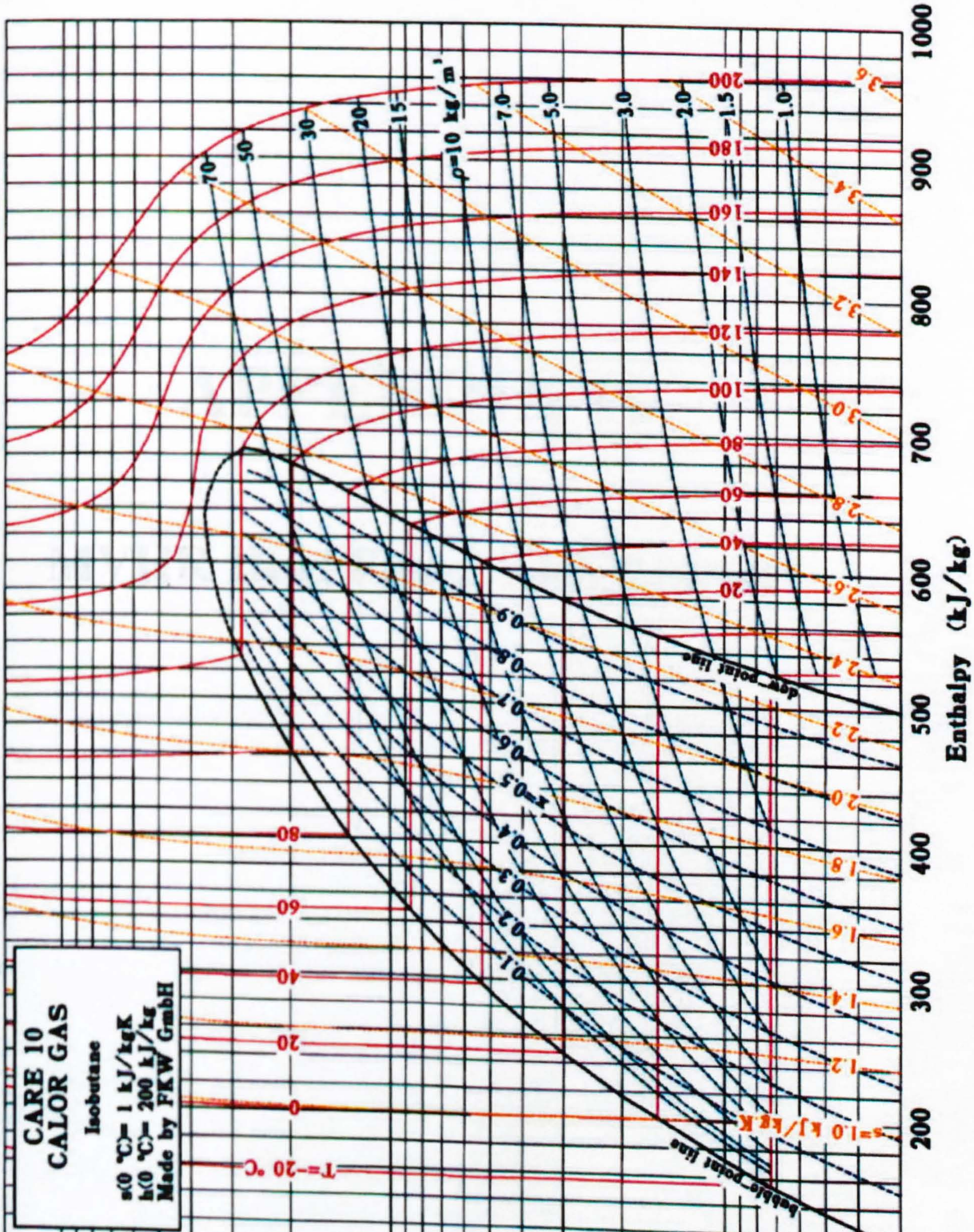
= $5.395 \times 10^{-3}kg$

Combined mass = $\mathbf{7.583 \times 10^{-3}kg}$

Therefore the total isobutane charge required for the system is the sum of the component masses = **0.308kg**

= **308g**

P-H Chart for CARE 10, Isobutane



APPENDIX G

MVHR Heat Pump with Ejector

Appendix G: MVHR Heat Pump with Ejector

Whilst mechanical compressors are fairly reliable, they do break down due to the wearing of moving parts. For this reason they require regular maintenance. One compressor that has no moving parts, and therefore requires very little or no maintenance and no oil lubrication, is the ejector compressor (or jet compressor). Ejectors are devices which use the kinetic energy of one vapour stream (the motive flow) to drive a second vapour stream (the suction flow) by direct mixing. Essentially, high pressure vapour (motive flow) is expanded in a nozzle to form a high speed jet at low pressure, which entrains vapour from the evaporator (suction flow). The combined streams from the mixing chamber are then diffused at an intermediate pressure (discharge flow) to the condenser. Figure G.1 illustrates an ejector compressor.

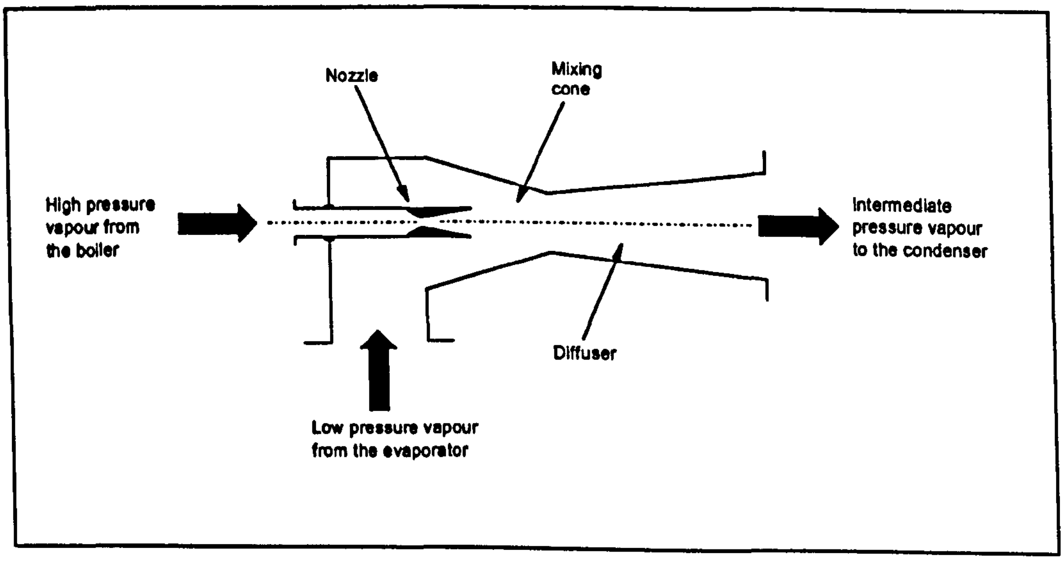


Figure G.1: Ejector Compressor

Jet compressor systems are driven by the heat input to the boiler and a feed pump is used to return a proportion of the condensate back to the boiler. These systems have low coefficients of performance (COP) but are economically viable if a large quantity of waste heat is available.

An ejector compressor could be incorporated into the prototype MVHR heat pump system to provide space heating/cooling and water heating. The system could be used in situations where there is a large waste heat source. Figure G.2 illustrates the system.

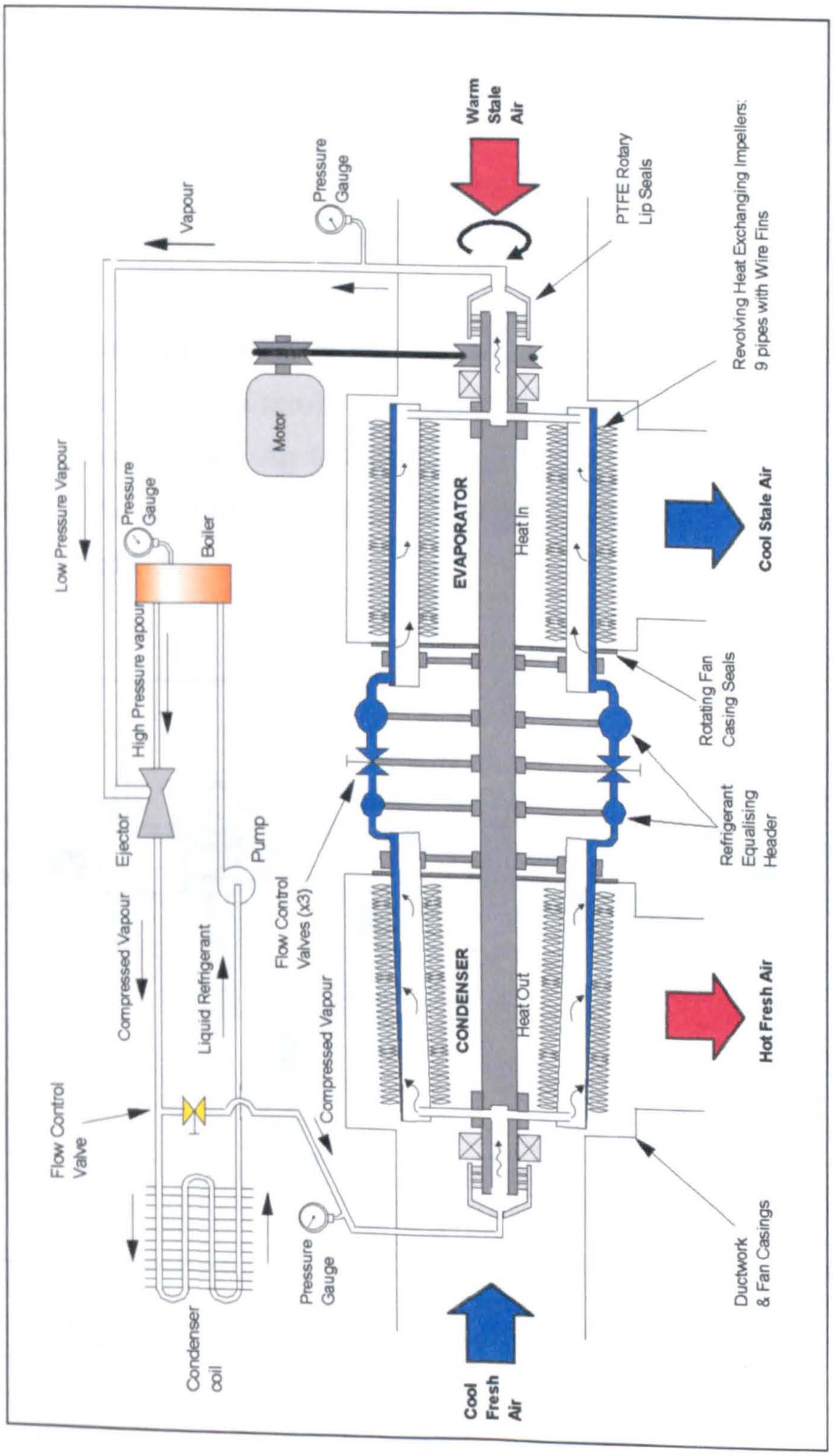


Figure G.2: MVHR Heat Pump with Ejector

Waste heat at around 100°C is used to create high pressure vapour in the boiler. The high pressure steam passing through the ejector creates a suction flow from the revolving evaporator where heat is taken in, producing a cooling effect. The vapour discharged from the ejector is split into two flows using a flow control valve. The majority of the vapour enters a static condenser coil where the heat could be used for water heating. The remainder of the vapour travels into the revolving condenser where it is used for warm air heating. The condensed refrigerant in the condenser coil is pumped back to the boiler using a piston pump, and the condensed refrigerant in the revolving condenser passes into the revolving evaporator pipes via the expansion valves.

A prototype was constructed to trial the concept. Existing components were used to reduce cost. As a result, the size of the system was constrained to fit the existing components. An isobutane fixed nozzle compressor supplied by Venturi Jet Pumps Ltd. was used. The ejector specification is shown in Figures G.3 and G.4.

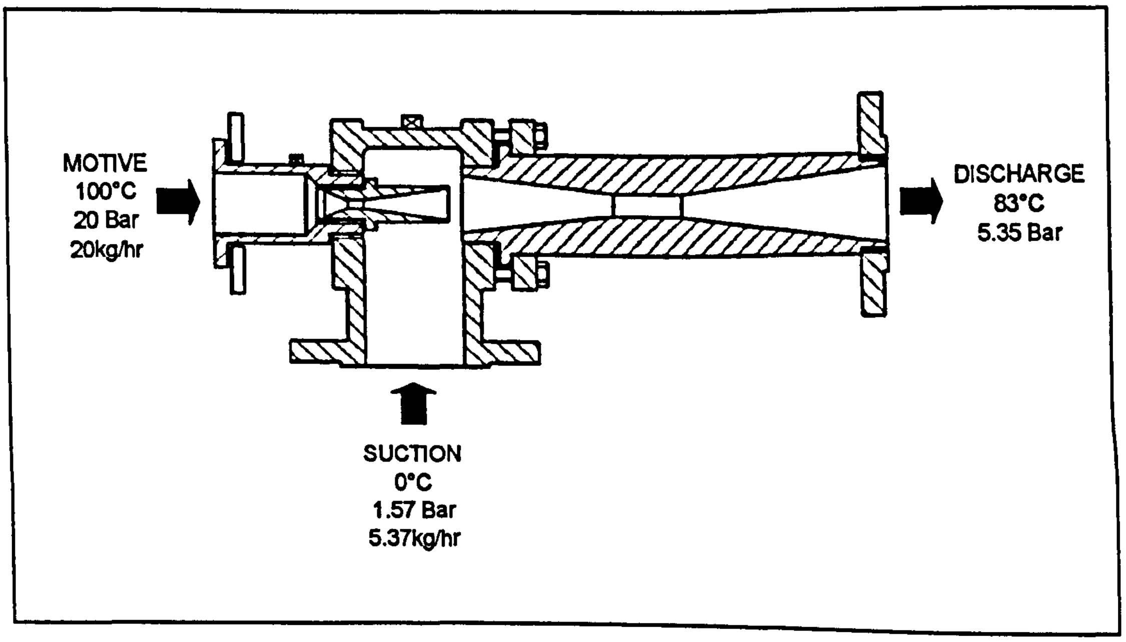


Figure G.3: Isobutane Ejector Operating Conditions

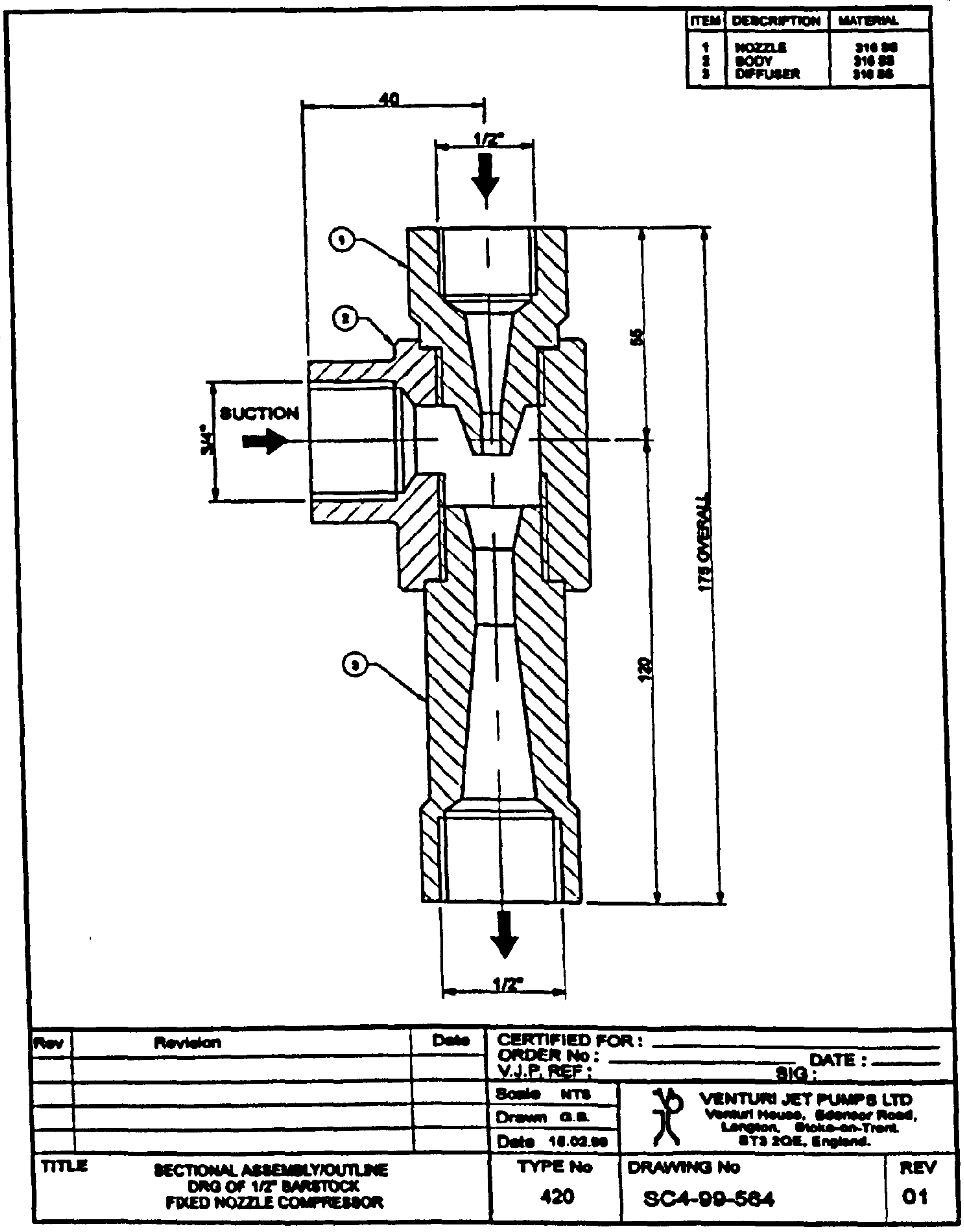
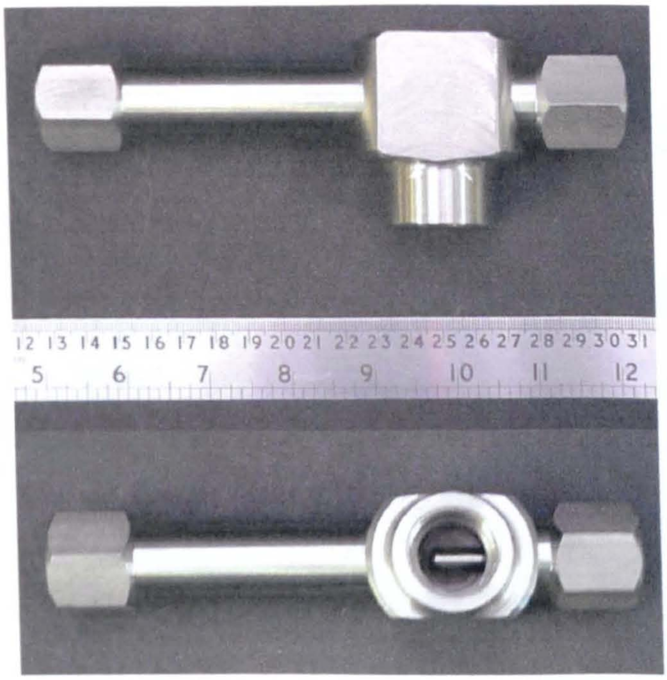
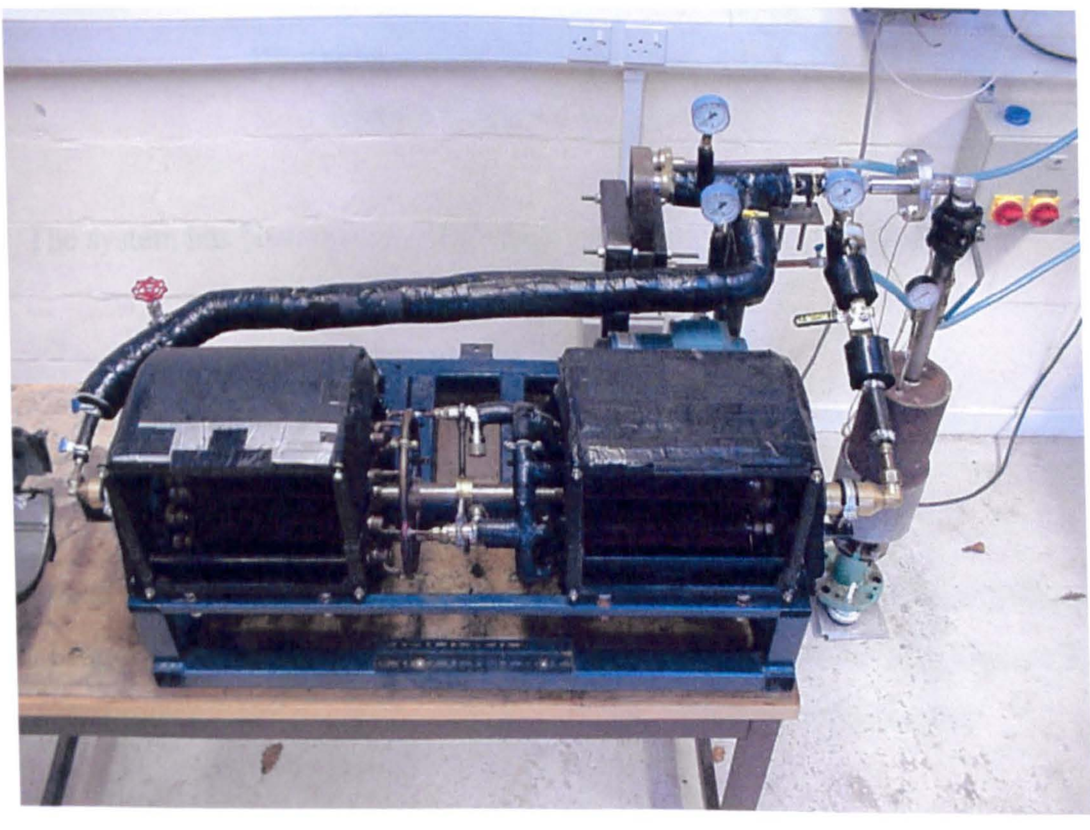


Figure G.4: Isobutane Ejector Specification

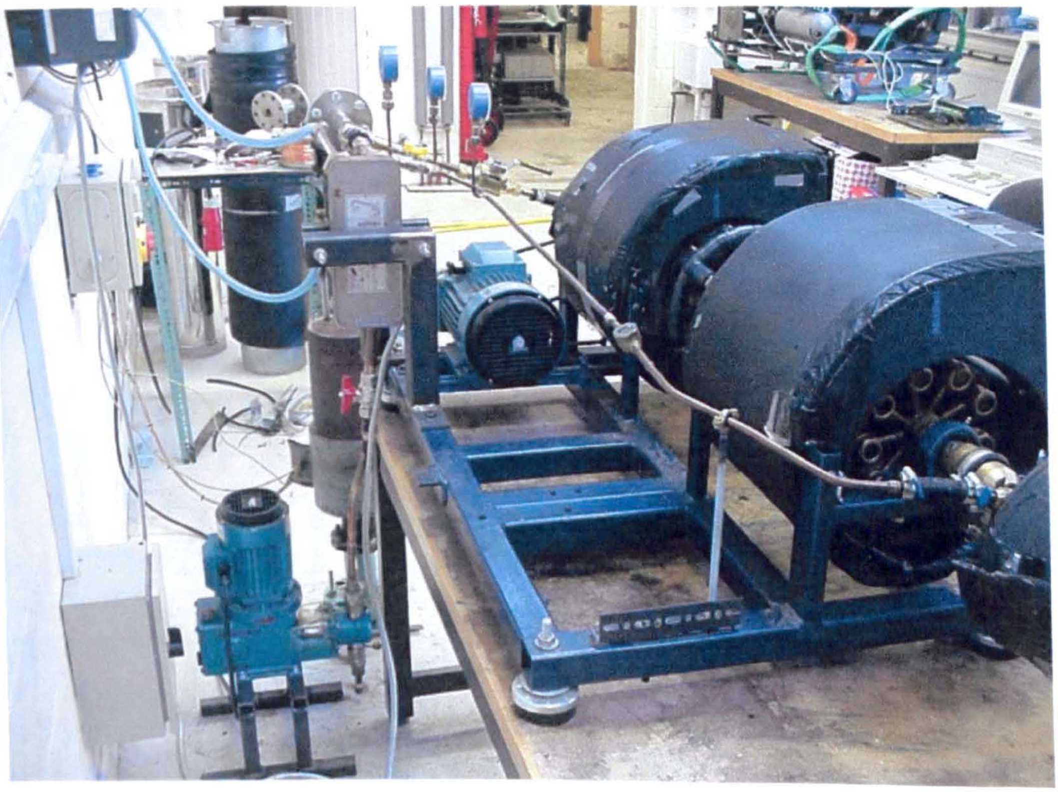
The ejector has a suction mass flow rate capable of providing approximately 500Watts of heating. A thermostatically controlled 6kW band heater is used to heat the boiler and a 127litres/hr (0.37kW) piston pump is used between the stationary condenser coil and the boiler. The stationary condenser coil was cooled using chilled water. Photograph G.1 shows the fixed nozzle compressor and Photographs G.2 and G.3 show the prototype system.



Photograph G.1: Isobutane Fixed Nozzle Compressor



Photograph G.2: MVHR Heat Pump with Ejector Compressor (front view)



Photograph G.3: MVHR Heat Pump with Ejector Compressor (rear view)

The system has been operated running under the following conditions:

Boiler: 100°C, 20Bar

Suction/evaporator: 1.0Bar

Revolving condenser: 3.5Bar

When running it was possible to feel the heating/cooling generated from the revolving pipes. Further testing is required using a heating/cooling load so that measurements can be taken to enable the calculation of a system COP.

APPENDIX H

Sample Results From MVHR Heat Pump Testing

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m^3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
16	44	50	28	-0.9	0.9	6/10	2175	700	2.0	257	0	77	4	20	39	18	0.291	0.146	0.366	2.5	0.8
16	44	100	56	-0.45	2.15	6/10	2175	700	2.0	257	0	38	17	20	47	13	0.287	0.381	1.484	3.9	2.2
16	44	150	84	-0.35	2.4	6/10	2175	700	2.0	257	0	33	18	20	63	9	0.283	0.462	1.566	3.4	2.1
16	44	200	112	-0.225	2.7	6/10	2175	700	2.0	257	0	31	20	20	71	7	0.287	0.526	1.727	3.3	2.1
16	44	250	140	-0.175	3.05	6/10	2175	700	2.0	257	0	29	22	20	74	7	0.283	0.535	1.887	3.5	2.3
16	44	300	169	-0.125	3.275	6/10	2175	700	2.0	257	0	26	23	20	77	6	0.283	0.574	1.965	3.4	2.3
16	44	350	197	-0.05	3.4	6/10	2175	700	2.0	257	0	25	24	20	78	6	0.287	0.605	2.043	3.4	2.3
16	44	400	225	0	3.55	6/10	2175	700	2.0	257	0	24	25	20	79	5	0.283	0.608	2.121	3.5	2.4
16	44	450	253	0.05	3.775	6/10	2175	700	2.0	257	0	23	26	20	82	5	0.283	0.616	2.198	3.6	2.4
16	44	500	281	0.075	3.875	6/10	2175	700	2.0	257	0	22	26	20	85	4	0.287	0.652	2.198	3.4	2.3
16	44	550	309	0.15	4.075	6/10	2175	700	2.0	257	0	20	28	20	85	4	0.306	0.680	2.350	3.5	2.4
16	44	575	323	0.2	4.25	6/10	2175	700	2.0	257	0	20	28	20	85	4	0.314	0.711	2.350	3.3	2.3

Varying refrigerant charge

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m^3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
16	44	550	309	0.2	3.8	2/10	2175	700	2.0	257	0	21	26	20	82	5	0.298	0.717	2.198	3.1	2.2
16	44	550	309	0.19	3.875	4/10	2175	700	2.0	257	0	23	26	20	77	6	0.302	0.708	2.198	3.1	2.2
16	44	550	309	0.175	3.8	6/10	2175	700	2.0	257	0	22	26	20	75	6	0.306	0.689	2.198	3.2	2.2
16	44	550	309	0.175	3.8	8/10	2175	700	2.0	257	0	21	26	20	75	6	0.302	0.694	2.198	3.2	2.2
16	44	550	309	0.2	3.6	10/10	2175	700	2.0	257	0	23	24	20	73	6	0.310	0.700	2.043	2.9	2.0

Varying valve setting

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m^3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
15.9	54	550	309	0.4	5.95	6/10	2175	300	0.5	64	0	17	36	20	75	6	0.328	0.820	0.734	0.9	0.6
15.9	54	550	309	0.525	7.1	6/10	2175	300	0.3	39	0	15	41	20	76	5	0.320	0.896	0.493	0.6	0.4
15.9	54	550	309	0.2	3.95	6/10	2175	500	1.5	193	0	21	25	20	81	6	0.272	0.711	1.591	2.2	1.6
15.9	54	550	309	0.125	4.2	6/10	2175	500	1.0	129	0	19	28	20	74	3	0.278	0.694	1.175	1.7	1.2
15.9	54	550	309	0.075	5.45	6/10	2175	500	0.5	64	0	15	37	20	69	1	0.278	0.683	0.752	1.1	0.8
15.9	54	550	309	0.125	3.3	6/10	2175	700	2.8	360	0	26	19	20	95	7	0.310	0.666	2.306	3.5	2.4
15.9	54	550	309	0.1	3.5	6/10	2175	700	1.6	206	0	25	22	20	90	4	0.302	0.658	1.509	2.3	1.6
15.9	54	550	309	0.3	7.65	6/10	2175	700	0.5	64	0	13	47	20	91	2	0.306	0.764	0.923	1.2	0.9
15.9	54	550	309	0.175	2.9	6/10	2175	900	4.0	514	0	24	14	20	81	8	0.362	0.672	2.471	3.7	2.4
15.9	54	550	309	0.1	3.5	6/10	2175	900	2.2	283	0	23	22	20	84	5	0.352	0.638	2.075	3.3	2.1
15.9	54	550	309	0.3	7.025	6/10	2175	900	0.5	64	0	10	47	20	94	4	0.337	0.717	0.923	1.3	0.9
15.9	54	550	309	0.125	2.7	6/10	2175	1100	5.0	643	0	29	13	20	85	9	0.427	0.582	2.878	4.9	2.9
15.9	54	550	309	0.125	3.25	6/10	2175	1100	2.5	321	0	26	21	20	90	6	0.422	0.580	2.259	3.9	2.3
15.9	54	550	309	0.225	6.525	6/10	2175	1100	0.5	64	0	9	46	20	91	3	0.427	0.666	0.906	1.4	0.8
15.9	54	550	309	0.175	2.7	6/10	2175	1300	6.0	772	0	34	14	20	78	10	0.549	0.566	3.706	6.6	3.3
15.9	54	550	309	0.1	2.925	6/10	2175	1300	3.0	386	0	32	18	20	84	7	0.535	0.563	2.349	4.2	2.1
15.9	54	550	309	0.25	6.35	6/10	2175	1300	0.5	64	0	9	44	20	93	4	0.535	0.655	0.873	1.3	0.7

Varying fan rpm

HEATING MODE TESTS

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
17.5	46	550	309	0.11	3	6/10	2175	700	3.0	386	0	33	18	20	67	8	0.306	0.560	2.349	4.2	2.7
17.5	46	550	309	0.09	3.1	6/10	2175	700	2.5	321	0	32	19	20	76	6	0.306	0.540	2.059	3.8	2.4
17.5	46	550	309	0.05	3.1	6/10	2175	700	2.0	257	0	31	20	20	79	5	0.306	0.512	1.727	3.4	2.1
17.5	46	550	309	0.02	3.4	6/10	2175	700	1.5	193	0	28	23	20	82	4	0.306	0.515	1.474	2.9	1.8
17.5	46	550	309	0	3.6	6/10	2175	700	1.0	129	0	26	26	20	78	2	0.302	0.507	1.099	2.2	1.4
17.5	46	550	309	0.04	5.5	6/10	2175	700	0.5	64	0	22	37	20	75	1	0.302	0.554	0.752	1.4	0.9

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
17.5	46	550	309	0.2	3.5	6/10	2175	700	3.0	386	0	26	21	25	75	10	0.306	0.585	2.711	4.6	3.0
17.5	46	550	309	0.2	3.5	6/10	2175	700	2.5	321	0	25	22	25	71	9	0.302	0.574	2.358	4.1	2.7
17.5	46	550	309	0.2	3.8	6/10	2175	700	2.0	257	0	25	24	25	73	8	0.298	0.580	2.043	3.5	2.3
17.5	46	550	309	0.1	3.8	6/10	2175	700	1.5	193	0	23	26	25	75	5	0.302	0.552	1.648	3.0	1.9
17.5	46	550	309	0.1	4.3	6/10	2175	700	1.0	129	0	21	30	25	77	4	0.302	0.540	1.250	2.3	1.5
17.5	46	550	309	0.1	5.5	6/10	2175	700	0.5	64	0	18	39	25	78	3	0.302	0.585	0.787	1.3	0.9

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
17	78	550	309	0.4	4.4	6/10	2175	700	3.0	386	0	28	25	30	70	13	0.302	0.720	3.181	4.4	3.1
17	78	550	309	0.35	4.3	6/10	2175	700	2.5	321	0	24	25	30	73	11	0.291	0.703	2.651	3.8	2.7
17	78	550	309	0.3	4.5	6/10	2175	700	2.0	257	0	22	27	30	78	8	0.291	0.686	2.274	3.3	2.3
17	78	550	309	0.3	4.8	6/10	2175	700	1.5	193	0	21	31	30	81	7	0.291	0.666	1.931	2.9	2.0
17	78	550	309	0.24	5.6	6/10	2175	700	1.0	129	0	18	36	30	82	5	0.291	0.666	1.469	2.2	1.5
17	78	550	309	0.3	7.4	6/10	2175	700	0.5	64	0	14	46	30	82	5	0.287	0.700	0.906	1.3	0.9

HEATING MODE TESTS

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
17	78	550	309	0.41	4.6	6/10	2175	700	3.0	386	0	20	26	35	77	15	0.283	0.680	3.297	4.8	3.4
17	78	550	309	0.4	4.7	6/10	2175	700	2.5	321	0	19	27	35	83	13	0.283	0.672	2.843	4.2	3.0
17	78	550	309	0.4	4.9	6/10	2175	700	2.0	257	0	18	30	35	92	11	0.279	0.669	2.500	3.7	2.6
17	78	550	309	0.35	5.3	6/10	2175	700	1.5	193	0	16	34	35	97	8	0.291	0.672	2.096	3.1	2.2
17	78	550	309	0.28	6.2	6/10	2175	700	1.0	129	0	13	40	35	93	5	0.283	0.655	1.610	2.5	1.7
17	78	550	309	0.3	8.1	6/10	2175	700	0.5	64	0	13	50	35	88	5	0.279	0.728	0.972	1.3	1.0

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
17.5	46	550	309	0.14	3.5	6/10	2175	700	3.0	386	5	31	22	20	74	8	0.298	0.574	2.187	3.8	2.5
17.5	46	550	309	0.04	3.5	6/10	2175	700	2.5	321	5	31	23	20	78	5	0.302	0.552	1.923	3.5	2.3
17.5	46	550	309	0.04	3.6	6/10	2175	700	2.0	257	5	30	24	20	83	4	0.306	0.529	1.618	3.1	1.9
17.5	46	550	309	0.04	4	6/10	2175	700	1.5	193	5	28	28	20	85	3	0.302	0.529	1.441	2.7	1.7
17.5	46	550	309	0.01	4.3	6/10	2175	700	1.0	129	5	23	32	20	82	2	0.310	0.526	1.117	2.1	1.3
17.5	46	550	309	0.13	6.1	6/10	2175	700	0.5	64	5	19	42	20	86	2	0.302	0.602	0.739	1.2	0.8

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
17.5	46	550	309	0.3	4.2	6/10	2175	700	3.0	386	5	29	25	25	54	12	0.302	0.630	2.545	4.0	2.7
17.5	46	550	309	0.25	4.2	6/10	2175	700	2.5	321	5	28	26	25	62	11	0.298	0.602	2.219	3.7	2.5
17.5	46	550	309	0.15	4.2	6/10	2175	700	2.0	257	5	29	27	25	71	7	0.298	0.566	1.853	3.3	2.1
17.5	46	550	309	0.12	4.4	6/10	2175	700	1.5	193	5	24	30	25	78	5	0.302	0.568	1.563	2.7	1.8
17.5	46	550	309	0.11	4.9	6/10	2175	700	1.0	129	5	20	34	25	78	4	0.298	0.588	1.192	2.0	1.3
17.5	46	550	309	0.11	6.4	6/10	2175	700	0.5	64	5	17	43	25	77	3	0.298	0.610	0.757	1.2	0.8

HEATING MODE TESTS

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
17.5	46	550	309	0.2	3.2	6/10	2175	700	3 0	386	-5	26	18	20	61	10	0.310	0 644	3.001	4.7	3.1
17.5	46	550	309	0.1	2.8	6/10	2175	700	2.5	321	-5	24	19	20	69	7	0.302	0 596	2 600	4.4	2.9
17.5	46	550	309	0.05	2.7	6/10	2175	700	2.0	257	-5	24	20	20	72	5	0.298	0.552	2.159	3 9	2.5
17 5	46	550	309	0.03	3.2	6/10	2175	700	1.5	193	-5	22	22	20	76	4	0.298	0.568	1.736	3.1	2.0
17.5	46	550	309	0 01	3.5	6/10	2175	700	1.0	129	-5	20	26	20	77	2	0.298	0 546	1.310	2.4	1.6
17.5	46	550	309	0.01	5	6/10	2175	700	0 5	64	-5	18	35	20	75	2	0.295	0.582	0.819	1.4	0.9

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
17.5	46	550	309	0.2	3.2	6/10	2175	700	3.0	386	-5	23	18	25	71	11	0.302	0.610	3.001	4 9	3.3
17.5	46	550	309	0.18	3.1	6/10	2175	700	2.5	321	-5	23	19	25	69	9	0.298	0.596	2.600	4 4	2.9
17.5	46	550	309	0.11	3.2	6/10	2175	700	2.0	257	-5	22	19	25	74	7	0.295	0.568	2.080	3.7	2.4
17.5	46	550	309	0.08	3.4	6/10	2175	700	1.5	193	-5	21	22	25	75	4	0.295	0.521	1.736	3.3	2.1
17.5	46	550	309	0.08	3 6	6/10	2175	700	1.0	129	-5	19	26	25	79	4	0.291	0.538	1.310	2 4	1.6
17.5	46	550	309	0.1	5 4	6/10	2175	700	0.5	64	-5	16	36	25	75	2	0.287	0.574	0.836	1.5	1.0

COOLING MODE TESTS

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Cooling (kW)	Cooling COP	System COP
20	42	550	309	0.55	5.75	6/10	2175	700	3.0	386	25	45	16	20	21	36	0.310	0.798	1.183	1.5	1.1
20	42	550	309	0.55	5 8	6/10	2175	700	2.5	321	25	54	14	20	18	38	0.306	0.784	1.213	1.5	1.1
20	42	550	309	0.45	5.7	6/10	2175	700	2.0	257	25	64	12	20	16	39	0.302	0.773	1.155	1.5	1.1
20	42	550	309	0.45	6	6/10	2175	700	1.5	193	25	71	11	20	15	42	0.302	0.770	0.937	1.2	0.9
20	42	550	309	0.4	6.4	6/10	2175	700	1.0	129	25	72	9	20	13	45	0.298	0.778	0.719	0 9	0.7
20	42	550	309	0.45	7.9	6/10	2175	700	0.5	64	25	74	8	20	12	51	0.295	0.795	0.383	0.5	0.4

COOLING MODE TESTS

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m³/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Cooling (kW)	Cooling COP	System COP
20	42	550	309	0.5	6	6/10	2175	700	3.0	386	30	44	18	20	20	38	0.298	0.882	1.566	1.8	1.3
20	42	550	309	0.5	6	6/10	2175	700	2.5	321	30	43	17	20	17	39	0.298	0.848	1.419	1.7	1.2
20	42	550	309	0.55	6.3	6/10	2175	700	2.0	257	30	50	14	20	16	41	0.295	0.829	1.412	1.7	1.3
20	42	550	309	0.45	6.3	6/10	2175	700	1.5	193	30	59	12	20	14	43	0.291	0.795	1.200	1.5	1.1
20	42	550	309	0.45	6.5	6/10	2175	700	1.0	129	30	64	10	20	13	46	0.287	0.790	0.895	1.1	0.8
20	42	550	309	0.5	8.5	6/10	2175	700	0.5	64	30	70	9	20	11	53	0.287	0.865	0.472	0.5	0.4

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m³/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Cooling (kW)	Cooling COP	System COP
20	70	550	309	0.5	7.1	6/10	2175	700	3.0	386	35	64	20	20	30	40	0.283	0.762	1.943	2.6	1.9
20	70	550	309	0.5	7.2	6/10	2175	700	2.5	321	35	68	17	20	27	41	0.279	0.762	1.964	2.6	1.9
20	70	550	309	0.5	7.5	6/10	2175	700	2.0	257	35	78	16	20	22	45	0.283	0.773	1.665	2.2	1.6
20	70	550	309	0.6	8.3	6/10	2175	700	1.5	193	35	86	13	20	17	51	0.279	0.784	1.461	1.9	1.4
20	70	550	309	0.55	8.4	6/10	2175	700	1.0	129	35	86	11	20	14	52	0.271	0.795	1.070	1.3	1.0
20	70	550	309	0.65	9.7	6/10	2175	700	0.5	64	35	87	10	20	12	58	0.271	0.798	0.560	0.7	0.5

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m³/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Cooling (kW)	Cooling COP	System COP
20	70	550	309	0.4	6.7	6/10	2175	700	3.0	386	40	46	25	20	28	38	0.283	0.770	1.909	2.5	1.8
20	70	550	309	0.47	7	6/10	2175	700	2.5	321	40	49	23	20	25	41	0.283	0.795	1.816	2.3	1.7
20	70	550	309	0.5	7.6	6/10	2175	700	2.0	257	40	58	20	20	23	44	0.279	0.818	1.727	2.1	1.6
20	70	550	309	0.61	8.5	6/10	2175	700	1.5	193	40	70	16	20	18	51	0.275	0.882	1.577	1.8	1.4
20	70	550	309	0.6	8.8	6/10	2175	700	1.0	129	40	72	13	20	14	55	0.287	0.862	1.196	1.4	1.0
20	70	550	309	0.62	10	6/10	2175	700	0.5	64	40	74	12	20	12	60	0.283	0.846	0.622	0.7	0.6

VARYING SUPPLY AIR INLET TEMPERATURE

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
19 8	58	550	309	0.01	3.7	6/10	2175	700	2.0	257	0	25	25	20	93	8	0.283	0.588	2.121	3.6	2.4
19 8	58	550	309	0.04	4.5	6/10	2175	700	2.0	257	5	28	29	20	96	7	0.283	0 622	2.007	3.2	2.2
19.8	58	550	309	0.08	5.3	6/10	2175	700	2.0	257	10	26	35	20	98	7	0.283	0 664	2.047	3.1	2.2
19 8	58	550	309	0.12	6	6/10	2175	700	2.0	257	15	25	40	20	98	7	0.271	0 692	2.012	2.9	2.1
19.8	58	550	309	0.15	6.7	6/10	2175	700	2.0	257	20	24	46	20	98	8	0 267	0.725	2.049	2.8	2.1

COLD EXHAUST AIR INLET

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
18.7	70	550	309	-0.15	2.8	6/10	2175	700	3.0	386	0	36	16	5	79	2	0.295	0.490	2.103	4.3	2.7
18.7	70	550	309	-0.15	3	6/10	2175	700	2.5	321	0	34	18	5	83	0	0.291	0.493	1.957	4.0	2.5
18.7	70	550	309	-0.13	3.1	6/10	2175	700	2.0	257	0	33	20	5	80	0*	0.291	0.498	1.727	3.5	2.2
18 7	70	550	309	-0.17	3.3	6/10	2175	700	1.5	193	0	31	23	5	74	-2	0.291	0.476	1.474	3.1	1.9
18 7	70	550	309	-0.2	3 4	6/10	2175	700	1.0	129	0	29	24	5	70	-3	0.283	0.442	1.022	2.3	1.4
18.7	70	550	309	-0.2	4.5	6/10	2175	700	0.5	257	0	28	30	5	63	-3	0.275	0.442	2.499	5.6	3.5

*Frosting of evaporator starts to occur

Lab temp (°C)	Lab RH (%)	Isobutane (R600a) Volume charge (ml)	Isobutane (R600a) Mass charge (grams)	Isobutane (R600a) Suction Pressure (bar, g)	Isobutane (R600a) Discharge Pressure (bar, g)	Valve setting	Compressor speed (rpm)	Fan Speed (rpm)	Outlet velocity (m/s)	Volume flow rate (m3/hr)	Supply Inlet Temperature (°C)	Supply Outlet RH (%)	Supply Outlet Temperature (°C)	Exhaust Inlet Temperature (°C)	Exhaust Outlet RH (%)	Exhaust Outlet Temperature (°C)	Fan power (kW)	Compressor power (kW)	Heating (kW)	Heating COP	System COP
18.7	70	550	309	-0.05	3.4	6/10	2175	700	3.0	386	0	32	18	10	98**	4	0.279	0.470	2.349	5.0	3.1
18.7	70	550	309	-0.1	3.5	6/10	2175	700	2.5	321	0	31	19	10	98**	4	0.279	0.470	2.059	4.4	2.7
18.7	70	550	309	-0.04	3.5	6/10	2175	700	2.0	257	0	31	21	10	98**	4	0.271	0.484	1.807	3.7	2.4
18.7	70	550	309	-0.04	4	6/10	2175	700	1.5	193	0	30	24	10	98**	2	0.271	0.487	1.533	3.1	2.0
18.7	70	550	309	-0.01	4.4	6/10	2175	700	1.0	129	0	24	30	10	98**	0	0.271	0.504	1.250	2.5	1.6
18.7	70	550	309	-0.02	5	6/10	2175	700	0.5	257	0	21	34	10	98**	0	0.271	0.504	2.792	5.5	3.6

*Frosting of evaporator starts to occur

**RH sensor got wet and not working

Pressure in the system when not operating is 1.8bar when the room temperature is 20°C

APPENDIX I

Further Design Ideas

Appendix I: Further Design Ideas

Air could be dried and cleaned using the system shown in Figure I.1. Absorbent solution (e.g. Lithium Bromide, Potassium Formate) is sprayed onto the evaporator sections of the revolving heat pipes. The solution covers all of the pipes surfaces, whilst the rotation ensures that the solution and air are fully mixed. Water vapour in the air is absorbed by the solution, resulting in a reduction of relative humidity. The absorbent solution also scrubs the air clean. The heat pipes transfer the heat generated due to absorption, to the cooler air passing over the condenser sections of the pipes. The water absorbed by the absorbent is driven off as steam from the generator using a heater.

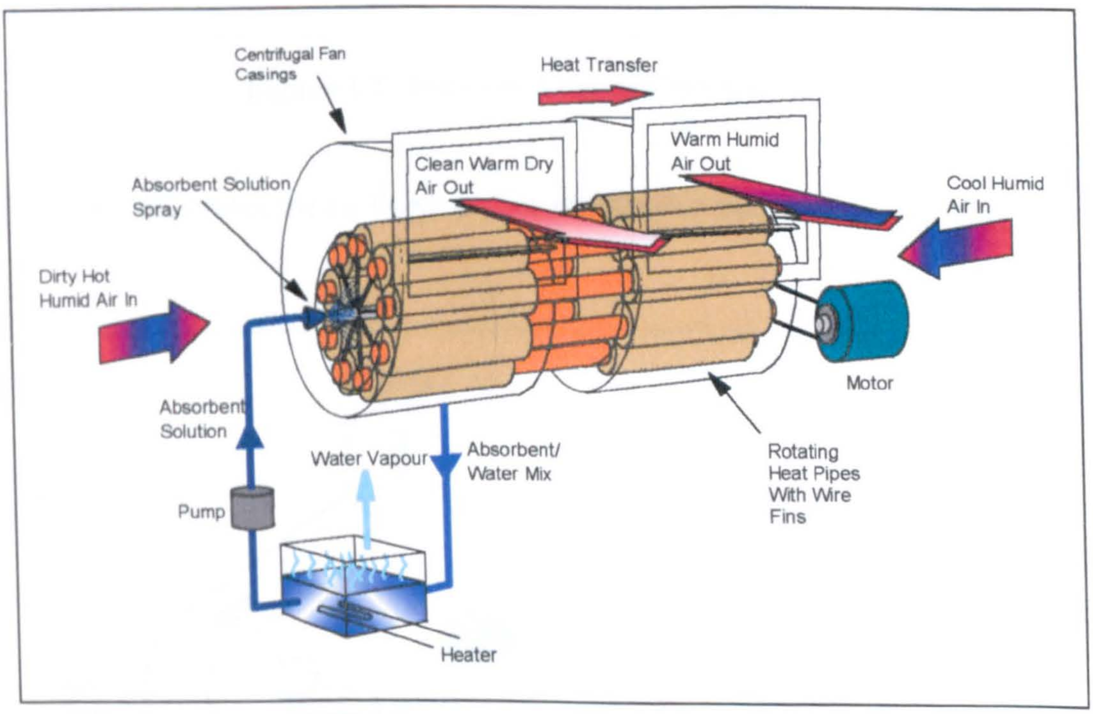


Figure I.1: Revolving Heat Pipe Absorption Dryer

Figure I.2 illustrates a system that provides cooling. Water is sprayed onto the condenser sections of the revolving heat pipes. The water picks up heat rejected from the condensers and it evaporates. The heat supplied to the condensers is from the air passing over the evaporators. Consequently, warm air passing over the evaporators is cooled. The revolving heat pipes impel the air, transfer heat and ensure the water is mixed fully with the air.

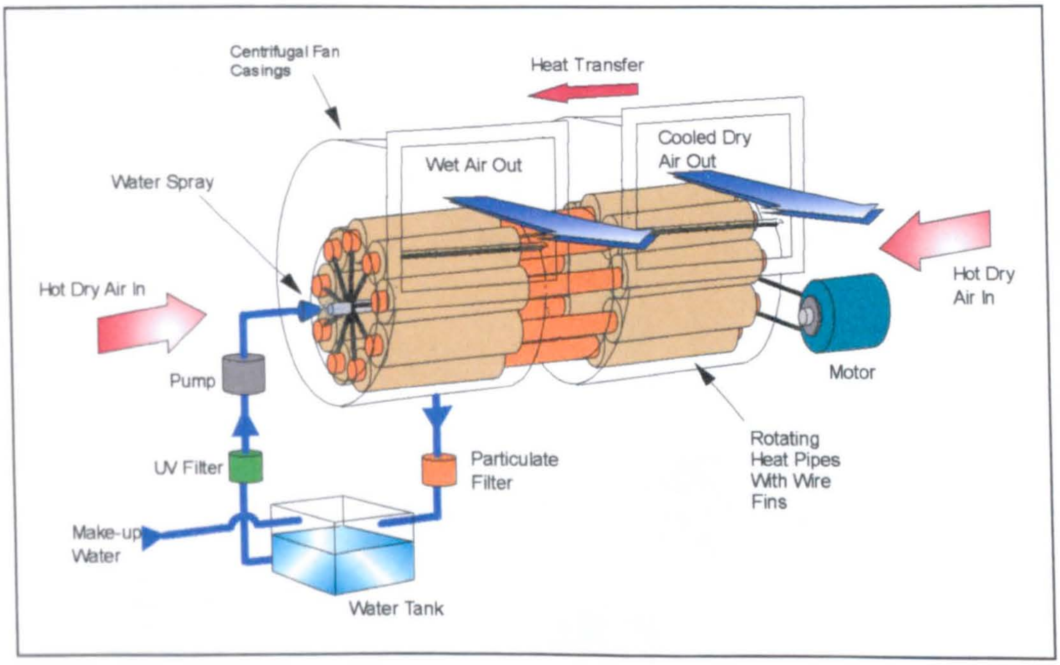


Figure I.2: Indirect Evaporative Cooler

Figure I.3 illustrates the air flows for the proposed evaporative cooler system.

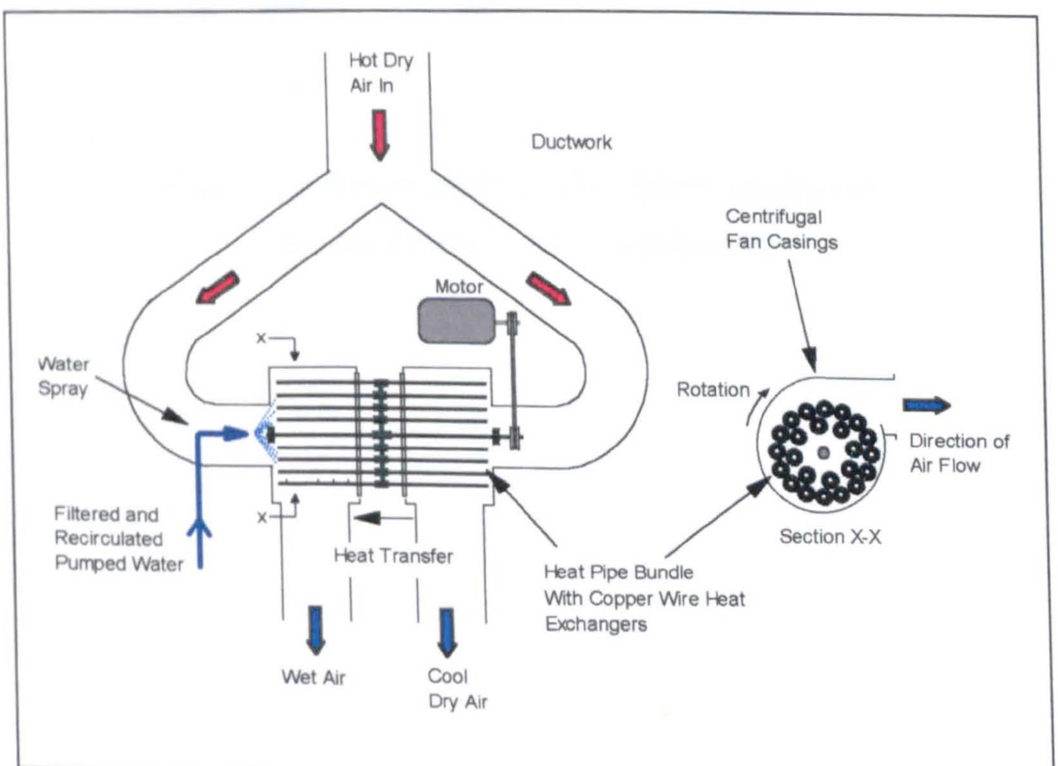


Figure I.3: Air Flows For Indirect Evaporative Cooler

Figure I.4 illustrates the proposed revolving heat pipe desiccant system for air quality and conditioning. The system is comprised of the absorption drier and indirect evaporative cooler shown above. The system is used to clean, dry and comfort cool supply air.

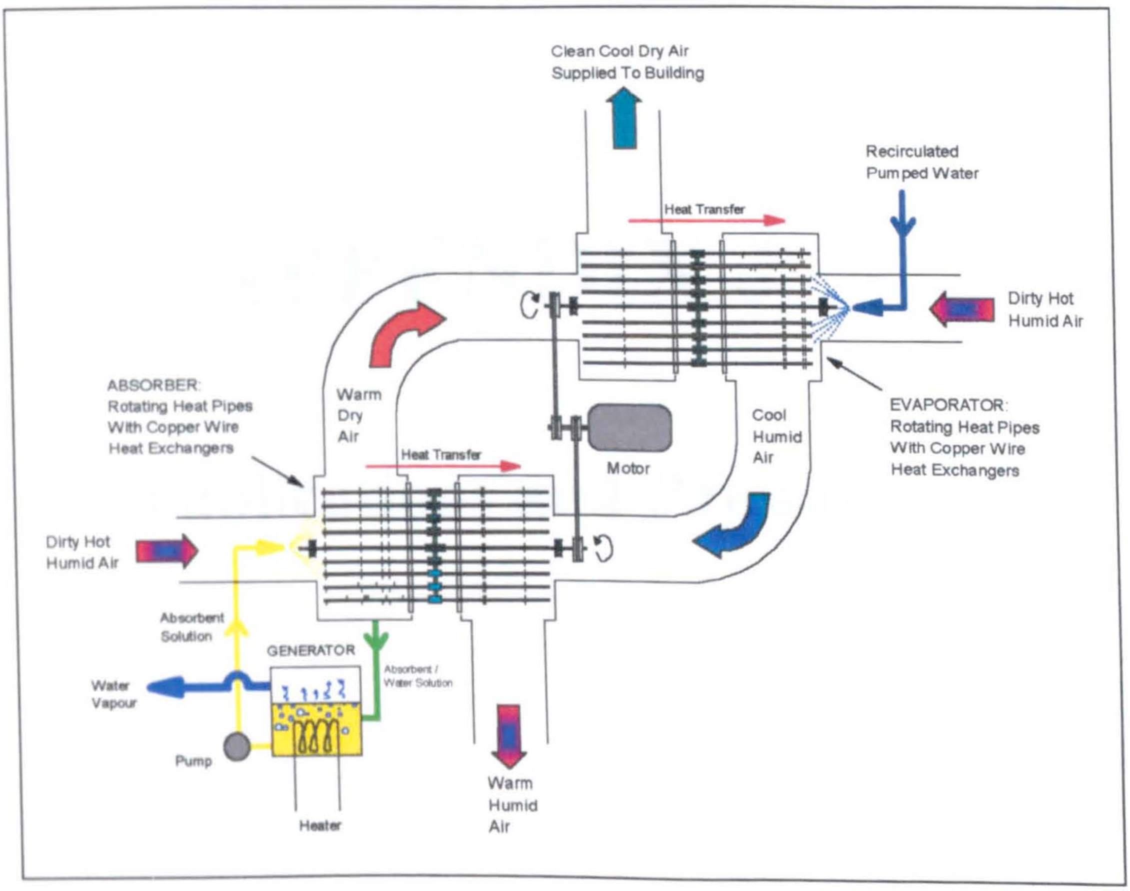


Figure I.4: Revolving Heat Pipe Desiccant System for Air Quality and Conditioning

APPENDIX J

Publications and Patents

Appendix J: Publications and Patents

List of publications by the author in chronological order:

Riffat, S. B., Shankland, N. J., Gillott, M. C., (1997). *Performance of a rotating heat exchanger.* Institute of Building Technology Internal Report, University of Nottingham, March1997.

Riffat, S. B., Gillott, M. C., Shankland, N. J., (1998). *Rotating heat pipes for combined ventilation with heat recovery.* Institute of Building Technology Internal Report, University of Nottingham, April 1998.

Riffat, S. B., Shankland, N. J., Gillott, M. C., (1998). *A Novel Ventilation/ Heat Recovery Heat Pump.* Proc. of 19th AIVC Annual Conference, Oslo, Norway, September 1998.

Riffat, S. B., Gillott, M. C., (1999). *A Novel Heat Pipe, Heat Pump System.* Proc. of 6th UK National Conference on Heat Transfer, Edinburgh, September 1999.

Riffat, S. B., Gillott, M. C., (2000). *A Novel Ventilation Heat Pump System.* Abstract accepted for the 21st AIVC Annual Conference, The Hague, Netherlands, September 2000.

Patents held by the University of Nottingham covering the technologies presented in this thesis:

Patent No. GB9522882.1, (1995). Improvements in or relating to energy apparatus, 8th November 1995, S. B. Riffat, The University of Nottingham, U.K.

Patent No. GB9522882.1, (1996). PCT 96/31750, Heat pipe with improved energy transfer, 10th October 1996, S. B. Riffat, The University of Nottingham, U.K.

Patent No. GB9507035.5, (1995). Improvements in or relating to energy apparatus. S. B. Riffat, The University of Nottingham, U.K.