**MODELLING AND EXPERIMENTAL STUDY OF AUTOMOTIVE
AIR-CONDITIONING SYSTEMS WITH CONSIDERATION OF
TWO-COMPONENT TWO-PHASE FLOW**

By

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Abstract

The purpose of this thesis work was to investigate the performance of an automotive air-conditioning system experimentally and theoretically, with consideration of two-phase flow and oil concentration. An automotive air-conditioning test system with real size automotive air-conditioning components has been established to operate as a Clutch Cycling Orifice Tube system, which can measure the vapour quality and oil concentration. A Stream Analysis Model for an accumulator of automotive air-conditioning systems has been developed, and validated by the experimental results. A swash plate compressor of automotive air-conditioning has been investigated experimentally, and the effects of two-phase flow and oil concentration on its performance have been studied. The effects of refrigerant charge, inlet air temperature to evaporator, condenser surroundings temperature, and compressor speed on the performance of automotive A/C systems have been also investigated experimentally. A new automotive A/C system with an accumulator-capillary tube-suction line heat exchanger has been developed and investigated experimentally with comparison to a conventional system without an internal heat exchanger, and a system with a liquid line-suction line heat exchanger. The results show that the new system has better performance than the conventional system, such as higher cooling load and coefficient of performance, and better compressor working conditions.

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Nomenclature

A	Empirical constant of calculation of compressor isentropic efficiency
a	Empirical constants of R-134a/oil mixture temperature
B	Empirical constant for calculating compressor volumetric efficiency
C	Empirical constants for calculation of oil viscosity
c	Oil concentration
c_p	Specific heat ($\text{kJ kg}^{-1} \text{K}^{-1}$)
D	Diameter (m)
d	Diameter (m)
E	Coefficient 1 for calculation of Dimensionless degree of superheat
F	Coefficient 2 for calculation of Dimensionless degree of superheat
e_c	Compression efficiency
e_i	Isentropic efficiency

e_m	Mechanical efficiency
e_v	Volumetric efficiency
f	Friction factor
H	Height (m)
h	Enthalpy (kJ kg^{-1})
K	Loss coefficient
L	Length (m)
M	Molecular weight (g mol^{-1})
$\dot{m}$	Mass flow rate (kg s^{-1})
N	Mole number (mol)
P	Pressure (kPa)
ΔP	Pressure drop (kPa)
Q	Cooling load (kW)
r	Ratio
Re	Reynolds number
T	Temperature ($^{\circ}\text{C}$)

T^*	Dimensionless degree of superheat for R-134a/PAG mixtures
V	linear speed (m s^{-1})
v	Compressor speed (rpm)
$\dot{V}_d$	Compressor displacement ($\text{m}^3 \text{ rev}^{-1}$)
W	Power (kW)
w	Liquid refrigerant fraction
x	Refrigerant vapour quality
y	Mole number fraction

Greek letters

ρ_f	Liquid density (kg m^{-3})
ρ_g	Vapour density (kg m^{-3})
ρ_o	PAG oil density (kg m^{-3})
ρ_{mix}	Density of R-134a/PAG mixture (kg m^{-3})
μ_f	Liquid viscosity (Pa s)

μ_g	Vapour viscosity (Pa s)
μ_o	PAG oil viscosity (Pa s)
ν	Kinematical viscosity ($\text{m}^2 \text{s}^{-1}$)

Subscript

A	Point A of the flow stream
acc	Accumulator
B	Point B of the flow stream
C	Point C of the flow stream
C	Cold reservoir, evaporator
char	Refrigerant charge
comp	Compression
cond	Condenser
D	Point D of the flow stream
dyn	Dynamic
E	Point E of the flow stream

evap	Evaporation
f	Liquid phase
g	Vapour phase
hot	Hot reservoir, condenser
i	Isentropic
in	Inlet
J	J-tube
liq	liquid
m	Mechanical
mix	R-134a/PAG mixture
o	Oil
out	Outlet
r	Refrigerant
s	Saturated
siph	Anti-siphon
stag	Stagnation

t Total (including refrigerant and oil)

V Volumetric

Abbreviations

AACTS Automotive Air Conditioning Test System

ACTSLHX Accumulator-capillary tube-suction line heat exchanger

COP Coefficient of performance

CTSLHX Capillary tube-suction line heat exchanger

IHX Internal heat exchanger

LLSLHX Liquid line-suction line heat exchanger

NIHX No internal heat exchanger

PAG Polyalkylene Glycol

Chapter 1 Introduction

1.1 Background

The development of automotive air conditioners began in earnest in 1930, when General Motors Research Laboratories conceived the idea of the vapour compression system with R-12 refrigerant. They determined the cooling capacity of the automotive air-conditioning (A/C) system to be 1 ton i.e., 200 Btu per minute. This estimate was half of the cooling capacity of the present system. The work on automotive A/C culminated at General Motors with the development of a prototype self-contained unit that was installed in the trunk of a 1939 Cadillac. In 1934, a joint venture between Houde Engineering of Buffalo and Carrier Engineering of Newark developed the first self-contained A/C system for a bus (Anon, 1935). In 1935, another bus A/C system, developed by McCord Radiator & Manufacturing, was tested and worked well on a Pacific Greyhound bus that ran between El Centro and Los Angeles through the torrid Imperial Valley during the summer of 1935 (Geschelin, 1935).

Before World War II, approximately 3,000 American cars were equipped with A/C. Most of the units were installed in expensive luxury cars. Although it was ballyhooed as a great luxury item, drivers had two major complaints: First, there was not provision for outside air. Smoking in the car made the air fetid and unbearable. Second, there were no interior controls. To shut off the system the driver had to get out of the car, open the hood, and remove a belt. After the war ended, A/C growth resumed with Cadillac advertising a

new, high-tech feature called air-conditioning controls. However, there was still one problem: The controls were mounted nearly 6 ft (1.8 m) from the driver's seat on the rear package shelf. By 1947, independent manufacturers began installing air conditioners on all makes of cars, creating a large aftermarket business.

In 1953, Harrison Radiator Division of General Motors developed a revolutionary air conditioner that could be mounted under the hood in the engine compartment (Homes, 1955). This was a much more efficient design. During 1953, about 29,000 cars were shipped with factory-installed air conditioning (Davitz, 1978). In 1954, Nash joined the select group of carmakers offering factory-installed air-conditioning. One of its most striking features was an all-season air-conditioning system. The entire Nash unit was located under the hood and weight and cost decreased significantly.

The popularity of automotive air-conditioning soared in the 1960s with the number of air conditioners installed in American cars nearly tripling from 1961 until 1964. Air-conditioning for trucks became a much-demanded accessory (Davitz, 1978). Throughout the 1960s, refinements continued to be made to render the air-conditioning system quieter and more reliable. An example of these refinements was the Chrysler Auto-Temp System that operated in response to a temperature setting selected by the driver. It provided proper velocity and temperature distribution of the air. General Motors further developed air-conditioning units in the mid-1960s; known as the Climate Control System, they were first offered on Cadillacs.

The 1970s were marked by the intense debate centered on the ozone depletion issue that threatened the continued use of the chlorofluorocarbon (CFC) compounds. Automotive air conditioners used CFC refrigerant designated R-12. Unprecedented growth of automotive air-conditioning reached an industry-wide penetration rate of 70 percent by 1980. An evaluation of alternate refrigerant was conducted in 1976. After a considerable amount of screening, R-134a was identified as the likely replacement for R-12 in the event of its total ban. Refinements to automotive air conditioners continued to be made throughout the 1970s as units became more efficient, more compact, and lighter (Bhatti, 1998).

The controversy centered on the link between CFC compounds and ozone depletion continued through the 1980s culminating in the adoption of the Montreal Protocol in September 1987. It called for a phase-out of the fully halogenated CFC compounds, including R-12. Faced with the prospect of an impending ban on R-12, carmakers developed an alternate air-conditioning system with the R-134a refrigerant. This entailed the development of new components like condensers and compressors as well as new materials like lubricant and desiccant (Bhatti, 1998).

The 1990s may be characterized as the decade of the conversion from R-12 to R-134a. Beginning in 1992, carmakers around the world started implementing the changes necessitated by the replacement of R-12 with R-134a. These changes were far from unobtrusive. Contrary to earlier expectations the conversion proved to be an onerous task since it entailed replacement of not only the refrigerant, but the lubricant and desiccant as well. Condenser, compressor, and control switches had to be changed in addition.

However, the replacement of R-12 with R-134a offered a benefit unanticipated. R-12 is a powerful greenhouse gas responsible for about 50 percent of the total equivalent warming impact of the entire vehicle. With the introduction of R-134a, the total equivalent warming impact of the air conditioner was reduced to 4.5 percent (Bhatti, 1998).

In the 2000s, the biggest change in automotive air-conditioning was the alternative refrigerants. The pursuit for environmentally friendly refrigerants caused CFCs (chlorofluorocarbons) and HCFCs (hydrochlorofluorocarbons) to gradually fade from use in the refrigeration industry. Many alternatives, such as R-600a and R-290, enter this area. Research interests in this field turned to fluids with a low GWP (Global Warming Potential) with particular attention given to carbon dioxide (CO_2), which is also known as R-744, as a candidate for refrigeration applications. CO_2 exists naturally and is a by-product of many processes such as beer or ammonia (NH_3) production. CO_2 is not produced as a refrigerant; it is captured from industrial processes, and then refined for use in refrigeration systems. Because it requires no special production effort, the overall volume of CO_2 gas in the atmosphere does not change, provided that the by-product is also released or not sequestered in some way. Therefore, using CO_2 as a refrigerant will not affect global warming. In addition to its environmental advantages, the CO_2 has attractive thermal characteristics that make it a viable alternative refrigerant (Chen and Gu, 2005).

1.2 Objectives

A goal of this work was to provide design engineers of accumulators for automotive A/C systems, with a method to determine the performance characteristics of new, high performance accumulators. To accomplish this task, a refrigeration loop capable of simulating various operating conditions of the automotive A/C system was built and empirically validated for three representative operating conditions: idle, city driving and highway driving conditions.

Another goal of this work was to investigate the effect of two-phase flow and lubricant oil on the performance of the compressor and the whole automotive A/C system, and to improve the design and development of new automotive A/C systems.

Chapter 2 Literature Review

2.1 Vapour-compression automotive air-conditioning system

2.1.1 Types of automotive A/C systems

The automotive cooling and heating systems may be thought of as two different sets of components that are combined inside the vehicle to produce the desired temperature. These circulating medium cycles are the A/C cycle and heater cycle. The A/C cycle and heater cycle share some common components: They are both mounted inside the heating-cooling module and use the module's blower to circulate the warm or cold air. They also share the same ducts and vents used to deliver the warm or cold air to the passenger compartment (Kargilis, 2003).

The basic components of the automotive A/C system are the same as other A/C systems. These components are a compressor, condenser, expansion device and evaporator. The difference between automotive and conventional A/C systems is in their operation and servicing. In a typical automotive A/C system, the belt-driven compressor is mounted on the engine. The condenser is mounted ahead of the vehicle radiator, which allows cooling air to flow over it. The evaporator is mounted inside the plenum chamber

in the passenger compartment. All of these devices are connected together by lines and hoses. Figure 2.1 shows an automotive air-conditioning system in a car.

Figure 2.1 An automotive air-conditioning system

There are two main types of automotive A/C systems: one is the Thermal Expansion Valve (TVX) system that regulates the rate of refrigerant flow into evaporator as governed by the evaporator outlet pipe temperature sensed by a sensing bulb, and the other is the Clutch Cycling Orifice Tube (CCOT) system that controls the evaporator temperature by turning the compressor on and off with a clutch cycling switch. A TVX automotive A/C system includes a compressor, a condenser, a thermal expansion valve,

an evaporator and a receiver drier as shown in Figure 2.2. A CCOT automotive A/C system includes a compressor, a condenser, a capillary tube, an evaporator and an accumulator as shown in Figure 2.3.

Figure 2.2 A TVX automotive A/C system

Refrigerant circulates through the system and exchanges heat with the environment. The compressor provides energy to the system that is driven by the automobile's engine. The vapour-compression cycle, as shown in Figure 2.4, may be divided into four functional thermodynamic processes: compression, condensation, expansion and evaporation.

In the compression process, which is shown as the line 1-2 in Figure 2.4, the compressor compresses the low temperature and pressure saturated or slightly superheated refrigerant gas to high temperature and pressure superheated gas. The amount of work required to compress the refrigerant gas is equal to the product of refrigerant mass flow rate and the change in enthalpy of the refrigerant across the compressor.

Figure 2.3 A CCOT automotive A/C system

In the condensation process, which is shown as the line 2-3 in Figure 2.4, high pressure and temperature superheated refrigerant gas is carried by a discharge line from

the compressor to the condenser where the gas cools in three phases: desuperheating, condensing and subcooling. First, the hot refrigerant transfers its heat to cooling air flowing across the condenser surface. The superheated gas cools to its dew point or condensing temperature, where it becomes saturation gas at constant pressure. Second, the saturation gas condenses at constant temperature until all of the gas becomes saturation liquid refrigerant. Finally, the saturation liquid subcools below its normal condensing temperature by further removing heat from the liquid refrigerant.

Figure 2.4 P-h diagram of vapour-compression refrigeration cycle

In the expansion process, which is shown as the line 3-4 in Figure 2.4, a liquid line transfers the high-temperature and high-pressure subcooled liquid refrigerant from the condenser to a flow control device, either an expansion valve in TVX systems or an orifice tube in CCOT systems, where it is adiabatically expanded to a low-pressure and

low-temperature mixture of liquid and gas. No heat is taken from the refrigerant during the expansion process.

In the evaporation process, which is shown as the line 4-1 in Figure 2.4, the liquid/vapour line transfers the low-temperature and low-pressure liquid/vapour mixture from the expansion device to the evaporator where the liquid fraction evaporates, and the vapour increases temperature after all the liquid evaporates. Warm ambient or circulated cabin air heats the refrigerant as it is blown across the evaporator surface by a blower and connective ducting. The evaporation process occurs at constant pressure. The difference in enthalpy between the incoming and exiting refrigerant streams is a measure of the amount of sensible and latent heat removed from the air crossing the evaporator surface.

2.1.2 Characteristics of automotive A/C systems

Compared with other A/C systems, automotive A/C systems have some significant characteristics. Therefore, automotive A/C systems present challenges not normally encountered in those found in the stationary A/C systems, such as those used in buildings. Specifically, the performance of an automotive A/C system is subjective, and consequently its design is based on the customer's comfort that must be achieved combined with specified cost, packaging, and timing constraints. The compressors of automotive A/C systems are belt-driven by the engines. The compressor speed will vary with engine speed. The refrigerant mass flow rate and cooling load change with compressor speed.

The working conditions change dramatically. The automotive A/C systems must provide comfort inside vehicles in all weather. They must heat, defog, and de-ice the vehicle, and remove dust, smoke, and odour from the air. The A/C panel outlet airflow direction, volume, velocity, and temperature must be adjustable over a wide range of climatic and driving conditions. The sun load through the windshield and side windows is much greater than that through buildings.

The system must be quiet, and the controls must be easy to understand and operate. The airflow through the ducts must meet noise-vibration-harshness standard, be free of any condensate carried off the evaporator, and any organic or foreign matter that could cause odour. The automotive A/C system must be capable of cooling down a hot vehicle quickly.

The compressor and drive, the condenser and plumbing, both A/C and heater, must be packaged in a hot restricted space. The condenser sizing must be compensated for hot ambient air during idle and city traffic driving conditions. The front and condenser airflow availability is limited by styling and bumper design constraints. The evaporator and heater blower must operate in ductwork packaged within a very small volume shared by the vehicle's instrument panel, air bags, wire harness, etc. The automotive A/C system must be small and light. It must function at idling speed, in the sun, and in the wind. There must be considerable excess capacity for normal driving speeds. All these aspects make the analysis of automotive A/C system more complicated than that of a stationary A/C system, as described by Bhatti (1998 and 1999), Kargilis (2003), and Althouse et al. (2004).

2.2 Analysis of automotive A/C systems

During the last two decades, automotive A/C systems have experienced worldwide significant advances introduced by industry and research institutes. Experimental analysis and computer simulation are effective methods to analyze automotive A/C systems. Rubas and Bullard (1995) presented a detailed experimental analysis of the refrigerant migration and its general effect on system COP. During the off-cycle in a capillary tube system, refrigerant flows from the high-pressure side to the low-pressure side and provides a thermal load on the evaporator, ultimately reducing the performance. In addition, the COP is degraded because of the finite time it takes, at the beginning of the on-cycle, to redistribute the migrated mass back to the high-pressure side. Quantification of the losses was not given in this analysis.

Buck (1996) measured short-term processes experimentally in both an automotive A/C system and a stationary refrigerating system. For this purpose, a hot-film anemometer, a pressure transducer, and a temperature sensor were used to determine the mass velocity of the gas phase, the mass velocity of the liquid phase and the superheat in the suction line. The measurements led to the conclusion that for the processes investigated, the use of superheat does not offer a safe means to control or regulate the filling of the evaporator with refrigerant. Also, the conclusion was drawn that for a fast-developing process, the superheat of gas phase in the suction line is not suitable to control the functioning of the expansion device or to control an electromechanical expansion valve.

Krause and Bullard (1996) and Coulter and Bullard (1997) quantified the losses of a cycling refrigerator with those that occur during steady state operation. A steady state model was developed and was used to compare an actual cycling refrigerator. The comparison was based on a time-averaged COP.

Kiatsiriroat and Euakit (1997) studied the performance of the vapour compression system of an automobile with R-22/R-124/R-152a refrigerant mixture. They developed a mathematical model of each component and used it to simulate the system performance. The vehicle has an engine capacity of 1300 cc, and the heat exchanges at the evaporator and the condenser are by water streams at different temperatures. They also simulated performance of the system at different working conditions. It was found that the system COP increases with the reduction of the mass fraction of R-22. A suitable composition could be obtained when the mass fraction of R-22 is in the range of 20-30 percent. As the fraction decreases from 30 percent, the COP slightly increases.

Park et al. (1999) performed experiments on a stationary refrigeration system using compact car components and simulated driving conditions. They conducted tests over a wide range of operating conditions and presented the effects of humidity, compression ratio, and air inlet temperatures on the COP.

Lee and Loo (2000) analyzed separate components of an automobile A/C system under various operating conditions. The A/C system consisted of a laminated type evaporator, a swash plate type compressor, a parallel flow type condenser, a receiver drier, and an externally equalized thermostatic expansion valve. They developed a performance

analysis computer program for the laminated type evaporator based of the overall heat transfer coefficient and pressure drop, which were obtained experimentally. They also combined the performance analysis programs of separate components of an automotive A/C system, and developed a performance simulation program for the integrated automobile A/C system. They concluded: An overcharge of 10 percent proved to be most effective for various operating conditions; any overcharge above this level, the COP tended to drop; and it is possible to select the most appropriate condenser size by executing the system performance simulation program for various operating conditions.

Cullimore and Hendricks (2001) described the need for dynamic (transient) simulation of automotive A/C systems, the reasons why such simulations are challenging, and the applicability of a general purpose of a thermohydraulic analyzer to answer such challenges. They presented an overview of modelling for the basic components, along with relevant approximations and their effects on speed and accuracy of the results.

Kaynakli and Horuz (2003) investigated an automotive A/C system in detail experimentally. They varied temperature of ambient air temperature of the evaporator and condenser, and the speed of the compressor, and investigated the performance of the system is. They analyzed cooling load, compressor power consumption, refrigerant mass flow rate, COP value, fluctuation of the minimum and maximum system pressures.

Ratts and Brown (2000) used an experimental method to analyze automotive A/C system COP, and focused on the relationships between the COP and the compressor revolution and vehicle speed. Their paper identified and quantified individual losses in

an automotive vapour-compression refrigeration system during compressor cycling. They used the second law of thermodynamics, in particular, nondimensional entropy generation, to quantify the thermodynamic losses of the refrigeration system's individual components under steady driving conditions at idle, 48.3 kph (30 mph), and 96.6 kph (60 mph). They instrumented a passenger vehicle containing a cycling-clutch orifice-tube vapour-compression refrigeration system to measure refrigerant temperature and pressure, and air temperature and relative humidity. They collected data under steady driving conditions at idle, 48.3 kph (30 mph), and 96.6 kph (60 mph). They presented a thermodynamic analysis to determine the refrigeration system's performance. This analysis showed that the performance of the system degraded with increasing vehicle speed: Thermodynamic losses increased 18 percent as the vehicle speed changed from idle to 48.3 kph (30 mph) and increased another 5 percent as the vehicle speed changes from 48.3 kph (30 mph) to 96.6 kph (60 mph). The compressor cycling rate increased with increased vehicle speed, thus increased the refrigeration system's losses. One conclusion was that the compressor's efficiency could be improved by reducing or eliminating cycling, such as could be accomplished by using a variable capacity compressor, or by not directly coupling the compressor to the engine. Another way to increase the compressor's volumetric efficiency during cycling would be to reduce the compressor operating range. This could be accomplished by using two compressors as is done in two-stage cascade refrigeration systems.

Jabardo, et al. (2002) developed a steady state computer simulation model for refrigeration circuits of automobile A/C systems, which included a variable capacity

compressor and a thermostatic expansion valve in addition to the evaporator and micro channel parallel flow condenser. In order to check results from model, the researchers developed an experimental bench made up of original components from the A/C system of a compact passenger vehicle. They equipped the refrigeration circuit with a variable capacity compressor run by an electric motor controlled by a frequency converter. They evaluated and simulated the effects on system performance of such operational parameters such as compressor speed, return air in the evaporator and condensing air temperatures by means of developed model. The following conclusions were obtained:

- 1) Increments in the particular parameter always affected the COP negatively. In the case of the refrigerant charge, this effect is only significant under overcharged conditions.
- 2) Variable parameters during the operation of the vehicle such as the condensing air temperature and compressor speed do not affect the refrigerating capacity.
- 3) The evaporator return air temperature significantly affected the refrigerating capacity.
- 4) Refrigerating capacity, mass flow rate and COP vary linearly with condensing and return air temperatures and compressor speed. Slight deviations from linear behaviour were found at low refrigerating capacities.

- 5) Condensing air temperature slightly affected the mass flow rate, refrigerating capacity, and COP; this was the opposite effect to that observed with the return air temperature in the evaporator.

Joudi, et al. (2003) investigated an automotive A/C system with alternative refrigerants, experimentally and by computer performance study. They studied five refrigerants by this model, included R-12, R-134a, R-290, R-600a and a mixture of propane and isobutane R-290/R-600a (62/38, molar percentage). They investigated the major part of this work experimentally, in which ninety two tests were conducted on refrigerants to study the effect of four parameters on system performance: outdoor air temperature, cooling load, compressor rotational speed, and the soaking temperature. They considered three outdoor air temperatures, 35, 40 and 50 °C, and employed six rotational speeds, 700, 1000, 1500, 2000, 2500 and 3000 rpm, whereas they simulated four cooling loads in the evaporator chamber, 1000, 2000, 3000 and 3500 W. The evaporator chamber was soaked at five temperatures, 45, 50, 55, 60 and 65 °C. They performed two types of tests, included single operation mode tests and multi-operation modes tests.

Gu et al. (2003) experimentally determined the two-phase flow inside an evaporator and noticed its effect on the heat transfer performance of an evaporator. Wang and Gu (2004) measured two-phase flow inside an accumulator and its effect on the performance of automotive A/C system. They measured two-phase flow using a custom-built vapour-liquid separation device and two Coriolis mass flowmeters. Data analysis showed that the accumulator quantitatively determined the two-phase state at each stage of the system,

e.g., refrigerant quality, two-phase flow rate etc, which are closely related to the performance of an automotive A/C system, but a lack of knowledge on this aspect still exists.

2.3 Accumulator

A CCOT system requires an accumulator, which is important for the performance for an automotive A/C system. However, most research work focused on improving the efficiencies of compressors, heat exchangers and expansion devices used in automotive A/C systems, while the accumulator appeared to be ignored in most studies.

Figure 2.5 shows a schematic diagram of a typical model of an accumulator and the path of the various flows through the accumulator. Feature number 1 indicates the accumulator inlet tube. Two-phase flow from the evaporator enters into the accumulator via the accumulator inlet. Feature number 2 is the accumulator wall, or housing. The housing is the part of the accumulator that holds liquid refrigerant back from entering the compressor.

Feature number 3 shows the position of the deflector. The deflector causes liquid and vapour of the two-phase flow to separate into purely liquid and purely vapour flows. The vapour flow continues into feature number 4 whereas the liquid collects in the bottom of the accumulator housing (feature number 5). Feature number 4 of the accumulator is the J-tube. During the operation of the A/C system, pure vapour is drawn

in through the J-tube inlet and past the oil bleed hole, feature number 6. The relatively high velocity of the vapour flow draws in a mist of liquid refrigerant/oil mixture through the oil bleed hole, the method by which the liquid is metered back safely to the compressor.

Figure 2.5 Schematic diagram of an accumulator

The two-phase flow safely exits the accumulator through the J-tube outlet, feature number 8. At this point in the flow, the accumulator meters an acceptable amount of oil back into the flow to prevent damage to the compressor.

The primary functions of the accumulator are to protect the compressor from the sudden rush of liquid that can occur at start-up, and to act as a reservoir for excess refrigerant (Radermacher and Schwarz, 2002). Once the accumulator has trapped a surge of liquid, the accumulator must meter the liquid refrigerant and oil back to the compressor at a rate that will prevent compressor damage (Grim, 1975). Modern accumulators also filter the liquid flow before it exits and removes foreign particles or water vapour that may be suspended in the refrigerant that could lead to a failure of the system (Radermacher and Schwarz, 2002). Presently, accumulators do not filter 100 percent of the flow. Currently there is no filter on the gas flow and the only filter is on the liquid entering the J-tube, to protect the compressor. Also, the desiccant bag does not provide 100 percent filtration of the water moisture flow.

Another benefit of the accumulator is its positive effect on regulating an overcharged system. The overcharging of an A/C system is potentially dangerous, but very desirable in an automotive A/C system. Rubber hoses isolate vibrations in automotive A/C system, but the rubber naturally aspirates a small amount of refrigerant, which over time could lead to a serious reduction in the charge within the system. To avoid having to recharge the system every year or so, the use of an accumulator can allow the system to be safely overcharged by up to 50 percent without seriously affecting the performance of the system (Radermacher and Schwarz, 2002).

Properly designed and applied, the accumulator offers the most cost-effective compressor protection against liquid slugging (Hall, 1975). The accumulator also controls the lubrication of the compressor by metering the return of liquid and oil to the compressor, which could affect the lubricating properties of the compressor oil or wash the lubrication from the cylinder and cylinder walls (Grim, 1975).

However, there are several disadvantages to using an accumulator: For instance, the accumulator introduces a slight pressure drop to the system that causes the compressor to run at lower suction pressures (Bottum, 1975). Selection methodology is also a disadvantage of the accumulator; the current selection criteria is haphazard (this is actually done more by precedent and experience than by design) and may not produce the desired results to the system (Grim, 1975). Perhaps another drawback to the use of an accumulator is the increased initial and maintenance costs of the system. However, balanced against the cost of prematurely replacing a compressor, the relative low cost of the accumulator is well justified. The only drawbacks, therefore, of using an accumulator relate to system efficiency. A proper design and optimization methodology will remedy the detrimental effects on the performance of the system.

To ensure sufficient surge suppression capacity, the accumulator volume is normally made larger than the entire volume of refrigerant (in liquid form) and compressor oil in the system. A small increase of the liquid level in the accumulator will therefore draw a significant percentage of the refrigerant charge out of the refrigeration cycle. The liquid refrigerant stored in the accumulator is effectively removed from the cycle, which does not contribute to the cooling effect of the system. Storing the excess

charge of the system is the method by which the accumulator regulates the evaporator outlet quality at all operating points. With a smaller charge, both the high-side and low-side pressures decrease, which lowers the saturation temperatures on both sides of the system. Lowering the saturation temperatures has the effect of lowering the heat load at the condenser and increasing the heat load at the evaporator. This changing of the heat loads causes the evaporator outlet quality to rise. When the evaporator outlet quality drops, the accumulator stores the excess liquid, reducing the amount of charge flowing through the system. The reduced charge has the effect of raising the evaporator outlet quality once more, so a balance is reached that ensures an appropriate refrigerant quality is always returned to the compressor. This virtual reduction in the charge of the system is the means by which the accumulator regulates the quality of the refrigerant flow to the compressor (Radermacher and Schwarz, 2002).

Modern accumulators use an internal desiccant bag to remove moisture from the system and to introduce a dye to the refrigerant/oil mixture to make finding leaks in the system much easier. The liquid refrigerant flow also passes through a filter to remove any potentially harmful particulate matter.

Yun (1998a) described the design of scroll compressors and the suction accumulator used in rotary compressors. It was said that the primary function of a suction accumulator in rotary compressor is to control excess liquid refrigerant. Suction accumulators also serve as a multi-function component in a rotary compressor, including the following functions:

- 1) Filter – to protect compressor from the entering of small particles.
- 2) Muffler – to act as a suction acoustic muffler, ranging widely in overall sound level and in specific frequency bands.
- 3) Reservoir – to separate and collect liquid-phase refrigerant from suction stream.
- 4) Performance – to affect compressor performance as a result of suction gas pulsation, pressure drop inside accumulator and heat transfer between accumulator, and surrounding ambient and compressor shell.
- 5) Liquid refrigerant evaporator – to evaporate liquid refrigerant when receiving heat from surroundings.
- 6) Noise radiator – act as a noise radiator as well as a generator. A major enhancement of an accumulator is introduced in this paper.

Yun (1998b) also proposed a new compressor accumulator that differed from conventional suction accumulators. The new accumulator distinguished itself from the conventional design in that it was pre-charged with oil. A concept was introduced to improve the oil/refrigerant ratio by having an accumulator with pre-charged oil. Basically, the concept involved an accumulator pre-charged with oil that is retained in the accumulator. As liquid refrigerant enters the accumulator, it is mixed with oil in the accumulator and improves the quality of oil/refrigerant mixture before entering the compression chamber of the rotary compressor. From the concept to experimental

verification the proposed accumulator, through development stages, has proven its basic feasibility, performance, and reliability.

Hwang (2001) proposed a new method of measuring the flow quality in the accumulator outlet of an automotive A/C system, based on the use of a stratification tube to stratify HFC-134a refrigerant/PAG oil two-phase flow in a horizontal tube without the use of additional heating or condensing sections. By measuring the vapour velocity with a hot-wire anemometer and the liquid level in the stratification tube, the flow quality and total mass flow rate can be determined. The feasibility of the proposed method was checked by conducting experiments using a nitrogen gas/water flow loop. The results showed that accurate measurements of flow quality, within 6.8 percent, can be made using the present method. A compressor-based HFC-134a refrigerant/PAG oil refrigeration loop was used to check if the presence of lubricant oil would cause any error in the determination of thermodynamic quality based on a heat balance. The present flow observations and the flow quality measurements showed substantial discrepancies between the thermodynamic qualities determined based on a heat balance and the measured flow quality. Hence, the work demonstrated that the lubricant oil can significantly alter the thermodynamic relationship of pure HFC-134a refrigerant, such that the thermodynamic quality of the refrigerant-lubricant oil mixture, can not be reliably determined based on a heat balance and equilibrium thermodynamic relationships available for pure HFC-134a refrigerant.

Schwarz et al. (2002) developed a mathematical model to simulate the behaviour of accumulators and their influence on the oil supply to the compressor. In the model, the

influence of anti-siphon hole was neglected, and the deflector was modelled by changing the loss coefficients when the accumulator geometry was reduced. According to their model, the primary tasks of the accumulator are to store excess refrigerant, to protect the compressor against larger quantities of liquid coming from the evaporator at start-up, and to ensure a sufficient oil supply to the compressor. The vapour quality was considered as one of the input conditions. However, the model was not validated by experimental data.

Figure 2.6 Schematic diagram of experimental set-up (Lee et al. 2003)

Lee et al. (2003) investigated the flow structure inside the accumulator of a rotary compressor using a hybrid particle tracking velocimetry (PTV) velocity measurement technique, as shown in Figure 2.6. A prototype accumulator was studied while operating

under real working conditions. The flow inside the accumulator was qualitatively visualized using a high-speed camera. The period of one cycle of the rotary compressor was divided into 4 phases and the velocity fields inside the accumulator were measured using a phase-averaging technique. The flow inside the accumulator showed a pulsating periodic flow structure according to the rotational phase of rotary compressor.

2.4 Compressor performance

In current automotive A/C systems, most compressors used are reciprocating piston compressors. In this study, a swash plate compressor, one kind of reciprocating piston compressor, is investigated. The compressor performance is affected by several factors, including compressor design, compressor speed, suction and discharge pressure and temperature, heat transfer, internal gas leakage, and valve inefficiencies. These effects on compressor performance are measured by volumetric and isentropic efficiencies. Volumetric efficiency is the ratio of actual volume of gas (or two-phase) entering the compressor to the geometric displacement of the compressor. Isentropic efficiency is the ratio of the work required for isentropic compression of the gas to work input to the compressor shaft (ASHRAE Handbook, 2000).

For a given reciprocating compressor, its efficiency varies with the operating conditions. According to Dossat (1991), the compressor efficiency is primarily a function of the compression ratio, and the efficiency decreases with an increasing compression

ratio. Tian et al. (2004) analyzed a variable displacement wobble plate compressor for an automotive A/C system using mathematical modelling and experiments. They modelled the efficiencies of the compressor as functions of the compression ratio and the compressor speed. SINDA/FLUINT (2003) also modelled compressor efficiency as functions of compression ratio and compressor speed. Both Tian et al. (2004) and SINDA/FLUINT (2003) obtained the results that the efficiency of a compressor decreases with an increasing compression ratio.

Cullimore and Hendricks (2001) characterized the compressor by a compressor displacement ($\dot{V}_d$), comprehensive isentropic and volumetric efficiencies. The compressor isentropic efficiency (e_i) and volumetric efficiency (e_v) are characterized by the following relationships, with respect to the compression ratio (r) and the compressor speed (v), respectively:

$$e_i = 1 - \left(\frac{A_0}{rv} + \frac{A_1}{r} + \frac{A_2}{v} + \frac{A_3}{r} + A_4 + \frac{A_5 r}{v} \right) \quad (2.1)$$

$$e_v = 1 - \left(\frac{B_0}{v} + B_1 + \frac{B_2 r}{v} + B_3 v + B_4 r \right) \quad (2.2)$$

where, the eleven constants A_0 through B_4 are curve fit coefficients that represent the compressor map of a specific compressor.

However, the above studies all ignored the effect of two-phase flow during compression process on compressor efficiencies. Theoretical computations indicate that

wet compression will bring about desirable gains in compression efficiencies and reduce the work of compression. This would be true if the small particles of liquid vaporized during the actual compression of the vapour. However, in practice, this is not the case. Since heat transfer is a function of time, and since compression of the vapour in a modern high-speed compressor takes place very rapidly, there is not sufficient time for the liquid to completely vaporize during the compression stroke. Hence, some of the liquid particles remain in the vapour in the clearance volume and vaporize during the early part of the suction stroke. This action may influence the compressor volumetric and isentropic efficiencies (Dossat, 1991).

Cho et al. (2003) performed an experimental study on the performance of the inverter-driven scroll compressor to investigate the optimum operating conditions of the liquid injection technique. The influence of the liquid injection on the performance is presented as a function of operating parameters and injection location by comparing the results with those for the non-injection case. It was observed that liquid injection under high frequency was very effective at attaining higher performance and reliability of the compressor, but injection under low frequency showed some disadvantages.

Dutta et al. (2001) investigated the influence of liquid refrigerant injection on the performance of a refrigerant scroll compressor using experimental and theoretical methods. In the theoretical analysis, a compression model of vapour/liquid mixture was developed by taking account of heat transfer from the cylinder wall to suction, compression and injection refrigerant. An experiment was done, under the condition of keeping the oil temperature constant, in order to investigate the fundamental influence of

the liquid refrigerant injection on the compressor performance, and the results were compared with the theoretical ones. It was found that the injection basically increased the compression power and decreased the compressor efficiency, though the situation depends on the condition of the heat transfer to the injection refrigerant. Furthermore, the performance of the liquid refrigerant injection compressor under practical operating condition without controlling the oil temperature was investigated. Under this condition, the compressor showed recovery and slight improvement of performance due to the decrease of the oil and cylinder temperatures by the injection.

Tassou and Qureshi (1998) investigated the performance of positive-displacement refrigeration compressors for variable-speed capacity control applications. Compressors tested included an open-type reciprocating, a semi-hermetic reciprocating and an open-type rotary vane. All three compressors tested under constant and variable head-pressure operating conditions, yielded results that indicated that all three compressors were designed for maximum efficiency at nominal speed. At constant head-pressure only the open-type compressor exhibited an improvement in the COP at reduced speeds. With variable head-pressure-control all three compressors showed an increase in the COP with a reduction in speed. The results of an energy analysis carried out based on the experimental results and weather conditions for two locations representing temperate and warm weather conditions showed that all three compressors, when operated at variable speed, offer energy savings compared to their fixed-speed counterparts. The analysis showed the open-type reciprocating compressor to be the most efficient system offering

12 percent savings when operating in a temperate climate and 24 percent savings when operating in a warm climate.

Longo and Gasparella (2003) developed a specific one-dimensional model to analyze the unsteady state of the compression cycle of a small hermetic reciprocating compressor for domestic refrigeration. The specific one-dimensional model of the valves was developed and the mass and energy balances were applied to the refrigerant inside the cylinder to determine the mass, pressure, and temperature behaviour and the heat and work transfer through the compression process. This analysis combined with a traditional steady state model of the compressor can evaluate the efficiency of the compression cycle and the performance of the compressor unit. The whole simulation code was validated against the experimental measurements carried out on an R-134a commercial unit in a wide range of operative conditions.

Sami and Aucoin (2003) investigated liquid injection impact on the performance of new alternative refrigerant mixtures such as: R-410A, R-507, R-407C and R-404A. The test results were obtained using an air-source heat pump set-up with enhanced surface tubing under various liquid injection ratios. Performance tests were conducted according to the ARI/ASHRAE standards. The performance data demonstrated that as liquid injection ratio increases, compressor head pressure and discharge temperature decrease. This has a positive effect in protecting the compressor. The effect of liquid injection on mixture behaviour varies from one mixture to another, depending upon the mixture's composition. Furthermore, liquid injection appears to have a significant influence on R-410A behaviour compared to the other mixtures in question.

Furthermore, Sami and Aucoin (2003) studied the impact of gas/liquid injection on the behaviour of new alternative refrigerant mixtures such as, R-410A, R-507, R-407C, and R-404A, and compared it to that of no injection. The test results under heating conditions demonstrated that gas/liquid injection has a positive effect in reducing the head pressure and temperature of the compressor, and therefore protecting the compressor. The study also showed that the effect of gas and gas/liquid mix injection on mixtures varied depending upon the mixture composition. It appeared from the sample results presented that the refrigerant hot gas injection had a positive impact on the COP on the condenser side compared to the liquid injection and the mix of gas/liquid cases. Some refrigerant mixtures with higher boiling points, such as R-404A and R-407C appear to have positive reaction to gas injection.

Shiva Prasad (2002) developed a simple method of estimating the cylinder pressure and "rod load" (force on the crosshead pin, in compressor terminology) in a double-acting reciprocating compressor used in natural gas pipeline and chemical industries. The results showed that the pressure increases with increasing liquid injection.

For common A/C systems, it is acceptable to neglect the influence of liquid in predicting the efficiency of compressors, because the liquid fraction is very low at compressor inlet. However, for automotive A/C systems, the liquid fraction is as high as 10 ~ 20 percent at the compressor inlet; therefore the effects of liquid cannot be neglected. According to Cheremisinoff (1992), of all the compressors, lubricated reciprocating compressors are the most sensitive to liquid carryover. The oil used in automotive A/C systems is miscible with refrigerant, and the oil concentration is about 5 percent. No

studies conducted analyzed the performance of reciprocating compressor with so much liquid refrigerant and oil in compression process.

2.5 Internal heat exchanger in A/C systems

Internal heat exchangers (IHX) were widely used in all kinds of refrigeration systems to improve system performance and ensure proper system operation. There are two kinds of IHX: liquid line-suction line heat exchanger (LLSLHX) and capillary tube-suction line heat exchanger (CTSLHX). The LLSLHX can be installed in all kinds of systems, but the CTSLHX can only be installed in systems where a capillary tube serves as the expansion device.

Generally, IHXs subcool the liquid refrigerant and superheat the suction gas (ASHRAE Handbook, 1998). They were used for one or more of the following functions: to increase the efficiency of the refrigeration cycle, subcool the liquid refrigerant to prevent flash gas at the expansion device, and to evaporate small amounts of expected liquid refrigerant returning from evaporator in certain applications.

Domanski et al. (1994) conducted a theoretical evaluation of the performance effects resulting from the installation of a LLSLHX. They examined cycle parameters and refrigerant thermodynamic properties that determined whether the installation resulted in improvement of COP and volumetric capacity. The study showed that the benefit of application of the LLSLHX depended on a combination of operating conditions and fluid

properties - heat capacity, latent heat, and coefficient of thermal expansion - with heat capacity being the most influential property. Fluids that performed well in the basic cycle were marginally affected by the LLSL-HX, and the impact on the coefficient of performance and volumetric capacity may be either positive or negative. Fluids that performed poorly in the basic cycle benefited from the LLSL-HX with an increase of the coefficient of performance and volumetric capacity. This paper described a fundamental analysis of LLSLHX application. The results presented were based on theoretical evaluations using thermodynamic properties, with the assumption of isentropic compression, no-pressure-drop infinite heat exchangers, and no-pressure-drop liquid-line/suction-line heat exchanger. In a real system other factors, such as transport properties, heat exchanger design, heat transfer fluid temperature profiles, and compressor performance, would also affect the system performance.

Apréa et al. (1999) studied the influence of adopting a LLSLHX on the COP of refrigeration systems, from a thermodynamic point of view, using several kinds of refrigerants such as CFCs, HCFCs, and other substitutes. They compared the coefficient of performance of two cycles, with and without a suction/liquid heat exchanger. It was found that a simplified choice criterion could be employed if some assumptions are made; these assumptions introduced errors which were verified to be less than 5 percent. The criterion was positively checked for several working fluids such as CFCs, HCFCs, and other substitutes.

Klein et al. (2000) studied the effect of LLSLHXs in refrigeration systems. This study can be distinguished from the previous studies in three ways: First, this paper

identified a new dimensionless group to correlate performance impacts attributable to liquid-suction heat exchangers. Second, the paper extended previous analyses to include new refrigerants. Third, the analysis included the impact of pressure drops through the liquid-suction heat exchanger on system performance. From detailed analyses, it can be concluded that the liquid-suction heat exchangers that have a minimal pressure loss on the low-pressure side are useful for systems using R-507A, R-134a, R-12, R-404A, R-290, R-407C, R-600, and R-410A. However, the liquid-suction heat exchanger is detrimental to system performance in systems using R-22, R-32, and R-717.

Kim (2002) performed tests for R-22 and four alternative fluids (R-134a, R-32/134a (30/70 percent), R-407C, and R-410A) at operating conditions typical for a residential air conditioner. The study was performed in an experimental breadboard water-to-water heat pump in which a water/ethylene glycol mixture was used as the heat transfer fluid. The heat exchangers representing the evaporator and condenser were counter flow and cross flow, respectively. In tests performed at the same capacity, application of the LLSLHX was beneficial to COPs of all fluids tested, and R-410A had the highest COP.

The design and analysis of capillary tubes have been studied both analytically and experimentally, mostly for adiabatic capillary tube (Bansal and Rupasinghe, 1998; Bittle and Pate, 1994; Jung et al., 1999; Kuehl, 1991; Melo et al., 1999; Sami and Tribes, 1998; Sami et al., 1998; Wong and Ooi, 1996; Wongwises et al., 2000; Wongwises and Pirompak, 2001). The flow characteristics in non-adiabatic capillary tube have received comparatively little attention in the literature.

Xu and Bansal (2002) developed a homogeneous model of refrigerant flow through capillary tube–suction line heat exchangers, which were widely used in small vapour compression refrigeration systems. The homogeneous model was based on fundamental conservation equations of mass, momentum and energy. Their analysis of the flow revealed that the flow characteristics inside a non-adiabatic capillary tube were determined by the interaction between the heat transfer and pressure drop effects of the refrigerant flow.

Garcia-Valladares et al. (2002a and 2002b) developed a detailed one-dimensional steady and transient numerical simulation of the thermal and fluid-dynamic behaviour of capillary tube expansion devices working with pure and mixed refrigerants. The governing equations used an implicit step-by-step method. A special treatment was implemented in order to consider transitions (subcooled liquid region, metastable liquid region, metastable two-phase region, and equilibrium two-phase region). All the flow variables, (enthalpy, temperatures, pressures, vapour quality, velocities, heat fluxes, etc.) together with the thermophysical properties were evaluated at each point of the grid in which the domain was discretized. The numerical model allowed analysis of aspects such as geometry, type of fluid (pure substances and mixtures), critical or non-critical flow conditions, metastable regions, adiabatic or non-adiabatic capillary tubes, and transient aspects. They compared the simulation results of the non-adiabatic tubes to the available experimental data of Melo et al. (2000).

Sinpiboon and Wongwises (2002) developed a mathematical model to study flow characteristics in non-adiabatic capillary tubes. The theoretical model was based on

conservation of mass, energy and momentum of fluids in the capillary tube and suction line. The mathematical model was categorized into three different cases, dependent on the position of the heat exchange process. The first case considered was when the heat exchange process started in the single-phase flow region, the second case considered was when the heat exchange process started at the end of the single-phase flow region, and the last case considered was when the heat exchange process took place in the two-phase flow region. Type of refrigerant, condenser temperature, evaporator temperature, suction line inlet temperature, degree of subcooling, inner diameter of capillary tube and suction line, initial length of capillary tube, heat exchange region length, relative roughness, and an appropriate two-phase viscosity model can be varied independently for different mass flow rate of refrigerant to determine the final length of capillary tube.

Bansal and Xu (2003) carried out an extensive parametric analysis of non-adiabatic capillary tube flow. It was found that the non-adiabatic capillary tube flow characteristic is discontinuous in some situations and the discontinuity was caused by the re-condensation of the refrigerant within the heat exchanger due to strong heat exchange with the suction line. This is particularly true for the refrigerant entering the capillary tube with higher condensing temperatures (e.g., 320.1 K) that caused convergence problems in the numerical solution process. The refrigerant, therefore, entered the outlet adiabatic region as sub-cooled liquid, which needed much longer outlet adiabatic length for it to reach critical state. This aspect needed special attention while modelling the non-adiabatic capillary tube flow.

Jain and Bullard (2004) explored the effect of CTSLHX geometry on system performance at various design and off-design conditions by embedding it in a system model. A detailed finite-volume model of the capillary tube and suction line, capable of handling all the phase-change complexities was used. All the CTSLHX configurations considered met the design constraints and did not affect the design COP very much. Capillary tubes with large inlet sections and relatively small outlets were found to give the best performance at all the simulated off-design perturbations.

Islamoglu et al. (2005) applied an artificial neural network (ANN) model to predict the performance of a non-adiabatic capillary tube suction line heat exchanger. The results showed that the ANN approach could be considered as an alternative and practical technique to evaluate the refrigerant suction line outlet temperature and mass flow rate. However, this approach is generally time-consuming to develop the best configuration in the training period.

Yang and Bansal (2005) investigated the effect of different parameters on the performance of capillary tube-suction line heat exchangers (CTSLHX), including condensing and evaporating temperatures, degrees of subcooling and superheat, tube diameter, tube length, and refrigerant flow inlet quality. The heat transfer rate from the capillary tube to the suction line was simulated for two environmentally friendly refrigerants, namely HFC-134a and HC-600a. It was found that R-134a performed better, in terms of heat transfer rate and evaporator capacity, than R-600a.

Boewe et al. (1999 and 2001) analyzed the influence of the LLSLHX with R-744 as the working fluid in automotive A/C systems, where the presence of the LLSLHX has a great influence on the energy efficiency.

Zhang et al. (NIST, 1998) applied the LLSLHX to a production vehicle in order to improve the performance of R-134a A/C systems, and found that by installing a drop-in IHX into an existing R-134a A/C system, most or all of the following benefits can be obtained:

- 1) Equal or higher cooling capacity with higher COP.
- 2) Equal or better A/C system performance with less charge.
- 3) Lower compressor discharge pressure (improved compressor life).
- 4) Better compressor protection (no "wet" compression).
- 5) "Flooded" evaporator in a TXV system.

In chapter 5, a CTSLHX was introduced, which was specially developed for automotive A/C systems and investigated with comparison to a conventional system without an IHX and a system with a LLSLHX. The experimental setup consisted of real-sized parts of automotive A/C system. Conclusions from experimental data revealed some features that were not presented in previous researches.

2.6 Carbon dioxide A/C systems

Refrigeration and air conditioning systems influence the environment. Currently used refrigerants are ozone-depleting substances or have the global warming potential. The discovery of ozone-depleting effects and the global warming potential of CFCs and HCFCs led to the Montreal Protocol and the UN Framework Convention on Climate Change and its Kyoto Protocol. In these documents, one of the most important actions relative to refrigeration is the replacement of the previously used CFCs/HCFCs refrigerant with low ODP and low GWP refrigerants.

The alternative to CFCs is the use of natural refrigerants such as, ammonia, propane, butane, carbon dioxide, water, and air. The ODP of these natural refrigerants is zero, and most of them have close to zero-GWP. However, some of them can be flammable and/or toxic. Carbon dioxide is one of the few natural refrigerants that is neither flammable nor toxic. It is inexpensive, widely available, and does not affect the global environment as do many other refrigerants. CO₂ has a GWP at 1, but the net global warming impact when used as a technical gas is 0, since the gas is a waste product from industrial production. CO₂ is an excellent alternative among the natural refrigerants, especially in applications where the toxicity and flammability of ammonia and hydrocarbons may be a problem. CO₂ has been regarded as favour across the broad spectrum of automotive, domestic, commercial, and industrial refrigeration and air conditioning systems (Neksa 2002).

In former days CO₂ was used extensively as a refrigerant, particularly in marine installations, owing to its inherent safety and other advantageous features, up to beyond 1950. It was abandoned in favour of the halocarbons as a result of the high power consumption required and loss of capacity at high ambient temperatures. The theoretical COP of CO₂ in simple cycle operation is lower than that for the other refrigerants, and probably equally because of the failure of the manufacturers to follow modern trends in compressor design. However, according to the research of Lorentzen and Pettersen (1993), an improvement in COP can be obtained by using an internal heat exchanger. By development and practical testing of a laboratory prototype, they concluded that CO₂ is a viable refrigerant in car air-conditioners, which would solve all present and future environmental problems associated with such systems. Use of CO₂ as a refrigerant may provide a number of advantages in the present situation, both from an environmental and practical point of view. It is their considered opinion that use of the old refrigerant CO₂ offers a key to the complete solution of the environmental problems in many areas of refrigeration usage.

Later, Lorentzen (1994) designed a new transcritical CO₂ system for use in motor-car air conditioning. Extensive testing demonstrated that it was very competitive with conventional equipment with regard to power consumption, compactness and cost. By using the special properties of CO₂ in transcritical operation, it is possible to achieve a nearly ideal fit to the gliding temperature, which is required in applications like large district-heating schemes. The high pressure differential and relatively small flash-gas

volume open the possibility of efficient recovery of the expansion energy in a compact expander.

Brown et al. (2002) did a comparative analysis of an automotive air conditioning systems operating with CO₂ and R-134a. In their research, the R-134a system had a current-production configuration, which consisted of a compressor, condenser, expansion device, and evaporator. The CO₂ system was additionally equipped with a liquid-line/suction-line heat exchanger. The analysis showed that CO₂ had an inferior COP to R-134a. The COP disparity depended on compressor speed (system capacity) and ambient temperature; the higher the speed and ambient temperature, the greater the COP difference. The entropy generation calculations indicated that CO₂ had a somewhat better performance than R-134a in the evaporator, but had significantly poorer performance in the gas cooler than R-134a in the condenser.

Liu et al. (2005) designed and constructed a CO₂ automotive air conditioner prototype. The lubricant, the CO₂ charge, the evaporator outlet pressure, the compressor speed, the air inlet temperature and flow rate of the gas cooler, the air flow rate of the evaporator, and the performance of the prototype was experimentally investigated. Also, they analyzed the cooling capacity, compressor power consumption, CO₂ mass flow rate, and COP value. The experimental results showed that the CO₂ system performance was greatly affected by different lubricants, the CO₂ system performance was sensitive to the mass charge, the high side pressure affected the system performance greatly, and the system needed a high side pressure controller.

Tamura et al. (2005) developed an automotive cooling and heating air conditioning system using CO₂ as a refrigerant, in order to solve a problem that the present automotive heating air conditioning system cannot provide sufficient heating capacity, because the heat released from the eco-car's engine decreased. In their system, a CO₂-based heat pump was utilized, whereby the waste heat recovered from the heat pump cycle during dehumidification of the incoming air (referred to as the dehumidifying condition) was used as an auxiliary heat source instead of an electric heater.

Niu et al. (2007) constructed a wet-compression absorption CO₂ refrigeration cycle by adding a non-volatile liquid into a CO₂ refrigeration cycle. In this system, the high side pressure was less than 35 bar, which was tremendously reduced compared to 120 bar in the transcritical CO₂ refrigeration cycle. The relationship between system performance and the cycle ratio and internal heat exchanger efficiency was tested. The components used in the demonstrator were entirely based on existing components and not optimized and considerable potential exists for efficiency improvements.

Chapter 3 Theoretical analysis

3.1 Properties of refrigerant and oil

3.1.1 Properties of refrigerant R-134a

The thermodynamic properties of R-134a were taken from NIST database (1998), and for the convenience of calculation, the correlations between pressure and other properties are in the form of polynomial equations. In order to reduce deviation between the correlations and NIST database, the pressure range selected is from 200 kPa to 500 kPa, corresponding to the pressure inside accumulator. The following equations were deduced:

$$T_s = 1.7276 \times 10^{-7} \times P^3 - 2.7845 \times 10^{-4} \times P^2 + 0.2135 \times P - 4.2985 \times 10 \quad (3.1)$$

$$\rho_f = -3.4835 \times 10^{-7} \times P^3 + 6.1940 \times 10^{-4} \times P^2 - 0.5849 \times P + 1.4219 \times 10^3 \quad (3.2)$$

$$\rho_g = 5.6795 \times 10^{-9} \times P^3 - 5.1317 \times 10^{-6} \times P^2 + 4.9076 \times 10^{-2} \times P + 3.5680 \times 10^{-1} \quad (3.3)$$

where, T_s , ρ_f and ρ_g are the saturated temperature, liquid density and vapour density, respectively, while P is the pressure. The deviation between the Equation (3.1) and NIST database is $\pm 0.039^\circ\text{C}$, and the deviation between Equations (3.2), (3.3) and NIST database is ± 0.044 percent. The viscosity of R-134a was assumed to be only temperature dependent, and can be obtained from DuPont Product Information (2005) in

tabular and equation forms. For convenience of calculation, the following equation forms were used:

$$\mu_f = (-0.0002191 \times T^3 + 0.039304 \times T^2 - 3.6494 \times T + 267.67) \times 10^{-6} \quad (3.4)$$

$$\mu_g = (11.021 + 0.038599 \times T) \times 10^{-6} \quad (3.5)$$

where, μ_f and μ_g are the liquid viscosity and vapour viscosity, respectively.

3.1.2 Properties of oil

The lubricant used for refrigerant R-134a is a synthetic lubricant generally referred to as Polyalkylene Glycol (PAG) oil. PAG oil is miscible in liquid R-134a. The equation for the density of PAG oil is an empirical model created from measurements taken by Grebner and Crawford (1992). It was assumed that the density is only a function of temperature and that the dependence is a linear one, and knowing that the density at a specific temperature (15°C) is $\rho_{15^\circ\text{C}} = 993.3 \text{ kg/m}^3$, the following equation is formulated:

$$\rho_o = 1004.054 - 0.716934T \quad (3.6)$$

where, ρ_o is the density of PAG oil at temperature T in °C.

The viscosity of PAG oil was also assumed to depend on temperature only and usually can be described for pure liquid by the following equation:

$$\ln(\mu) = C_1 + \frac{C_2}{T} + C_3T + C_4T^2 \quad (3.7)$$

All four terms are necessary for temperatures varying over a wide range. For a small temperature range, this equation can be reduced to only two terms and rearranging gives:

$$\mu = e^{C_1 + \frac{C_2}{T}} \quad (3.8)$$

In this study, 46 ISO VG PAG oil was used. The kinematic viscosities of this PAG oil at two different temperatures, as obtained from ASHRAE Handbook (1998), are: $\nu_{40^\circ\text{C}} = 46 \times 10^{-6} \text{ m}^2 \text{ s}^{-1}$ and $\nu_{100^\circ\text{C}} = 7.732 \times 10^{-6} \text{ m}^2 \text{ s}^{-1}$. With this data and the density at these temperatures, the values of C_1 and C_2 is determined as $C_1 = -12.995$ and $C_2 = 3124.094$. Therefore the viscosity of PAG oil can be described as:

$$\mu_o = e^{-12.995 + 3124.094/(T + 273.15)} \quad (3.9)$$

where T is in °C.

3.1.3 Properties of refrigerant/oil mixture

In general, at the pressure existing inside the accumulator under normal operating conditions of an air conditioner, PAG oil can be regarded as only one liquid phase. In order to describe the properties of two-phase two-component (refrigerant and oil) flow, the following equations are used to define its composition:

$$x = \frac{\dot{m}_g}{\dot{m}_g + \dot{m}_f + \dot{m}_o} \quad (3.10)$$

$$c = \frac{\dot{m}_o}{\dot{m}_g + \dot{m}_f + \dot{m}_o} \quad (3.11)$$

$$w = \frac{\dot{m}_f}{\dot{m}_f + \dot{m}_o} \quad (3.12)$$

where, x is the refrigerant vapour quality, c is the oil concentration, w is the liquid refrigerant fraction, $\dot{m}_g$ is the mass flow rate of refrigerant vapour, $\dot{m}_f$ is the mass flow rate of liquid refrigerant, $\dot{m}_o$ is the mass flow rate of PAG oil.

In general, the vapour quality is defined as the ratio of mass flow rate of refrigerant vapour to mass flow rate of total refrigerant, where the effects of oil are neglected. In this work, the effects of oil are accounted for by defining the vapour quality according to Equation (3.10).

The liquid refrigerant fraction, w , can also be specified as a function of vapour quality, x , and oil concentration, c , by the equation:

$$w = \frac{1 - x - c}{1 - x} \quad (3.13)$$

The saturation temperature of refrigerant/oil mixture deviates from that of pure refrigerant. The composition of the liquid phase determines the saturation temperature of the entire two-phase mixture. Therefore, there is no constant saturation temperature over the entire two-phase range. A seven-term empirical model has been developed by

Grebner and Crawford (1992) to predict the dimensionless degree of superheat, T^* , for R-134a/PAG mixtures. This model is given by the equation:

$$T_{s-mix} = (1 + T^*) \cdot T_s \quad (3.14)$$

$$T^* = (1 - w) \cdot (E + FP) \quad (3.15)$$

$$E = a_1 + \frac{a_2}{w^{0.5}} \quad (3.16)$$

$$F = a_3 + \frac{a_4}{w^{0.5}} + \frac{a_5}{w} + \frac{a_6}{w^{1.5}} + \frac{a_7}{w^2} \quad (3.17)$$

The dimensionless degree of superheat for the R-134a/PAG mixture is a function of the saturation temperature of R-134a/PAG mixture, T_{s-mix} , and the saturation temperature of the pure refrigerant, $T_s(P)$, where P is the vapour pressure of the mixture. The constants a_1 through a_7 were empirically determined from a multivariate linear least squares regression for the mixture data. The values of these empirical constants are in the Table 3.1.

This model is accurate in predicting T^* for concentrations from 5 percent to 100 percent R-134a over the temperature range of -46°C to 121°C .

Table 3.1 Values of empirical constants

Empirical constants	Values
a ₁	-7.1524667E-2
a ₂	5.9851602E-2
a ₃	1.7133122E-3
a ₄	-2.9748674E-3
a ₅	1.7002776E-3
a ₆	-3.7562142E-4
a ₇	2.9984011E-5

The individual component density can be added together to determine the overall mixture density:

$$\rho_{mix} = \frac{1}{\frac{x}{\rho_g} + \frac{c}{\rho_o} + \frac{1-x-c}{\rho_f}} \quad (3.18)$$

where, ρ_g is the density of refrigerant vapour, ρ_f is the density of liquid refrigerant, ρ_o is the density of PAG oil, and ρ_{mix} is the density of R-134a/PAG mixture.

There are several ways to calculate the viscosity of a mixture of PAG oil and R-134a. Using results from measurements performed by Baustian et al. (1986), the following equation reproduces the mixture viscosity quite well, because it considers the large difference in the molecule size of PAG and R-134a ($M_{PAG}=2000 \text{ g mole}^{-1}$, $M_{R-134a}=102 \text{ g mole}^{-1}$):

$$\sqrt[3]{\mu_{mix}} = y_r \sqrt[3]{\mu_r} + y_o \sqrt[3]{\mu_o} \quad (3.19)$$

$$y_r = \frac{n_r}{n_r + n_o} \quad (3.20)$$

$$n_r = \frac{\dot{m}_r}{M_r}, n_o = \frac{\dot{m}_o}{M_o} \quad (3.21)$$

$$y_o = 1 - y_r \quad (3.22)$$

To calculate the viscosity of the two-phase R-134a, the following equation is used:

$$\frac{1}{\mu_r} = \frac{x}{(1-c)\mu_g} + \frac{1-x-c}{(1-c)\mu_f} \quad (3.23)$$

3.2 Stream Analysis Model for accumulator

A stream analysis model was developed to simulate the behaviour of accumulators and their influence on the automotive A/C systems. It allowed a comprehensive steady state simulation with a set of input conditions such as refrigerant vapour mass flow rate and pressure at the inlet of an accumulator. In this study, the refrigerant/oil mixture is R-134a/PAG oil, which are miscible, but could be any A/C refrigerant/oil, including carbon dioxide/oil. The model accounts for all major effects inside the accumulator, such as friction, bends, sudden expansion, sudden contraction, and heat exchange. The outputs are quality, pressure and temperature at various positions of accumulator.

3.2.1 Physical effects and mathematical description

3.2.1.1 Assumptions

In order to set up the Stream Analysis Model, the following assumptions were made:

- (1) System is operating at steady state.
- (2) No heat exchange with the environment.
- (3) The vapour bulk in the accumulator has a uniform pressure.

A test was done to investigate the influence of external heat transfer to A/C systems. The A/C system ran at conditions with and without heat transfer between accumulator and the surroundings. The results showed deviations of 2.72 percent for COPs and 2.35 percent for cooling loads. For practical purposes, it was acceptable to neglect the external heat transfer, so it can be assumed that there was no heat transfer between the accumulator and surroundings, and no liquid refrigerant evaporated inside accumulator.

3.2.1.2 Pressure drop due to sudden contraction and expansion

Whenever a fluid is forced to change its velocity instantaneously, a stagnation pressure drop occurs, leading to a loss in the sum of static pressure and dynamic pressure. However, one of these pressures may increase. A general equation to describe pressure drop due to sudden contraction and expansion is:

$$\Delta P = K \cdot \frac{1}{2} \rho V^2 \quad (3.24)$$

where, K is the loss coefficient. In general, K has a value of ca. 1.5 for sudden contraction and expansion, but there is a deviation for refrigerant/oil two-phase flow. An empirical value of 1.84 was obtained from experimental data using the equipment shown in Figure 4.2.

3.2.1.3 Pressure drop due to a bend

Swirling effects in bends cause additional losses. These losses can also be calculated using Equation (3.24) as for sudden contraction and expansion, but using a different value of K . The value of K is determined by bend angle, bend radius, pipe diameter, and roughness height of inner surface of pipe. In the case of pressure drop due to a bend, the dynamic pressure remains constant while the static pressure decreases.

3.2.1.4 Pressure drop due to friction

Turbulence and inner friction (shearing because of velocity profile) cause a pressure drop in pipes. The pressure drop depends on velocity, density, pipe dimensions (length and diameter), and the friction factor and was given by:

$$\Delta P = f \cdot \frac{1}{2} \rho V^2 \cdot \frac{L}{D} \quad (3.25)$$

The friction factor f may be defined as a function of the Reynolds number, Re :

$$Re = \frac{\rho V D}{\mu} \quad (3.26)$$

There are several equations to calculate friction factor f , but each one gives reasonable results only for a specific range of Reynolds numbers. The following equations are for the range of Reynolds number of flows inside accumulators.

Figure 3.1 A common layout for an accumulator

For $2320 \leq Re < 10^5$, f can be calculated by the Blasius equation (ASHRAE Handbook, 1993):

$$f = 0.316 \times Re^{-0.25} \quad (3.27)$$

For $10^5 \leq Re < 10^6$, f can be calculated by the following equation (ASHRAE Handbook, 1993):

Figure 3.2 Accumulator geometry

$$f = 0.0032 + \frac{0.221}{Re^{0.237}} \quad (3.28)$$

In general, these two equations are used to calculate the friction factor for single-phase flows, but according to Carey (NIST, 1998); these correlations can also be used for two-phase flow if it is turbulent. For typical operating range (the speed of compressor is between 700 rpm and 2000 rpm), calculations were done that show Reynolds numbers for the entire flow in the region above 30,000. The flow can be considered as turbulent, therefore fulfilling the condition to use the single-phase flow correlations.

3.2.2 Descriptions of accumulators

Accumulators are canisters used in a CCOT system and are found on the low-pressure side of the system between the evaporator outlet and the compressor inlet. They typically contain a filtering material and a desiccant bag to remove moisture and particles from the refrigerant/PAG oil mixture as well as a dye to make it easy to spot leaks in the system. Figure 3.1 shows a common layout for one kind of accumulator.

It was commonly believed that one main function of an accumulator is to protect the compressor from large quantities of liquid and to ensure a sufficient oil supply to the compressor. This is done through the use of an internal feature to separate the two phases, the liquid from the vapour, and a second feature that provides a path for the vapour to exit out of the accumulator, while the liquid remains stored inside. There is also a mechanism to allow a supply of oil to exit with the vapour to provide cooling and lubrication at the compressor. Figure 3.2 shows the real and simple geometry of an accumulator.

The accumulator is also thought to perform functions such as:

- 1) Separates the two-phase refrigerant mixture coming from the evaporator into its liquid and vapour components.
- 2) Acts as a reservoir for excess liquid refrigerant.
- 3) Removes moisture and debris from the refrigerant.
- 4) Returns oil to the compressor for lubrication and cooling.

Figure 3.3 Flow description

3.2.3 Flow description

Figure 3.3 shows the flow description of an accumulator. A two-phase flow enters the accumulator at the inlet tube. There are several effects that cause pressure drops inside the accumulator:

- 1) The friction and a bend in the inlet tube, sudden expansion and a bend from inlet tube to volume between top and reflector, and then

the sudden contraction and sudden expansion through the gap between the wall and reflector when the flow enters the accumulator vessel and its velocity drop down to almost zero.

- 2) The liquid accumulates and can only leave through the orifice or evaporation. The amount of evaporating liquid depends on the heat transfer through the wall.
- 3) The vapour leaves through the J-tube. At the inlet of the J-tube, the vapour suddenly contracts and the velocity has to increase from almost zero to the outlet velocity, which causes another pressure drop.
- 4) Part of the vapour enters the J-tube through the anti-siphon hole through a sudden contraction and expansion. The vapour from the first part of J-tube and anti-siphon hole combine inside J-tube near anti-siphon hole with the same pressure.

Furthermore, there are losses caused by friction and bends in the J-tube and the outlet pipe. At the inlet of the J-tube, there is friction because it takes time/distance to form a steady velocity profile. Depending on the losses in the first part of the J-tube (between inlet and orifice) and the height of the liquid, there is a variable pressure difference through the orifice. This causes a variable liquid mass flow rate leaving the accumulator. All these effects are a function of velocity and density, which are functions of mass flow rate, vapour quality, and pressure.

Figure 3.4 Flow streams in an accumulator

3.2.4 Flow streams in an accumulator

The flow streams inside an accumulator are illustrated in Figure 3.4. The points defined in Figure 3.4 are:

- A: Two-phase flow at inlet pipe.
- B: Vapour bulk in the container.
- C: Meet point of streams 2 and 3.
- D: Meet point of streams 4 and 5.
- E: Two-phase flow at outlet pipe.

Streams defined in Figure 3.4:

- 1: Two-phase flow stream from inlet pipe to vapour bulk in the container, $\dot{m}_1$.
- 2: Main vapour flow stream through the J-Tube, $\dot{m}_2$.
- 3: Liquid stream flowing through the pinhole at the bottom of J-Tube, $\dot{m}_3$.
- 4: Vapour flow stream through the anti-siphon hole, $\dot{m}_4$.
- 5: Liquid and vapour mixed flow stream (stream 2 + stream 3).
- 6: Liquid and vapour mixed flow stream (stream 4 + stream 5).

Figure 3.5 Pressure drop network of the accumulator

Figure 3.6 The dimensions of the accumulator used

A pressure drop network of the flows inside accumulators can be expressed in Figure 3.5. At a same junction the pressures for different streams are the same. Based on this principle, we have:

$$\Delta P_2 = \Delta P_3 \quad (3.29)$$

$$\Delta P_2 + \Delta P_5 = \Delta P_4 \quad (3.30)$$

3.2.5 Pressure drop along streams

3.2.5.1 Stream 1

Figure 3.6 gives the dimensions of the accumulator that will be used in the following equations. The stream 1 is the flow between point A and B, in which the flow is two-phase. At point A, the total mass flow rate, $\dot{m}$, and the inlet static pressure, P_A , are input values, and the oil concentration, c , is assumed constant. The stagnation pressure, $P_{\text{stag-A}}$, is the sum of the static pressure, P_A , and dynamic pressure, $P_{\text{dyn-A}}$.

$$P_{\text{dyn-A}} = \frac{1}{2} \rho_A V_A^2 \quad (3.31)$$

$$P_{\text{stag-A}} = \frac{1}{2} \rho_A V_A^2 + P_A \quad (3.32)$$

where, ρ_A is the density of mixture and V_A is the average velocity at point A.

$$\frac{1}{\rho_A} = \frac{x}{\rho_v} + \frac{c}{\rho_o} + \frac{1-x-c}{\rho_f} \quad (3.33)$$

$$V_A = \frac{4\dot{m}}{\pi \rho_A D_{out}^2} \quad (3.34)$$

From point A to point B, the stagnation pressure drops due to several effects. Total stagnation pressure drops include: ΔP_{1-1} due to a 90° bend in the inlet tube, ΔP_{1-2} due to friction in inlet pipe, ΔP_{1-3} due to sudden expansion and ΔP_{1-4} due to 90° bend from inlet tube to volume between top and reflector, ΔP_{1-5} due to sudden contraction and sudden expansion through the gap between the wall and reflector when the flow enters the accumulator vessel.

$$\Delta P_{1-1} = K_{1-1} \cdot \frac{1}{2} \rho_A V_A^2 \quad (3.35)$$

where, K_{1-1} is determined by the ratio of bend radius to pipe diameter and the ratio of material roughness to pipe diameter. For current accumulator, $K_{1-1} = 0.24$.

$$\Delta P_{1-2} = f_{1-2} \cdot \frac{1}{2} \rho_A V_A^2 \cdot \frac{L_{in}}{D_{out}} \quad (3.36)$$

where, f_{1-2} is determine by Re_{1-2} .

$$Re_{1-2} = \frac{\rho_A V_A D_{out}}{\mu_A} \quad (3.37)$$

Then f_{1-2} can be calculated using Equation (3.27) or Equation (3.28).

$$\Delta P_{1-3} = K_{1-3} \cdot \frac{1}{2} \rho_A V_A^2 \quad (3.38)$$

In general pipe flow, as shown in Figure 3.7, the loss coefficient for sudden expansion can be calculated by the following equation:

$$K = \left[1 - \left(\frac{d}{D} \right)^2 \right]^2 \quad (3.39)$$

In above equation, $(d/D)^2 = A_d/A_D$, where A_d and A_D are the areas of the two pipes.

For sudden expansion in accumulator, $A_d = \pi d_{out}^2/4$, and $A_D = \pi d_{out} h_{ref}$. So K_{1-3} can be calculated by the following equation:

$$K_{1-3} = \left(1 - \frac{d_{out}}{4h_{ref}} \right)^2 \quad (3.40)$$

$$\Delta P_{1-4} = K_{1-4} \cdot \frac{1}{2} \rho_A V_A^2 \quad (3.41)$$

where, ΔP_{1-4} is the pressure drop due to 90° bend from inlet tube to the volume between the top and the reflector, the loss efficient K_{1-4} takes a value of 0.24.

$$\Delta P_{1-5} = K_{1-5} \cdot \frac{1}{2} \rho_A V_{1-5}^2 \quad (3.42)$$

Figure 3.7 Sudden expansion in pipe flow

Figure 3.8 Sudden expansion in accumulator

where, ΔP_{1-5} is the pressure drop due to the sudden contraction and expansion through the gap between the wall and the reflector when the flow enters the accumulator vessel, $K_{1-5}=1.5$.

$$V_{1-5} = \frac{4\dot{m}}{\rho_A \pi (D^2 - d^2)} \quad (3.43)$$

$$P_{stag-B} = P_{stag-A} - \Delta P_{1-1} - \Delta P_{1-2} - \Delta P_{1-3} - \Delta P_{1-4} - \Delta P_{1-5} \quad (3.44)$$

At point B, the flow velocity can be regarded as zero, so the dynamic pressure is zero, and the static pressure equals stagnation pressure.

$$P_B = P_{stag-B} \quad (3.45)$$

3.2.5.2 Stream 2

Part of vapour contracts to enter the J-tube along stream 2 and accelerates from zero to the velocity in the J-tube. The rest of vapour contracts to enter the J-tube along stream 4 through the anti-siphon hole. The ratio of vapour flow rate along stream 2 to the total vapour flow rate is r_2 . The total stagnation drops along stream 2 include: ΔP_{2-1} due to sudden contraction, ΔP_{2-2} due to friction in first part of J-tube and ΔP_{2-3} due to a 90° bend.

$$V_2 = \frac{4r_2 r_v \dot{m}}{\pi \rho_v D_j^2} \quad (3.46)$$

$$\Delta P_{2-1} = K_{2-1} \cdot \frac{1}{2} \rho_v V_2^2 \quad (3.47)$$

where, $K_{2-1}=0.5$.

$$\Delta P_{2-2} = f_{2-2} \cdot \frac{1}{2} \rho_v V_2^2 \cdot \frac{L_1}{D_j} \quad (3.48)$$

where, the determination of f_{2-2} is same as that of f_{1-1}

$$\Delta P_{2-3} = K_{2-3} \cdot \frac{1}{2} \rho_v V_2^2 \quad (3.49)$$

where, K_{2-3} is the loss efficient of 90° bend, and its value is 0.15 by checking the table.

$$P_{stag-C} = P_{stag-B} - \Delta P_{2-1} - \Delta P_{2-2} - \Delta P_{2-3} \quad (3.50)$$

$$P_{dyn-C} = \frac{1}{2} \rho_v V_2^2 \quad (3.51)$$

$$P_C = P_{stag-C} - P_{dyn-C} \quad (3.52)$$

3.2.5.3 Stream 3

In stream 3, from point B to C, there is a pressure increase ΔP_{3-1} due to the liquid height, and then a pressure drop ΔP_{3-2} due to a sudden contraction and expansion through the oil bleeding hole. The velocities of liquid are zero in the accumulator vessel and inside the J-tube at point C.

$$V_3 = \frac{4(1-r_v)\dot{m}}{\pi \rho_{f-o} d_{ori}^2} \quad (3.53)$$

$$\frac{1-r_v}{\rho_{f-o}} = \frac{1-r_v-r_o}{\rho_f} + \frac{r_o}{\rho_o} \quad (3.54)$$

$$\Delta P_{3-1} = -\rho_{l-o} g h_{liq} \quad (3.55)$$

$$\Delta P_{3-2} = K_{3-2} \cdot \frac{1}{2} \rho_{f-o} V_3^2 \quad (3.56)$$

where, $K_{3-2}=1.5$. The pressure at the bottom of accumulator is:

$$P_{bot} = P_{stag-B} - \Delta P_{3-1} \quad (3.57)$$

The static pressure at point C is:

$$P_C = P_{stag-B} - \Delta P_{3-1} - \Delta P_{3-2} \quad (3.58)$$

The pressure drops through stream 2 and stream 3 are equal:

$$\Delta P_{2-1} + \Delta P_{2-2} + \Delta P_{2-3} + P_{dyn-C} = \Delta P_{3-1} + \Delta P_{3-2} \quad (3.59)$$

3.2.5.4 Stream 4

Along stream 4, from point B to point D, there is a pressure drop due to asudden contraction and expansion through the anti-siphon hole.

$$V_4 = \frac{4(1-r_2)r_v \dot{m}}{\pi \rho_v D_{siph}^2} \quad (3.60)$$

$$\Delta P_{4-1} = K_{4-1} \cdot \frac{1}{2} \rho_v V_4^2 \quad (3.61)$$

where, $K_{4-1}=1.5$.

$$P_D = P_B - \Delta P_{4-1} \quad (3.62)$$

3.2.5.5 Stream 5

At point C, after mixing the vapour and liquid-oil, the static pressure remains constant, but the dynamic pressure changes, because the velocity and average density change.

$$\frac{1-r_{siph}}{\rho_5} = \frac{r_v-r_{siph}}{\rho_v} + \frac{r_o}{\rho_o} + \frac{1-r_v-r_o}{\rho_f} \quad (3.63)$$

$$V_5 = \frac{4(1 - r_{siph})\dot{m}}{\pi \rho_5 D_j^2} \quad (3.64)$$

Along stream 5, the dynamic pressure keeps constant, but static pressure decreases. The total pressure drops include: ΔP_{5-1} due to a 90° bend and ΔP_{5-2} due to friction through the second part of J-tube.

$$\Delta P_{5-1} = K_{5-1} \cdot \frac{1}{2} \rho_c V_5^2 \quad (3.65)$$

where, $K_{5-1}=0.15$.

$$\Delta P_{5-2} = f_{5-2} \cdot \frac{1}{2} \rho_c V_5^2 \cdot \frac{L_2}{D_j} \quad (3.66)$$

$$P_D = P_C - \Delta P_{5-1} - \Delta P_{5-2} \quad (3.67)$$

The combination of Equation (3.62) and Equation (3.67) give:

$$P_B - \Delta P_{4-1} = P_C - \Delta P_{5-1} - \Delta P_{5-2} \quad (3.68)$$

3.2.5.6 Stream 6

At point D, after stream 4 and stream 5 join, the static pressure remains constant, but the dynamic pressure changes, because the velocity and average density change. The average density at point D is same as at point A, because they have the same vapour quality when the heat transfer between accumulator and its surroundings is ignored.

$$\rho_D = \rho_A \quad (3.69)$$

$$V_{6-1} = \frac{\dot{m}}{\pi \rho_D D_j^2} \quad (3.70)$$

$$P_{dyn-D} = \frac{1}{2} \rho_D V_{6-1}^2 \quad (3.71)$$

$$P_{stag-D} = P_D + P_{dyn-D} \quad (3.72)$$

From points D to E, the total pressure drops include: ΔP_{6-1} due to friction through l_3 , ΔP_{6-2} due to friction through l_{out} , and ΔP_{6-3} due to a 90° bend.

$$\Delta P_{6-1} = f_{6-1} \cdot \frac{1}{2} \rho_D V_{6-1}^2 \cdot \frac{L_3}{D_j} \quad (3.73)$$

$$V_{6-2} = V_A \quad (3.74)$$

$$\Delta P_{6-2} = f_{6-2} \cdot \frac{1}{2} \rho_D V_{6-2}^2 \cdot \frac{L_{out}}{D_{out}} \quad (3.75)$$

$$\Delta P_{6-3} = K_{6-3} \cdot \frac{1}{2} \rho_D V_{6-2}^2 \quad (3.76)$$

where, $K_{6-3}=0.24$.

$$P_{stag-E} = P_{stag-D} - \Delta P_{6-1} - \Delta P_{6-2} - \Delta P_{6-3} \quad (3.77)$$

$$P_{dyn-E} = \frac{1}{2} \rho_D V_{6-2}^2 \quad (3.78)$$

$$P_E = P_{stag-E} - P_{dyn-E} \quad (3.79)$$

3.2.6 MATLAB program

For the program in this work, a set of input parameters that fully describes the problem is defined. First, these are the geometric variables and the dimensions of the accumulator. Then, the vapour mass flow rate, the pressure at accumulator inlet, the oil concentration, which is the ratio of oil flow rate to total mass flow, and the liquid height inside accumulator are taken as input parameters.

The solution of the mathematical model involves a trial and error procedure. It proceeds from the evaluation of vapour quality and ratio of vapour mass flow rate through stream 2, to total vapour mass flow rate, to the calculation of pressure drops and Equations (3.59) and (3.68). At last, the true value of vapour quality is obtained when Equations (3.59) and (3.68) are fulfilled.

From all inputs and the vapour quality, other outputs such as the total mass flow rate, the pressures and temperatures at various positions in accumulator, and the pressure drop through accumulator can be calculated. The flow chart for the accumulator model (SAM) is shown in Figure 3.9.

Figure 3.9 Flow chart for accumulator model (SAM)

3.3 Investigation of compressor performance

3.3.1 Thermodynamic properties

The oil in the mixture can be regarded always as liquid phase; therefore, the enthalpy is only dependent on the temperature of the oil. The following equation is from Grebner and Crawford (1992):

$$c_p (\text{Btu/lb} \cdot \text{F}) = 0.48 + 0.0003 \cdot T(^{\circ}\text{F}) \quad (3.80)$$

where, the unit of c_p is $\text{Btu lb}^{-1} \text{F}^{-1}$ and the unit of T is F .

The enthalpy can be calculated from the following equation:

$$h = \int_{T_0}^T c_p dT \quad (3.81)$$

The reference point is at 0°C and 200000 J kg^{-1} and is consistent with the refrigerant properties. Equations (3.80) and (3.81) combined and applied to the reference point lead to the equation for the PAG enthalpy; the units were converted to SI unit system:

$$h = 200 + 2.1 \cdot T + 1.13 \times 10^{-3} \cdot T^2 \quad (3.82)$$

where the unit of h is kJ kg^{-1} , and the unit of T is $^{\circ}\text{C}$.

The density of PAG oil is only dependent on temperature, and the following equation is developed by Grebner and Crawford (1992):

$$\rho_o = 1004.054 - 0.716934 \cdot T \quad (3.83)$$

The entropy of PAG oil can be expressed as:

$$s_o = 1.2256 \times s_f \quad (3.84)$$

The individual component enthalpies can be added together to determine the overall mixture enthalpy and density:

$$h_{mix} = x \cdot h_g + (1 - x - c) \cdot h_f + c \cdot h_o \quad (3.85)$$

$$s_{mix} = x \cdot s_g + (1 - x - c) \cdot s_f + c \cdot s_o \quad (3.86)$$

$$\rho_{mix} = \frac{1}{\frac{x}{\rho_g} + \frac{c}{\rho_o} + \frac{1-x-c}{\rho_f}} \quad (3.87)$$

where,

h_g = enthalpy of refrigerant vapour (kJ kg^{-1}),

h_f = enthalpy of liquid refrigerant (kJ kg^{-1}),

h_o = enthalpy of PAG oil (kJ kg^{-1}),

h_{mix} = enthalpy of R-134a/PAG mixture (kJ kg^{-1}).

s_g = entropy of refrigerant vapour ($\text{kJ K}^{-1} \text{kg}^{-1}$),

s_f = entropy of liquid refrigerant ($\text{kJ K}^{-1} \text{kg}^{-1}$),

s_o = entropy of PAG oil ($\text{kJ K}^{-1} \text{kg}^{-1}$),

s_{mix} = entropy of R-134a/PAG mixture ($\text{kJ K}^{-1} \text{kg}^{-1}$).

ρ_g = density of refrigerant vapour (kg m^{-3}),

ρ_f = density of liquid refrigerant (kg m^{-3}),

ρ_o = density of PAG oil (kg m^{-3}),

ρ_{mix} = density of R-134a/PAG mixture (kg m^{-3}).

The thermodynamic properties of R-134a are taken from NIST database (1998).

3.3.2 Effects of vapour quality and oil concentration

In a refrigeration cycle, the compression process is the process by which the low pressure, low temperature, saturated or superheated refrigerant vapour is compressed to a high pressure, high temperature and superheated vapour. Liquid refrigerant and oil are neglected in the compression process, which is acceptable when vapour quality is very high and oil concentration is very low. However, in some refrigeration cycles, such as automotive A/C systems, in which vapour quality is usually 80~90 percent and oil concentration is about 5 percent, liquid refrigerant and oil in circulation (OIC) need to be considered in the compression process for higher accuracy.

In a refrigeration cycle with two-phase flow and OIC considered, the compression process occurs when the low pressure, low temperature, lower quality refrigerant/oil two-phase mixture compresses to high pressure, high temperature, higher quality refrigerant/oil two-phase mixture. At the inlet of the compressor, the relationship of pressure, P_{in} , liquid refrigerant fraction, w_{in} , and the saturation temperature of mixture, T_{in} , are given by Equations (3.14) and (3.15). During the compression process, the pressure and temperature increase, and the liquid refrigerant evaporates partially. The evaporation of liquid refrigerant causes a decrease in the liquid refrigerant fraction, but there is always some liquid refrigerant left due to the existence of PAG oil. If the compression process is slow enough, the final state will reach a saturated state, in which the relationship of pressure, P_{out} , liquid refrigerant fraction, w_{out} , and the saturation temperature of mixture, T_{out} , are given by Equations (3.14) and (3.15); w_{out} is less than w_{in} due to the evaporation of the liquid refrigerant. However, at normal speed of compressor, there is not enough time for liquid refrigerant to evaporate. The compression process cannot reach saturated state, and the actual liquid refrigerant fraction is always between w_{in} and w_{out} .

Figure 3.10 shows the influence of the two-phase flow and OIC on the refrigeration cycle of an A/C system, especially the compression process. In Figure 3.10(a), without consideration of two-phase flow and oil concentration, the saturated or slightly superheated refrigerant vapour compresses to superheated vapour. In part (b), with consideration of two-phase flow and without consideration of oil concentration, two-phase refrigerant compresses to superheated vapour. In part (c), with consideration of

both two-phase flow and oil concentration, the dashed curve is for pure R-134a, and the solid curve illustrates the saturation curve for R-134a/PAG mixture at an assumed liquid refrigerant fraction, which could be determined by Equations (3.14) and (3.15). Figure 3.10 (c) shows that the refrigerant is two-phase at both inlet and outlet of compressor because of the existence of oil.

3.3.3 Efficiencies of the compressor

Compressor performance is the result of design compromises involving physical limitations of the refrigerant, the compressor itself, and the motor. The actual compressor performance can be measured by volumetric efficiency, compression efficiency, mechanical efficiency, and isentropic efficiency. In current literature, volumetric efficiency and isentropic efficiency are modelled as functions of compression ratio and compressor speed, ignoring two-phase flow and OIC. The following set of calculations, coupled with experimental data, show the effects of vapour quality on the volumetric efficiency and isentropic efficiency.

Volumetric efficiency, e_v , is the ratio of actual volume of gas entering the compressor to the geometric displacement of the compressor, which can be calculated by the following equation:

$$e_v = \frac{\dot{m}_1 / \rho_{mix}}{\dot{V}_d v} \quad (3.88)$$

(a)

(b)

(c)

Figure 3.10 P-h diagram of refrigeration cycle: (a) two-phase flow neglected; (b) two-phase flow considered; (c) two-phase flow and oil concentration considered.

where, $\dot{m}_t$ is the total mass flow rate, ρ_{mix} is the density of mixture from Equation (3.87), $V_d (\text{m}^3 \text{ rev}^{-1})$ is the compressor displacement, and v (rpm) is the compressor speed.

Compression efficiency, e_c , is the ratio of the work required for isentropic compression of the gas to the work delivered to the gas with the compression volume, and it is a measure of the deviation of actual compression from isentropic compression.

Mechanical efficiency, e_m , is the ratio of power delivered to the gas to the power input on the compressor shaft, and it is a measure of losses resulting from the mechanical friction in the compressor. According to Dossat (1991), the mechanical efficiency of current swash plate compressor is about 90 percent.

Isentropic efficiency, e_i , is the ratio of power required for isentropic compression of the mixture to the power input on the compressor shaft, which can be calculation by the following equation:

$$e_i = \frac{W_i}{W_{in}} \quad (3.89)$$

where, W_{in} (kW) is the power input on the compressor shaft, which is measured by experiment. W_i (kW) is the power required for the isentropic compression of the mixture, which is calculated from experimental data by equation:

$$W_i = \dot{m}_{\text{mix}} (h_{\text{mix-2s}} - h_{\text{mix-1}}) \quad (3.90)$$

The determination of point 2s is easier if two-phase and oil concentration effects are ignored. P_1 and T_1 are measured by experiment, and then $s_{\text{mix-1}}$ is determined. T_{2s} can be determined by P_1 and $s_{\text{mix-2s}}$ (equal to $s_{\text{mix-1}}$). However, with consideration of two-phase and oil concentration, the determination of point 2s is more complicated, since there are liquid refrigerant, vapour refrigerant, and oil at point 2s. The determination of point 2s is by solving two equations:

$$s_{\text{mix-1}} = s_{\text{mix-2s}} \quad (3.91)$$

$$T_{2s} = (1 + T_{2s}^*) \cdot T_{\text{sat}} \quad (3.92)$$

where, Equation (3.92) is the model of Grebner and Crawford (1992), in which T_{2s}^* is calculated by Equations (3.15), (3.16) and (3.17). By solving Equations (3.91) and (3.92), the temperature T_{2s} , and mass flow rate of liquid, vapour refrigerant and oil are determined. A Matlab program was developed to solve the equations, which is in the appendix. The flow chart for this program is shown in Figure 3.11.

From the definitions of e_c , e_m and e_i , their relationship is as the following equation:

$$e_i = e_c e_m \quad (3.93)$$

For an A/C system, the performance of compressor can be described by volumetric efficiency and isentropic efficiency. The compression efficiency and mechanical efficiency are represented by the isentropic efficiency. Therefore, only volumetric efficiency and isentropic efficiency are studied in this work.

Figure 3.11 Flow chart for the program determining point 2s

Chapter 4 Experiment

This work details the investigation of a CCOT automotive A/C system. The experimental setup, Automotive Air Conditioning Test System (AACTS), is shown in Figure 4.1, which includes a compressor, a condenser, an expansion device, an evaporator and an accumulator, and the schematic illustration is shown in Figure 4.2. It is a closed refrigeration loop charged with R-134a as a working fluid and Polyalkylene Glycol (PAG) oil as the lubricant for the compressor. The main components are a compressor stand, a water-cooled condenser stand, an evaporator stand, two mass flowmeters, an accumulator stand, a separator, and various process instruments. These blockwise stands enable easy access for servicing and for changing system components (Halla, 2002).

Figure 4.1 The automotive air-conditioning test system

Figure 4.2 Schematic illustration of the experimental setup

4.1 Experimental setup

4.1.1 The compressor

The compressors primary function is to compress and pressurize gaseous refrigerant. However, in all systems the compressor needs lubrication and cooling and this is accomplished by returning some liquid and oil back to the compressor. The compressor sucks in cool, low pressure refrigerant from the accumulator into its inlet or suction port, compresses it into a hot pressurized gas, and pushes it out the outlet or discharge port to the condenser. Table 4.1 shows the compressor information.

An electromagnetic clutch powers the compressor by engaging or disengaging the compressor pulley. The speed of engine is adjustable and controlled by taking the input signal from a data acquisition system. This pulley always turns when the engine is running, but the compressor only runs when the pulley engages to the compressor drive shaft via the armature plate and hub assembly that are fastened to the compressor drive shaft. When the A/C system is on, current flows through the solenoid in the clutch; this magnetically clamps the armature plate against the side of the turning pulley and thereby drives the compressor. When the A/C is turned, the current stops flowing through the electromagnetic coil and the armature plate releases from the pulley. When it is not driving the compressor, the clutch pulley turns on a double row of ball bearings.

Two type K thermocouples and absolute pressure transducers were used to measure the temperature and pressure at the inlet and outlet of the compressor. The compressor speed was measured by a speed sensor with a reading LCD right on the sensor, and output to the data acquisition system. A power sensor, Universal Power Cell (Model UPC) by Load Controls Inc., was used to measure the motor power, which is the power input to the compressor.

Table 4.1 Compressor information (displacement is 170 cc/rev)

Speed	Discharge pressure	Suction		Power input	Torque input	Effective volume	Volumetric efficiency
		Pressure	Temperature				
rpm	P (kPa)	P (kPa)	T (°C)	kW	N-m	cc/rev	%
1188	1853.41	253	17.78	1.13	15.83	108.8	64
1000	2370.16	310	18.33	1.19	20.37	108.8	64
3000	1694.94	138	2.78	1.16	12.85	76.5	45

4.1.2 The sub-cooled condenser

The condenser functions as a heat exchanger as it removes the heat collected from the inside of the car, gathered in the liquid refrigerant, and places it into the outside environment. The condenser location is usually in front of the radiator, as it requires good airflow. This is usually accomplished by taking advantage of the engine cooling fan. As the hot compressed refrigerant gas enters the top of the condenser, the air flowing over the condenser cools the gas and condenses it into a high-pressure liquid, which exits at the bottom. However, in a real A/C system, the condenser may not fully condense the refrigerant, and instead a two-phase flow exits. To determine the actual flow of refrigerant in the system using a flowmeter, the mixture must be single phase. Therefore in order to obtain 100 percent liquid at the exit of the condenser, a sub-cooled condenser was built.

The sub-cooled condenser is in a tank filled with water. The tank is equipped with a stirrer, a liquid level switch, a type K thermocouple, an electrically actuated solenoid water valve, and a water pump. A desired water bath temperature was manually set, and the temperature was kept at this value through the opening and closing of the valve and pump.

Figure 4.3 Condenser coil

4.1.3 The expansion device

The expansion device in this test system is an inlet control valve located directly before the evaporator. The warm liquid from the condenser is rapidly expanded into a cold low-pressure two-phase mixture. This phase change occurs because the liquid is forced to suddenly contract and expand causing some of liquid to change into a gas state. A thermocouple and a pressure sensor were used to measure the temperature and pressure at upstream of the expansion valve.

Figure 4.4 Accumulator stand

4.1.4 The evaporator

The evaporator stand consists of a long tube or coil that goes back and forth through a multitude of cooling fins usually mounted inside the passenger compartment. The evaporator removes the heat from the inside of the vehicle. A secondary function of the evaporator is dehumidification. Refrigerant enters the bottom of the evaporator as a low-pressure two-phase flow. As the warm air in the interior of the car blows through the evaporator fins, the cold low-pressure refrigerant moving through the evaporator absorbs the heat from the air and vaporizes. The refrigerant releases heat to the outside of

the vehicle. Because the heat has been removed from the inside air, the air is now cooled and gets distributed with a blower fan around the interior of the vehicle, which cools it off.

In the test system, the evaporator stand consists of an automotive evaporator, an electric heater, an air-blower to circulate the air inside the duct to simulate environmental conditions, a temperature controller. A type K thermocouple is between the heater and the evaporator to measure the actual temperature. A power sensor (Load Controls Inc. UPC) measures the power input to the heater. The well-insulated evaporator stand minimizes heat losses. This ensures that the power input from heater equals to the cooling load. The COP can be calculated by dividing the cooling load by power input to the compressor. Inside the evaporator, the refrigerant absorbs heat from the outside air and liquid refrigerant evaporates. Two type K thermocouples and pressure sensors measure the temperatures and pressures of refrigerant-PAG oil mixture at both the inlet and outlet of the evaporator.

4.1.5 The accumulator

Accumulators are canisters in a CCOT system and are on the low-pressure side of the system between the evaporator outlet and the compressor inlet. They typically contain a filtering material and a desiccant bag to remove moisture and particles from the refrigerant/PAG oil mixture, as well as dye to make it easy to spot leaks in the system. The accumulator stand consists of an accumulator, an electric heater to simulate environment temperature, a temperature controller to keep the air at a desired temperature, and an insulated box to contain these components. A sight glass tube placed at the outlet

of accumulator allows the observation of the refrigerant/PAG oil flow (two-phase). Figure 4.4 shows the accumulator stand.

4.1.6 Vapour quality measurement device

An aluminium pipe, 1.2 m long and 52 mm in diameter, placed downstream of the sight glass tube, allows the annular flow to slow down and separate. The flow separates completely in the liquid/vapour separator. The separator is vertically, thus the vapour rises to the top and flows out through the upper outlet, and the liquid stay at the bottom and flows out through the lower outlet. The refrigerant vapour flows through the mass flowmeter where the gas flow rate can be measured. A sight glass tube placed on the outlet side of the mass flowmeter enables an observer to confirm that the flowmeter chamber fills with vapour refrigerant only. The vapour refrigerant and liquid refrigerant/PAG oil mix again at the inlet of compressor.

Another mass flowmeter installed at the condenser outlet measures the total mass flow rate of the refrigerant/PAG oil flow. The vapour quality, the ratio of gas mass flow rate to total mass flow rate, can be calculated. A sight glass tube placed at the outlet of the flowmeter confirms that the flowmeter chamber fills with sub-cooled liquid refrigerant/PAG oil single phase.

4.1.7 Oil in circulation

In order to measure the Oil In Circulation (OIC), percent of oil mass flow rate to total mass flow rate, four OIC canisters were used. The empty sample canisters were

weighed and recorded before the samples were taken. Four samples were taken from the high side port after the condenser using the OIC canisters. The sample canisters were re-weighed and recorded. Then the refrigerant was recovered from the canisters slowly (two to five hours), to avoid recovery of the oil. The sample canisters were re-weighed again and recorded. The difference of weights of empty canisters and canisters after sampling equalled total weight of refrigerant and oil. The difference of weights of empty canisters and canisters after recovery equalled the weight of oil. The ratio of oil weight to total weight equalled the OIC.

4.1.8 Data acquisition and processing device

To conduct data acquisition and processing, control the operation of A/C system, a control and measurement system was set up, which comprised of thermocouples, sensors, transducers, data acquisition system, signal conditioning extensions for instrumentation devices, and a computer with LabVIEW software. Using this system, the temperature and pressure at each component, mass flow rate, power, and compressor speed are measured and displayed on LabVIEW windows directly or further processed to obtain other parameters. All the data can be recorded to an Excel file in the interval up to once a second. This system can also control the operation of the A/C system, such as switching the power on or off, setting the temperature of the evaporator and the condenser, and adjusting the compressor speed. Two manometers are at the suction and discharge of the compressor to show the gauge pressure at these two locations. Table 4.2 shows all the values the AACTS can measure.

Table 4.2 AACTS stand measurables

A/C components	Temperatures	Pressures
	Ambient	
Condenser (subcooled)		
	Water temp	
Evaporator		
	Chamber temp	Pressure before orifice
	Temp in	Press in
	Temp out	Press out
Compressor		
	Temp in	Press in
	Temp out	Press out
	Skin Body temp	
Accumulator		
	Temp in	Differential pressure (Inlet to outlet)
	Temp out	Press out
	Top temp	Press top
	Bottom temp	Press bottom
	Gas temp	Press orifice
	Liquid temp	
	Under deflector temp	
Separator		
	Liquid temp	
	Gas temp	
Total mass flow		
Gas flow rate		
Quality out of accumulator		
Compressor rpm		
Accumulator box heater power		
Evaporator heater power		
Compressor power		
Cop		
Oil in circulation		
Liquid Height in the Accumulator		

4.2 Experimental facility improvement

The experiment equipment AACTS was originally designed and set up by Halla Climate Control Canada Inc. The goal was to gain a better understanding of how accumulators work and what their effect was on system performance, to compare the relative performance of different accumulators (or the same accumulator with different internal features), to provide valuable information about the performance of their products versus those of their competitors, and to show the relative performance of new designs compared with existing ones. In order to do further experimental research and simulation on the accumulator, the compressor and the system, some change to AACTS had been made by the author, or by Halla under the suggestion of the author.

4.2.1 The accumulator

In the original AACTS, the accumulator is a real product of Halla Climate Control Inc., and fixed sizes and shape, which was suitable for the test of new accumulators, but not enough for the research on the performance of accumulators. Under the author's suggestion, Halla made a test accumulator with a detachable cover, with a variable size of J-shape and oil bleed hole. Figure 4.5 shows the accumulator, and Figure 4.6 shows the J-shape tube inside the accumulator.

Figure 4.5 Detachable accumulator

4.2.2 Change of expansion device

In the original AACTS, the expansion device is an adjustable valve, which is easy to change the opening size of expansion. However, the expansion process through an adjustable valve is far from that through an orifice tube used in automotive A/C system. Orifice tubes with variable diameters replaced the adjustable valve in this experimental research.

Figure 4.6 J-shape tube

4.2.3 Compressor power input measurement

In original AACTS, the power input value measured by the power sensor includes energy consumption values of both the compressor and the motor. In the calculation of COP, the power input should be only power input to the compressor. The power consumption of motor only was measured through experiment me by switching the compressor off at variable compressor speed. Then, a revision to LabVIEW made the test results more accurate.

4.2.4 Heat lost through evaporator chamber and accumulator box

To neglect the heat lost through the walls of the evaporator chamber and the accumulator box, their walls were well insulated. In order to make measurement more accurate, the author measured the heat lost of the evaporator chamber and accumulator box, and modified the calculation.

4.2.5 Design and building of ACTSLHX

In the research of automotive A/C systems with internal heat exchanger (IHX), the author designed a new kind of IHX, named accumulator-capillary tube-suction line heat exchanger (ACTSLHX). The ACTSLHX combines an accumulator and a capillary tube. The capillary tube is a helical shape, and fits inside the J tube of the accumulator, to function as an expansion device and an internal heat exchanger.

4.3 Graphical user interface

This section contains a short manual for the program and a description of the graphical user interface.

When the user starts the program, the main screen appears and gives the choice of the different input and output modes and between steady state and transient mode, by selecting the corresponding tab.

Figure 4.7 Stand controls window

Figure 4.8 Stand measurement window

4.3.1 Stand controls window

The first screen is for stand controls, as shown in Figure 4.7, which includes the following control buttons:

Compressor drive controls: motor on/off

Refrigeration loop controls: clutch on/off

Evaporator controls: set evaporator heater power (constant power or constant temperature), evaporator mixer on/off

Condenser controls: set condenser water temperature, condenser mixer on/off

Set compressor rpm: set compressor speed until a limit (pressure cutout)

Accumulator stand controls: heater on/off, fan on/off

4.3.2 Stand measurement window

The second screen is for stand measurements, as shown in Figure 4.8, which includes the following control buttons:

State switch: transient state or steady state

Data view: view on screen only or view data and write to file

Power values: compressor, cooling load, accumulator heater power, and COP

Accumulator: pressure and temperature data display

Separator: pressure and temperature data display

Compressor: pressure, temperature and speed data display

Evaporator: pressure and temperature data display

Condenser: pressure and temperature data display

Time recorder: elapsed time since start and reset counter time

Figure 4.9 A/C system window

4.3.3 A/C system window

The third screen shows the A/C system with the values measured simultaneously, , as shown in Figure 4.9, which include temperature, pressure, power input, cooling load, and total and gas mass flow rate of all components of the systems.

4.4 Experimental uncertainty

From experimental test runs, the heat transfer rate between the evaporator chamber and the environment is less than 0.6 percent of the cooling capacity at same conditions. The estimated uncertainty in the temperature data collected by the data acquisition system using type K thermocouples is $\pm 0.2^{\circ}\text{C}$; pressure sensors, ± 10 kPa; Coriolis mass flowmeters, ± 0.1 kg/h; refrigerant charge, ± 0.0001 kg, compressor power and cooling capacity, $\pm 0.5\%$ of full scale. The thermostats provided precise control of the desired constant temperatures, to within $\pm 1^{\circ}\text{C}$.

Chapter 5 Results and discussion

5.1 Accumulator model validation and results

A Stream Analysis Model to simulate the functions of accumulators in automobile A/C systems has been developed, and experiment has been done to validate the model. The results of experimental and theoretical study show a new function of the accumulator, determining the vapour quality (two-phase flow) at compressor inlet, which thus affects the behaviour of automotive A/C systems. Based on the research on the accumulator, a design method has been developed with computational simulation program for industry.

Figure 5.1 Simulated and experimental values of vapour quality at variable diameter of oil bleed hole with $v = 700$ rpm and $v = 2000$ rpm

5.1.1 Model validation

The simulated and experimental values of vapour quality at variable diameter of oil bleed hole shown in Figure 5.1 for compressor speeds of 700 rpm and 2000 rpm, respectively. This figure shows that the simulated results by the stream analysis model of the accumulator agree well with the experimental data. The deviations between the experimental and simulated vapour quality are less than 5 percent.

The simulated and experimental values of pressure drop through the accumulator graphed in Figure 5.2 for compressor speeds of 700 rpm and 2000 rpm, respectively. This figure shows that the simulated results by the mathematical model of accumulator agree well with the experimental data. The deviations between the experimental and simulated pressure drops are less than 13 percent.

Figure 5.2 Simulated and experimental values of pressure drop through the accumulator with $v = 700$ rpm and $v = 2000$ rpm

Figure 5.3 Predicted effects of diameter of oil bleed hole on pressure drop with

$\dot{m}_v = 120 \text{ kg/h}$, $P_{\text{acc-in}} = 260 \text{ kPa}$, $c = 4.5\%$, $D_J = 10\text{mm}$, and $H_{\text{liq}} = 100 \text{ mm}$

5.1.2 Parametric studies

5.1.2.1 Variable oil bleed hole

Figure 5.3 shows the effect of the diameter of the oil bleed hole on pressure drop, and Figure 5.4 shows the effect of the diameter of oil bleed hole on vapour quality. During this simulation, the vapour mass flow rate, $\dot{m}_v$, is 120 kg/h, the inlet pressure, $P_{\text{acc-in}}$, is 260 kPa, the oil concentration, c , is 4.5 percent, the diameter of J-tube, D_J , is 10 mm and the liquid height, H_{liq} , is 100 mm. The pressure drop is the difference between the accumulator inlet pressure and outlet pressure. The vapour quality is at the

accumulator outlet as well as at the compressor inlet, neglecting heat transfer between the pipe and the surroundings. Furthermore, the vapour quality is also the same at the accumulator inlet, based on the assumption of no heat transfer between accumulator and environment. Pressure drop increases, and the vapour quality decreases with increased diameter of oil bleed hole. Since more liquid flows into J-tube through oil bleed hole, vapour quality decreases and pressure drop increases due to an increase in flow speed.

Figure 5.4 Predicted effects of diameter of oil bleed hole on vapour quality with

$$\dot{m}_v = 120 \text{ kg/h}, P_{\text{acc-in}} = 260 \text{ kPa}, c = 4.5\%, D_j = 10\text{mm}, \text{ and } H_{\text{liq}} = 100 \text{ mm}$$

Figure 5.5 Predicted effects of diameter of J-tube on pressure drop with

$\dot{m}_v = 120 \text{ kg/h}$, $P_{\text{acc-in}} = 260 \text{ kPa}$, $c = 4.5\%$, $D_o = 1.5 \text{ mm}$, and $H_{\text{liq}} = 100 \text{ mm}$

Figure 5.6 Predicted effects of diameter of J-tube on vapour quality with

$\dot{m}_v = 120 \text{ kg/h}$, $P_{\text{acc-in}} = 260 \text{ kPa}$, $c = 4.5\%$, $D_o = 1.5 \text{ mm}$, and $H_{\text{liq}} = 100 \text{ mm}$

5.1.2.2 Variable J-tube diameter

Figure 5.5 shows the effect of internal diameter of J-tube on the pressure drop, and Figure 5.6 shows the effect of internal diameter of the J-tube on the vapour quality. During this simulation, the vapour mass flow rate, $\dot{m}_v$, is 120 kg/h, the inlet pressure, $P_{\text{acc-in}}$, is 260 kPa, the oil concentration, c , is 4.5 percent, the diameter of oil bleed hole, D_o , is 1.5 mm and the liquid height, H_{liq} , is 100 mm. The pressure drop decreases with increasing diameter of J-tube, because increasing its diameter causes a decrease in flow velocity. The vapour quality increases with increasing diameter of J-tube, because the pressure drop due to sudden contraction and expansion through oil bleed hole decreases, as does the mass flow rate of liquid refrigerant/oil mixture.

Figure 5.7 Predicted effects of liquid height on pressure drop with $\dot{m}_v = 120$ kg/h,

$$P_{\text{acc-in}} = 260 \text{ kPa}, c = 4.5\%, D_o = 1.5 \text{ mm}, \text{ and } D_J = 10 \text{ mm}$$

Figure 5.8 Predicted effects of liquid height on vapour quality with

$\dot{m}_v = 120 \text{ kg/h}$, $P_{\text{acc-in}} = 260 \text{ kPa}$, $c = 4.5\%$, $D_o = 1.5 \text{ mm}$, and $D_J = 10 \text{ mm}$

5.1.2.3 Variable liquid height

Figure 5.7 shows the effect of the liquid height on the pressure drop, and Figure 5.8 shows the effect of the liquid height on the vapour quality. During this simulation, the vapour mass flow rate, $\dot{m}_v$, is 120 kg/h, the inlet pressure, $P_{\text{acc-in}}$, is 260 kPa, the oil concentration, c , is 4.5 percent, the diameter of oil bleed hole, D_o , is 1.5 mm and the diameter of the J-tube, D_J , is 10 mm. When the liquid height changes from 20 mm to 220 mm, the vapour quality decreases from 82.12 percent to 80.49 percent. The technical reason for this behaviour is that a higher liquid height causes a larger liquid mass flow rate through oil bleed hole into J-tube, which means the vapour quality becomes lower. The outlet pressure changes from 251.46 kPa to 251.70 kPa, so the pressure drop remains unchanged with liquid height for practical purposes. The technical reason for this

behaviour is that ΔP_{3-1} increases because of higher liquid height, and ΔP_{3-2} increases because of larger liquid mass flow rate, but their difference, pressure drop ΔP_3 , remains nearly constant. The pressure drops through other parts of accumulator are due to the vapour mass flow rate, which is almost constant, so the total pressure drop through accumulator can be regarded as not changing.

5.1.2.4 Variable oil concentration

Figure 5.9 shows the effect of oil concentration on pressure drop, and Figure 5.10 shows the effect of oil concentration on vapour quality. During this simulation, the vapour mass flow rate, $\dot{m}_v$, is 120 kg/h, the inlet pressure, $P_{\text{acc-in}}$, is 260 kPa, the diameter of J-tube, D_J , is 10 mm, the diameter of oil bleed hole, D_o , is 1.5 mm and the liquid height, H_{liq} , is 100 mm. When the oil concentration changes from 1.5 percent to 7.5 percent, the outlet pressure changes from 251.36 kPa to 251.24 kPa, while the vapour quality changes from 78.79 percent to 79.53 percent. For practical purposes, both pressure drop and vapour quality remain unchanged with oil concentration. The technical reason for this is that the oil mass flow rate is relatively small compared with that of the liquid refrigerant, and the change of oil concentration nearly does not affect density of oil-refrigerant mixture, as well as the pressure drop and vapour quality.

Figure 5.9 Predicted effects of oil concentration on pressure drop with

$\dot{m}_v = 120 \text{ kg/h}$, $P_{\text{acc-in}} = 260 \text{ kPa}$, $D_o = 1.5 \text{ mm}$, $D_j = 10 \text{ mm}$, and $H_{\text{liq}} = 100 \text{ mm}$

Figure 5.10 Predicted effects of oil concentration on vapour quality with

$\dot{m}_v = 120 \text{ kg/h}$, $P_{\text{acc-in}} = 260 \text{ kPa}$, $D_o = 1.5 \text{ mm}$, $D_j = 10 \text{ mm}$, and $H_{\text{liq}} = 100 \text{ mm}$

**Figure 5.11 Effects of vapour quality on volumetric efficiency with $r = 2.89$,
and $v = 700$ rpm**

5.2 Compressor performance

This experimental analysis of the performance of a reciprocating compressor for automotive A/C systems takes into account two-phase flow and the oil in circulation. Some results obtained with two-phase flow measurements reveal some unknown aspects of the compression process of automotive A/C systems. The vapour quality does not affect the volumetric efficiency, but affects the isentropic efficiency of compressor. Isentropic efficiency decreases with increasing vapour quality in the range of investigation (80 ~ 90 percent). During the compression process, only a portion of the liquid refrigerant evaporates. Therefore, at the outlet of the compressor, the refrigerant/oil

mixture never reaches an equilibrium state. The evaporation ratio decreases with increasing compressor speed, and increasing vapour quality as well.

**Figure 5.12 Effects of vapour quality on volumetric efficiency with $r = 4.83$,
and $v = 1500$ rpm**

5.2.1 Effects of vapour quality on volumetric efficiency

Figures 5.11, 5.12 and 5.13 show the effects of vapour quality on volumetric efficiency at 3 different compressor speeds, which represent idle, city, and highway driving conditions, respectively. Figure 5.11 shows the experiment, in which the compression ratio is 2.89, the compressor speed is 700 rpm, and the vapour quality ranges from 86.7 to 89.2 percent. The trend line of the volumetric efficiency is parallel to the vapour quality coordinate, and the average of efficiencies is 0.69. Figure 5.12 shows

the experiment, in which the compression ratio is 4.83, and compressor speed is 1500 rpm. The vapour quality ranges from 80.1 to 83.3 percent. The trend line of the volumetric efficiency is also parallel to the vapour quality coordinate, and the average of efficiencies is 0.55. Figure 5.13 shows the experiment, in which the compression ratio is 5.38, and the compressor speed is 2000 rpm. The vapour quality ranges from 80.2 to 86.0 percent. The trend of the volumetric efficiency is level, as in the previous two cases, and the average of efficiencies is 0.50. These experimental results show that the vapour quality does not influence the volumetric efficiency, which is determined by compression ratio and compressor speed.

**Figure 5.13 Effects of vapour quality on volumetric efficiency with $r = 5.38$,
and $v = 2000$ rpm**

**Figure 5.14 Effects of vapour quality on isentropic efficiency with $r = 2.88$,
and $v = 700$ rpm**

**Figure 5.15 Effects of vapour quality on isentropic efficiency with $r = 4.86$,
and $v = 1500$ rpm**

**Figure 5.16 Effects of vapour quality on isentropic efficiency with $r = 5.32$,
and $v = 2000$ rpm**

5.2.2 Effects of vapour quality on isentropic efficiency

Figures 5.14, 5.15 and 5.16 demonstrate the effects of vapour quality on isentropic efficiency. Figure 6 shows the experiment, in which the compression ratio is 2.88, the compressor speed is 700 rpm, and the vapour quality ranges from 87.6 to 89.2 percent. The trend line of the volumetric efficiency in Figure 5.11 is nearly parallel to the vapour quality coordinate, but the value in Figure 5.14 shows that the isentropic efficiency decreases slightly with increasing vapour quality. Figure 5.15 shows the experiment, in which the compression ratio is 4.86, and the compressor speed is 1500 rpm. The vapour quality ranges from 81.5 to 83.3 percent. The isentropic efficiency decreases with increasing vapour quality. Figure 5.16 shows the experiment, in which the compression

ratio is 5.32, and the compressor speed is 2000 rpm. The vapour quality ranges from 80.2 to 86.0 percent. The isentropic efficiency decreases sharply with the increasing vapour quality. The above experimental results show that isentropic efficiency decreases with increasing vapour quality from 80.2 to 89.2 percent, which is within the range of the vapour quality during automotive A/C systems' operation. The magnitude of this decrease is proportional to the compressor speed. The reason is that liquid evaporates in cylinder during compression stroke to reduce the temperature of the whole compressor, as well as the discharge temperature, and the decrease in temperature improves isentropic efficiency. With increasing compressor speed, the discharge temperature increases, thus the effects of liquid evaporation to isentropic efficiency show more prominently.

5.2.3 Evaporation ratio in compression process

At inlet of the compressor, the relationship of pressure, P_{in} , liquid refrigerant fraction, w_{in} , and the saturation temperature of mixture, T_{in} , are given by Equations (3.14) and (3.15), which are the equilibrium equations of the two-phase two-component mixture. During compression process, the pressure and temperature increase, and some of liquid refrigerant evaporates. If the compression process is slow enough, the final state could reach equilibrium, for which the relationships of pressure, P_{out} , liquid refrigerant fraction, w_{out} , and the saturation temperature of mixture, T_{out} , are given by Equations (3.14) and (3.15); w_{out} is usually smaller than w_{in} due to the evaporation of the liquid refrigerant. However, at normal speed of compressor, there is not enough time for liquid refrigerant to evaporate. The compression process cannot reach an equilibrium state, and the actual

liquid refrigerant fraction, w_{act} , is between w_{in} and w_{out} . In order to describe the evaporation of liquid refrigerant, the evaporation ratio is defined by the following equation:

$$R = \frac{w_{in} - w_{act}}{w_{in} - w_{out}} \quad (5.1)$$

where, w_{in} is calculated from measured vapour quality, x_{in} , and oil concentration, c_{in} , at inlet of compressor, and w_{out} is calculated from measured pressure, P_{out} , temperature, T_{out} , and oil concentration, c_{out} , (equals c_{in}) by using Equations (3.14) and (3.15).

The actual liquid refrigerant fraction, w_{act} , is obtained by solving the following equation:

$$\dot{m}_t(h_{mix,out} - h_{mix,in}) = W_c \quad (5.2)$$

where, $\dot{m}_t$ is the measured total mass flow rate, h is calculated using Equation (3.85), and W_c is from experimental data.

Figure 5.17 shows the evaporation ratio at compressor speeds of 700, 1500, and 2000 rpm. At these three speeds, the vapour qualities are all 86.0 percent. The trend of the evaporation ratio shows a clear decrease with an increase in compressor speed. The reason is that there is less retention time for the liquid to evaporate at higher compressor speed.

Figures 5.18, 5.19 and 5.20 show the effects of vapour quality on evaporation ratio at compressor speeds of 700, 1500, and 2000 rpm, respectively. In all three figures, the evaporation ratio decreases with increasing vapour quality. The reason could be that the liquid refrigerant fraction decreases with increasing vapour quality, which makes it difficult for refrigerant molecules to escape from of the liquid refrigerant/oil mixture, which has a low refrigerant concentration.

Figure 5.17 Effects of compressor speed on evaporation ratio with $x = 86\%$

Figure 5.18 Effects of vapour quality on evaporation ratio with $v = 700$ rpm

Figure 5.19 Effects of vapour quality on evaporation ratio with $v = 1500$ rpm

Figure 5.20 Effects of vapour quality on evaporation ratio with $v = 2000$ rpm

5.3 A new automotive A/C system

In this section, an accumulator-capillary tube-suction line heat exchanger was developed and investigated experimentally with comparison to conventional systems with and without a liquid line-suction line heat exchanger. The experiments were carried out on AACTS, at three compressor speeds of 700, 1500, and 2000 rpm. The working fluid used in the experiment was R-134a, and lubricating oil was Polyalkylene Glycol (PAG) oil. Variables investigated include mass flow rate, vapour quality, discharge pressure, compression ratio, cooling capacity, power input, coefficient of performance, discharge temperature, and isentropic and volumetric efficiencies of compressor. The results

showed that both kinds of internal heat exchanger have positive influences on the performance of automotive A/C systems.

5.3.1 Accumulator-internal heat exchanger

In a normal automotive CCOT system with no IHX (NIHX), the oil concentration is about 5 percent, and the vapour quality of the refrigerant at inlet of compressor is about 80–90 percent, determined by the accumulator, according to Wang et al. (2006). The effects of vapour quality on performance of automotive A/C systems have been investigated and results reported by Wang et al. (2005).

Figure 5.21 Automotive air-conditioning systems with (a) NIHX, (b) ACTSLHX and (c) LLSLHX

Figure 5.22 Internal structure of ACTSLHX

According to the requirements of automotive A/C systems in size and weight, a CTS�HX for automotive CCOT systems was developed. This CTS�HX, named accumulator-capillary tube-suction line heat exchanger (ACTSLHX), was installed inside the accumulator. Figure 5.21 shows schematics of three kinds of A/C systems studied: (a) the conventional A/C system with no internal heat exchanger (NIHX); (b) the A/C system with ACTSLHX; (c) the A/C system with LLSLHX. Figure 5.22 shows the internal structure of the ACTSLHX. A capillary tube is inside a J-shaped tube or J-tube as an expansion device and heat exchanger. From an energy utilization point of view, the

capillary tube is in the two-phase flow part of the accumulator, between the bleeding hole and the outlet of J-tube. Refrigerant inside the capillary tube transfers heat to refrigerant inside the J-tube.

5.3.2 Comparison investigation of A/C systems

5.3.2.1 Experimental conditions

Figure 5.23 shows the schematic illustration of the experimental setup of the NIHX system used in this experimental investigation.

Figure 5.23 Experimental setup for NIHX system

Figure 5.24 Experimental setup for ACTSTHX system

Figure 5.25 Experimental setup for LLSLHX system

Table 5.1 Experimental conditions

Systems	NIHX			ACTSLHX			LLSLHX		
Compressor speed (rpm)	700	1500	2000	700	1500	2000	700	1500	2000
Evaporation temperature (°C)	1.62	-3.42	-3.94	0.83	-4.69	-5.67	2.51	-3.42	-4.69
Condensation temperature (°C)	51.24	61.93	66.10	48.67	57.82	60.52	43.47	51.25	56.93
Sub-cooling (°C)	12.29	22.83	27.00	9.47	18.87	21.32	5.27	12.79	18.43
Superheat (°C)	2.56	9.44	12.95	2.87	12.03	18.41	5.25	18.73	25.11
Discharge temperature (°C)	56.41	74.12	81.85	54.27	72.91	82.04	52.21	73.77	82.83

Figure 5.24 shows the schematic illustration of the experimental setup of the ACTSLHX system. The capillary tubes used in the NIHX system and the ACTSTHX system are of the same dimensions, which are 3.18 mm outer diameter, 1.65 mm inner diameter and 500 mm length. In the ACTSTHX system, the effective length of the capillary tube for heat exchange is 400 mm.

Figure 5.25 shows the schematic illustration of the experimental setup of the LLSLHX system. The dimensions of liquid line tube are 12.70 mm outer diameter and 9.53 mm inner diameter. The dimensions of the suction line tube are 25.4 mm outer diameter and 22.23 mm inner diameter. The effective length for heat exchange is 600 mm.

Figure 5.26 Total mass flow rate at compressor inlet for variable compressor speed

The experiments of these three systems were carried out at identical operating conditions: the refrigerant charge is 1.8 kg, the evaporator air inlet temperatures is 25°C, the condenser water temperature is 40°C, and other experimental conditions are shown in Table 5.1. The compressor speeds are 700, 1500 and 2000 rpm. Several parameters were measured, charted, and discussed in the following paragraphs, such as the total mass flow rate, gas flow rate, vapour quality (R-134a and PAG) at the inlet of compressor, the discharge pressure, compression ratio, the cooling capacity, power input, and COP.

5.3.2.2 Mass flow rate and vapour quality at inlet of compressor

Figure 5.26 and 5.27 show the total mass flow rate and vapour mass flow rate at inlet of compressor at three compressor speeds. For three systems, both the total mass

flow rate and the vapour mass flow rate increase with the increase of compressor speed. The LLSLHX system has the lowest total mass flow rate, and highest vapour mass flow rate. The NIHX system has the highest total mass flow rate, and lowest vapour mass flow rate. The ACTSLHX system has total mass flow rate and vapour mass flow rate between that of the LLSLHX and NIHX systems.

Figure 5.27 Vapour mass flow rate at compressor inlet for variable compressor speed

Figure 5.28 shows the vapour quality, at the inlet of the compressor. The LLSLHX system has the highest vapour quality, while the NIHX has the lowest. The difference between the highest and the lowest increases with an increase of compressor speed. The main reason for the above phenomena is that part of the liquid refrigerant in the refrigerant/oil two-phase flow from the accumulator absorbs heat from the liquid line or

capillary tube and evaporates, which increases the vapour mass flow rate and vapour quality at the inlet of the compressor. The vapour quality of the LLSLHX is higher than that of the ACTLLHX, because the LLSLHX has a larger heat transfer area. In the compression process, refrigerant/oil mixture with a higher vapour quality is of lower density, which causes a decrease in the liquid and total mass flow rate, thus protects the compressor from having too much liquid during the compression process.

Figure 5.28 Vapour qualities at compressor inlet for variable compressor speed

5.3.2.3 Discharge pressure and compression ratio in compression process

Figure 5.29 shows the discharge pressure of the three systems at three compressor speeds. The LLSLHX system has the lowest discharge pressure, while the NIHX has the highest. The discharge pressure of the LLSLHX system is lower than that of the NIHX

system by 16 percent, 20 percent, and 23 percent at compressor speeds of 700 rpm, 1500 rpm and 2000 rpm, respectively. A lower discharge pressure improves the working conditions of the compressor.

Figure 5.30 shows the compression ratio of three systems at three compressor speeds. The NIHX system has the highest compression ratio, while the LLSLHX system has the lowest. The reason could be that the NIHX system has the lowest vapour quality i.e. highest liquid refrigerant fraction, which evaporates in compression process, thereby increases the discharge pressure. For the three systems, the suction pressures are nearly the same, so the LLSLHX with the lowest discharge pressure has the lowest compression ratio. Normally, compressor efficiencies increase with a decrease in compression ratio; therefore, IHXs have a positive effect on compressor performance.

Figure 5.29 Discharge pressure at compressor inlet for variable compressor speed

Figure 5.30 Compression ratio at compressor inlet for variable compressor speed

Figure 5.31 Cooling load at compressor inlet for variable compressor speed

5.3.2.4 Coefficient of performance

Figures 5.31, 5.32 and 5.33 show the cooling load, power input and COP of the three systems at three compressor speeds, respectively. The NIHX system has the lowest cooling load and highest power input, and thereby has the lowest COP. The LLSLHX system has the lowest power input and nearly the same cooling load as that of the ACTSLHX system, thus it has the highest COP. The ACTSLHX system has COP between that of the NIHX and the LLSLHX systems. The COP of the LLSLHX is higher than that of the NIHX by 13 percent and 21 percent at compressor speeds of 700 rpm and 2000 rpm, respectively, which shows that IHXs have a improvement on COP at higher compressor speed.

Figure 5.32 Power input at compressor inlet for variable compressor speed

Figure 5.33 COP at compressor inlet for variable compressor speed

Table 5.2 Compressor isentropic and volumetric efficiencies

Systems	NIHX			ACTSLHX			LLSLHX		
Compressor speed (rpm)	700	1500	2000	700	1500	2000	700	1500	2000
Isentropic efficiency	0.598	0.426	0.343	0.613	0.456	0.383	0.605	0.457	0.398
Volumetric efficiency	0.615	0.460	0.378	0.631	0.475	0.398	0.608	0.457	0.395

5.3.2.5 Compressor efficiencies

Table 5.2 shows the isentropic and volumetric efficiencies of the three systems at three compressor speeds. At the same speed, the ACTSLHX and the LLSLHX systems have higher values than the NIHX system in isentropic efficiencies, by 7 percent in

average, but the three systems have nearly equal volumetric efficiencies. The reason for the increase of isentropic efficiency is that the ACTSLHX and the LLSLHX systems have lower compression ratios, which have a positive effect on compressor performance.

5.3.2.6 P-h diagram of vapour compression refrigeration cycles

Following the experimental results and discussion presented, a P-h diagram of vapour compression refrigeration cycles for the three A/C systems was drawn as Figure 5.34. Here the pressure through pipes was ignored, while the effects of oil on refrigerant properties were considered. The dashed curve is for pure R-134a, and the solid curve illustrates the saturation curve for R-134a/PAG mixture at a certain liquid refrigerant fraction, and moves according to the change of liquid refrigerant fraction. The solid curve will move towards the dashed curve with the increase of liquid refrigerant fraction, and it reaches the dashed curve when liquid refrigerant fraction equals 1, which means no existence of oil. Figure 5.34 shows that the refrigerant is two-phase at both inlet and outlet of compressor because of the existence of oil, even though the temperature at outlet is higher than saturated temperature of pure R-134a.

In Figure 5.34, processes 1-2-3-4-1, 1'-2'-3'-4'-1', and 1''-2''-3''-4''-1'' indicate the cycles of NIHX, ACTSLHX, and LLSLHX systems, respectively. $(h_{4'} - h_{3'})$ and $(h_{1'} - h_1)$ are the heat removed from the capillary tube to the suction line. $(h_{4''} - h_3)$ and $(h_{1''} - h_1)$ are the heat removed from the liquid line to the suction line. The heat transfer causes an increase in specific cooling capacities that are $(h_{1'} - h_{4'})$ for the ACTSLHX system, and

$(h_{1''} - h_{4''})$ for the LLSLHX system, compared with the specific cooling capacity of the NIHX system, $(h_1 - h_4)$. The COPs can be expressed as the following:

$$\text{COP} = \frac{h_1 - h_4}{h_2 - h_1} \quad (5.3)$$

$$\text{COP}' = \frac{h_{1'} - h_{4'}}{h_{2'} - h_{1'}} \quad (5.4)$$

$$\text{COP}'' = \frac{h_{1''} - h_{4''}}{h_{2''} - h_{1''}} \quad (5.5)$$

From Figure 5.34, the following can be obtained:

$$\text{COP} < \text{COP}' < \text{COP}'' \quad (5.6)$$

Figure 5.34 P-h diagram of vapour compression refrigeration cycles

5.3.3 Parametric studies of new system

5.3.3.1 Experimental conditions for ACTSLHX systems

The experiments of three ACTSLHX systems were carried out at the same operating conditions: the refrigerant charges is 1.8 kg, the evaporator air inlet temperatures is 25°C, the condenser water temperatures 40°C, and the compressor speeds are 700, 1500 and 2000 rpm, respectively. Capillary tubes with diameters of 0.07, 0.08 and, 0.09 inch (1.778, 2.032 and 2.286 mm), and lengths of 750, 1000, and 1250 mm, were investigated. Several parameters were measured, charted and discussed in the following paragraphs, such as the total mass flow rate, the gas flow rate, the vapour quality at the inlet of the compressor, the discharge pressure, the compression ratio, and the discharge temperature.

5.3.3.2 Effect of capillary tube length

The experiment was done with three capillary tubes that have diameter of 1.778 mm and lengths of 750, 1000, and 1250 mm, respectively. Figure 5.35 shows effect of capillary tube length on discharge temperature in terms of compressor speed. Figure 5.36 shows effect of the capillary tube length on discharge pressure in terms of compressor speed. Both discharge temperature and pressure increase with the increase of compressor speed and capillary tube length. The reason is that the increase in compressor speed and capillary tube length cause an increase in pressure drop when refrigerant flows through capillary, which increases discharge pressure and temperature, and decreases temperature

and pressure at the inlet of the compressor. Figure 5.37 shows the effect of capillary tube length on the compression ratio in terms of compressor speed. The compression ratio increases with the increase of the compressor speed and the capillary tube length. The reason is the same as that phenomenon described by Figures 5.35 and 5.36.

Figure 5.35 Effect of capillary length on discharge temperature in terms of compressor speed ($d = 1.778$ mm)

Figures 5.38, 5.39 and 5.40 show the effect of capillary tube length on the total mass flow rate, the vapour mass flow rate and the vapour quality, respectively, in terms of the compressor speed. Both the total and vapour mass flow rate decrease with an increase of capillary tube length. The reason is that the increase of capillary tube length causes an increase in flow resistance. Vapour quality increases with an increase of capillary tube length.

Figure 5.36 Effect of capillary length on discharge pressure in terms of compressor speed ($d = 1.778$ mm)

Figure 5.37 Effect of capillary length on compression ratio in terms of compressor speed ($d = 1.778$ mm)

Figure 5.38 Effect of capillary length on total flow rate in terms of compressor speed ($d = 1.778$ mm)

Figure 5.39 Effect of capillary length on vapour flow rate in terms of compressor speed ($d = 1.778$ mm)

Figure 5.40 Effect of capillary length on vapour quality in terms of compressor speed ($d = 1.778$ mm)

Figure 5.41 Effect of capillary diameter on discharge temperature in terms of compressor speed ($l = 1000$ mm)

Figure 5.42 Effect of capillary diameter on discharge pressure in terms of compressor speed ($l = 1000$ mm)

Figure 5.43 Effect of capillary diameter on compression ratio in terms of compressor speed ($l = 1000$ mm)

5.3.3.3 Effect of capillary tube diameter

An experiment was done with three capillary tubes that have length of 1000 mm and diameters of 0.07, 0.08, and 0.09 inch (1.778, 2.032 and 2.286 mm), respectively. Figure 5.41 shows the effect of capillary tube diameter on discharge temperature in terms of compressor speed. Figure 5.42 shows the effect of capillary tube diameter on discharge pressure in terms of compressor speed. Both the discharge temperature and pressure increase with the increase of compressor speed, and the decrease of capillary tube diameter. The reason is that the increase in compressor speed and decrease in capillary tube diameter cause an increase in pressure drop when the refrigerant flows through capillary, which increases the discharge pressure and temperature, and decreases the temperature and pressure at the inlet of the compressor. Figure 5.43 shows the effect of the capillary tube diameter on compression ratio in terms of compressor speed. The compression ratio increases with the increase of compressor speed and the decrease of capillary tube diameter. The reason is the same as that phenomenon described by Figures 5.21 and 5.22.

Figures 5.44, 5.45, and 5.46 show the effect of the capillary tube diameter on the total mass flow rate, vapour mass flow rate, and vapour quality, respectively, in terms of compressor speed. The total mass flow rate increases with the increase of capillary tube diameter. The vapour mass flow rate increases with an increase of capillary tube diameter under a compressor speed 700 rpm, but vapour mass flow rate first increases, then decreases with the increase of capillary tube diameter. The vapour quality decreases with the increase of capillary tube diameter.

Figure 5.44 Effect of capillary diameter on total flow rate in terms of compressor speed ($l = 1000$ mm)

5.4 System analysis

The experiments were carried out at different refrigerant charges, evaporator air inlet temperatures, condenser water temperatures, and compressor speeds. Several parameters were measured, charted and discussed in following paragraphs, such as the cooling capacity, the compressor power consumption, total flow rate, gas flow rate, the vapour quality (R-134a and PAG) at the inlet of the compressor, the pressure at outlet of compressor, and the pressure at the outlet of the evaporator.

Figure 5.45 Effect of capillary diameter on vapour flow rate in terms of compressor speed ($l = 1000$ mm)

Figure 5.46 Effect of capillary diameter on vapour quality in terms of compressor speed ($l = 1000$ mm)

Figure 5.47 The effects of refrigerant charge on total mass flow rate and gas flow rate with $T_{\text{evap-air}} = 30^{\circ}\text{C}$, $T_{\text{cond-water}} = 35^{\circ}\text{C}$, and $v = 2000$ rpm.

5.4.1 Effects of refrigerant charge

In an automotive A/C system, permeation of refrigerant through elastomeric hoses and fittings causes a charge loss over time. The practical solution is to overcharge the systems, and let the accumulators contain the spare refrigerant. It is time to recharge the system when the spare refrigerant permeates completely. The effects of refrigerant charge on system operation are shown in Figures 5.47-5.52. During this experiment, the evaporator air inlet temperature, $T_{\text{evap-air}}$, is at 30°C , condenser water temperature, $T_{\text{cond-water}}$, is 35°C and the compressor speed, v , is 2000 rpm.

Figure 5.48 The effects of refrigerant charge on vapour quality with

$T_{\text{evap-air}} = 30^{\circ}\text{C}$, $T_{\text{cond-water}} = 35^{\circ}\text{C}$, and $v = 2000 \text{ rpm}$.

Figure 5.47 shows that as the total mass flow rate (R-134a and PAG oil) increases slightly, gas flow rate stays at a constant value or decreases slightly with the increase of refrigerant charge. Figure 5.48 shows the vapour quality at accumulator outlet decreases slightly with the increase of refrigerant charge. The quality is believed to have a major influence on the lubrication of compressor. The quality is controlled by the accumulator, in which the vapour R-134a, liquid R-134a and PAG oil separate, and the PAG oil in R-134a is metered back to the compressor. The high speed vapour R-134a flow causes liquid (R-134a and PAG oil) carryover. When the refrigerant charge increases, more refrigerant is stored in the accumulator, and more liquid refrigerant and PAG oil can be carried. The mechanism of oil return will be discussed in a separate work.

Figure 5.49 The effects of refrigerant charge on cooling load and compressor power input with $T_{\text{evap-air}} = 30^{\circ}\text{C}$, $T_{\text{cond-water}} = 35^{\circ}\text{C}$, and $v = 2000$ rpm.

Figure 5.50 The effects of refrigerant charge on COP with $T_{\text{evap-air}} = 30^{\circ}\text{C}$, $T_{\text{cond-water}} = 35^{\circ}\text{C}$, and $v = 2000$ rpm.

Figure 5.49 shows that cooling load and compressor power input increase slightly with the increase of refrigerant charge. Figure 5.50 shows COP decreases slightly with the increase of refrigerant charge. The decrease of quality, as shown in Figure 5.48, means relatively less gas is compressed in the compressor, which causes the decrease of COP.

Figure 5.51 shows the compressor outlet pressure and inlet pressure increase slightly with the increase of refrigerant charge, but Figure 5.52 shows the compression ratio decreases with the increase of refrigerant charge.

Figure 5.51 The effects of refrigerant charge on compressor outlet and inlet pressure with $T_{\text{evap-air}} = 30^{\circ}\text{C}$, $T_{\text{cond-water}} = 35^{\circ}\text{C}$, and $v = 2000$ rpm.

The compression ratio directly affects the volumetric efficiency of the compressor. The volumetric efficiency of the compressor is the ratio of the actual delivered gas volume to the swept volume of the cylinder. When the compression ratio increases, the actual delivered gas volume decreases. Therefore, the volumetric efficiency decreases with the increase of the compression ratio, because of a high fraction residual of gas.

In general, all the parameters, such as quality, cooling load, and COP, change slightly or keep constant when the refrigerant charge is between 1.2 kg and 1.8 kg for this specific system. When refrigerant charge is less than 1.2 kg or more than 1.8 kg, COP decreases apparently. Jabardo et al. (2002) obtained similar results in their experiments. These results showed one of the important functions of the accumulator, which is to store excess refrigerant, by Lee, Kim and Huh (2003). When the refrigerant charge varies within 1.2 kg to 1.8 kg, the accumulator performs well to hold excess refrigerant with no or slight effect on the system. A refrigerant charge lower than 1.2 kg cannot meet the minimum demand of the system; the condenser could not get enough refrigerant vapour, and therefore, the system runs at a reduced capacity. A refrigerant charge more than 1.8 kg causes excess refrigerant more than accumulator can hold; the large amount of liquid returning back to the compressor will negatively affect the vapour compression, so the COP decreases.

Figure 5.52 The effects of refrigerant charge on compression ratio with

$T_{\text{evap-air}} = 30^{\circ}\text{C}$, $T_{\text{cond-water}} = 35^{\circ}\text{C}$, and $v = 2000$ rpm.

5.4.2 Effect of inlet air temperature to evaporator

Figures 5.53-5.64 show the effects of the evaporator inlet air temperature. During these experiments, the charged R-134a, m_{char} , is 1.8 kg, condenser water temperature, $T_{\text{cond-water}}$, is 43°C and the compressor speed, v , is 1500 rpm. When the evaporator inlet air temperature increases, the temperature difference between air and refrigerant increases, which improves the heat exchange of the evaporator.

Figure 5.53 The effects of evaporator air inlet temperature on total mass flow rate and gas flow rate with $m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{cond-water}} = 43^\circ\text{C}$, and $v = 1500 \text{ rpm}$

Figure 5.54 The effects of evaporator air inlet temperature on vapour quality with $m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{cond-water}} = 43^\circ\text{C}$, and $v = 1500 \text{ rpm}$

Figure 5.53 shows that both the total mass flow rate (R-134a and PAG oil) and gas flow rate at the compressor inlet increase with an increase of evaporator air side temperature. Figure 5.54 shows the quality decreases sharply to a point, then increases slightly with an increase of evaporator air side temperature. When the velocity of the vapour is very low, it can only carry a limited amount of liquid R-134a and PAG oil, but when the velocity approaches a certain value, liquid R-134a and PAG oil flows at high velocity because of the interfacial shear force, which dominates the liquid carryover and the total quality, according to the analysis by Huang and Gu (1989).

Figure 5.55 The effects of evaporator air inlet temperature on cooling load and compressor power input with $m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{cond-water}} = 43^\circ\text{C}$, and $v = 1500 \text{ rpm}$

Figure 5.56 The effects of evaporator air inlet temperature on COP with

$m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{cond-water}} = 43^\circ\text{C}$, and $v = 1500 \text{ rpm}$

Figure 5.57 The effects of evaporator air inlet temperature on compressor outlet

and inlet pressure with $m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{cond-water}} = 43^\circ\text{C}$, and $v = 1500 \text{ rpm}$

Figure 5.55 shows that the cooling load and compressor power input increase with the increase of evaporator air inlet temperature. Figure 5.56 shows that the COP increases with an increase of evaporator air inlet temperature. For an ideal vapour-compression refrigeration cycle, COP can also be expressed as

$$COP = \frac{T_c}{T_h - T_c} \quad (5.7)$$

where, T_c is the temperature of the refrigerant in the evaporator and T_h is the temperature of the refrigerant in the condenser. Although Equation (5.7) can not be used to depict the actual cycle quantitatively, it can be used to express the trend of COP with the change of T_c and T_h qualitatively. T_c and T_h increase with the increase of surrounding temperature of the evaporator and condenser, respectively. In this experiment, the surrounding temperature of the evaporator increases and that of the condenser keeps constant, which causes the COP to increase.

Figure 5.57 depicts the compressor outlet pressure and inlet pressure increase with the increase of evaporator air inlet temperature. Figure 5.58 shows that the compression ratio decreases with the increase of the evaporator air inlet temperature. The compressor outlet pressure is mainly determined by the surrounding temperature of condenser, increases with the increase of the surrounding temperature of condenser. The compressor inlet pressure is mainly determined by the surrounding temperature of the evaporator and increases with the increase of the surrounding temperature of the evaporator. In this case, the condenser water temperature is constantly at 43°C, and the evaporator air inlet temperature increases, so the compression ratio decreases.

Figure 5.58 The effects of evaporator air inlet temperature on compression ratio

with $m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{cond-water}} = 43^\circ\text{C}$, and $v = 1500 \text{ rpm}$

Figure 5.59 The effects of condenser temperature on total mass flow rate and gas

flow rate with $m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{evap-air}} = 30^\circ\text{C}$, and $v = 1500 \text{ rpm}$

5.4.3 Effect of condenser water temperature

The effects of the condenser water temperature are shown in Figures 5.59-5.64. During this experiment, the charged R-134a mass, m_{char} , is 1.8 kg, the evaporator air inlet temperature, $T_{\text{evap-air}}$, is at 30°C and the compressor speed, v , is 1500 rpm. When the condenser water temperature increases, the temperature difference of the water and refrigerant in the condenser decreases, which reduces the heat exchange in the condenser. Figure 5.59 shows that the total mass flow rate (R-134a and PAG oil) and gas flow rate at the compressor inlet increase with the increase of condenser water temperature. Figure 5.60 shows the quality decreases with the increase of condenser water temperature.

Figure 5.61 shows that the cooling load decreases, but the compressor power input increases with the increase of condenser water temperature. Figure 5.62 shows that the COP decreases with the increase of condenser water temperature. Figure 5.63 shows that the compressor outlet pressure increases sharply and inlet pressure increases slightly with the increase of condenser water temperature. This is because the compressor outlet pressure is mainly determined by the condenser surrounding temperature, which increases from 25°C to 50°C, and the compressor inlet pressure is determined by the evaporator surrounding temperature that is constantly at 30°C in this experiment. The above factors result in an increase of the compression ratio with the increase of the condenser water temperature, as shown in Figure 5.64, which causes a decrease of the volumetric efficiency of the compressor.

Figure 5.60 The effects of condenser temperature on vapour quality with

$m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{evap-air}} = 30^\circ\text{C}$, and $v = 1500 \text{ rpm}$

Figure 5.61 The effects of condenser temperature on cooling load and compressor

power input with $m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{evap-air}} = 30^\circ\text{C}$, and $v = 1500 \text{ rpm}$

Figure 5.62 The effects of condenser temperature on COP with $m_{\text{char}} = 1.8 \text{ kg}$,

$T_{\text{evap-air}} = 30^\circ\text{C}$, and $v = 1500 \text{ rpm}$

Figure 5.63 The effects of condenser temperature on compressor outlet and inlet

pressure with $m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{evap-air}} = 30^\circ\text{C}$, and $v = 1500 \text{ rpm}$

Figure 5.64 The effects of condenser temperature on compression ratio with

$m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{evap-air}} = 30^\circ\text{C}$, and $v = 1500 \text{ rpm}$

Figure 5.65 The effects of compressor speed on total mass flow rate and gas flow

rate with $m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{evap-air}} = 35^\circ\text{C}$, and $T_{\text{cond-water}} = 45^\circ\text{C}$

5.4.4 Effect of compressor speed

The effects of compressor speed are shown in Figures 5.65-5.70. During this experiment, the charged R-134a mass, m_{char} , is 1.8 kg, evaporator air inlet temperature, $T_{\text{evap-air}}$, is 35°C and the condenser water temperature, $T_{\text{cond-water}}$, is 43°C. Figure 5.65 shows that the total mass flow rate (R-134a and PAG oil) and gas flow rate at the compressor inlet increase with the increase of compressor speed. Figure 5.66 shows that the quality decreases with the increase of compressor speed.

Figure 5.66 The effects of compressor speed on vapour quality with $m_{\text{char}} = 1.8$ kg,

$$T_{\text{evap-air}} = 35^{\circ}\text{C}, \text{ and } T_{\text{cond-water}} = 45^{\circ}\text{C}$$

Figure 5.67 The effects of compressor speed on cooling load and compressor power input with $m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{evap-air}} = 35^\circ\text{C}$, and $T_{\text{cond-water}} = 45^\circ\text{C}$

Figure 5.68 The effects of compressor speed on COP with $m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{evap-air}} = 35^\circ\text{C}$, and $T_{\text{cond-water}} = 45^\circ\text{C}$

Figure 5.69 The effects of compressor speed on compressor outlet and inlet pressure with $m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{evap-air}} = 35^\circ\text{C}$, and $T_{\text{cond-water}} = 45^\circ\text{C}$

Figure 5.67 shows that the cooling load and compressor power input increase with the increase of compressor speed. Figure 5.68 shows that the COP decreases with the increase of compressor speed.

Figure 5.69 shows that the compressor outlet pressure increases and inlet pressure decreases slightly with the increase of compressor speed. Figure 5.70 shows that the compression ratio increases with the increase of compressor speed, which results in a decrease of the volumetric efficiency of the compressor.

Figure 5.70 The effects of compressor speed on compression ratio with

$m_{\text{char}} = 1.8 \text{ kg}$, $T_{\text{evap-air}} = 35^{\circ}\text{C}$, and $T_{\text{cond-water}} = 45^{\circ}\text{C}$

Chapter 6

Conclusions and recommendations

6.1 Conclusions

A full size Automotive Air-conditioning Test System with two-phase measurement was set up and used to carry out experiments, and a theoretical investigation was done. The following conclusions were obtained:

- A Stream Analysis Model (SAM) for the accumulator was developed, and the simulated and experimental values of vapour quality at variable diameters of oil bleed hole were compared at the compressor speeds of 700 rpm and 2000 rpm, respectively. It was shown that the simulated results by the SAM for the accumulator agreed well with the experimental data. The deviations between the experimental and simulated vapour quality were less than 5 percent. The simulated and experimental values of pressure drop through the accumulator were compared when the compressor speeds were 700 rpm and 2000 rpm, respectively. It was shown that the simulated results by the mathematical model of accumulator agreed well with the experimental data. The deviations between the experimental and simulated pressure drops were less than 13 percent. The results of the model showed that accumulators determined the vapour quality (two-phase flow) at compressor inlet, and thus affected the behaviour of the

automotive A/C systems. Among the dimensions of the accumulator, the oil bleed hole diameter and diameter of J-tube are the most important parameters that affect the behaviour of accumulators. Pressure drop increases, vapour quality decreases with a increased oil bleed hole diameter at constant vapour mass flow rate. The pressure drop decreased with an increased diameter of the J-tube, and vapour quality increased with an increased diameter of the J-tube at constant vapour mass flow rate. Through changing the diameters of the oil bleed hole and J-tube, the accumulators were optimized to have suitable vapour quality and the lowest pressure drop. The liquid height and oil concentration inside the accumulator had no effect on either the vapour quality or the pressure drop, and this characteristic enabled the accumulator to store excess refrigerant without negative influence on the performance of A/C systems.

- The performance of a reciprocating compressor for automotive A/C systems was experimentally analyzed taking into account of two-phase flow and oil in circulation. Some results were obtained with two-phase flow measurements, which revealed some unknown aspects of the compression process of automotive A/C systems. Due to the oil in circulation, at the inlet and outlet of the compressor, there was always liquid refrigerant in the refrigerant/oil mixture and the equilibrium relationship between the pressure and temperature deviated from that of pure refrigerant. The vapour quality did not affect the volumetric efficiency, but affected the isentropic efficiency of the compressor. Isentropic efficiency decreased with increased vapour quality in the range of this

investigation (80 ~ 90 percent). During the compression process, only a portion of the liquid refrigerant evaporated. Therefore, at the outlet of the compressor, the refrigerant/oil mixture never reached an equilibrium state. The evaporation ratio decreased with increased compressor speed, and increased vapour quality as well.

- An experimental automotive CCOT A/C system where different cycles with and without the adoption of either the LLSLHX or the ACTSLHX was done, and the main conclusions were drawn as follows: First, the ACTSLHX decreased the total and liquid mass flow rate, but increased the vapour mass flow rate, thereby increased the vapour quality by up to 8 percent, which protected the compressor from wet compression. Second, the ACTSLHX decreased the discharge pressure by up to 11 percent, the compression ratio by up to 6 percent, and increased isentropic efficiency by up to 12 percent. Third, the ACTSLHX increased the cooling load by up to 9 percent and the COP by up to 15 percent. Fourth, compared with the LLSLHX, the ACTSLHX had similar cooling loads, but had a lower COP. Fifth, both IHXs did not change discharge temperatures nor volumetric efficiencies. Finally, the ACTSLHX improved performance of A/C systems without increased weight and size of A/C systems, which is very important for automotive A/C systems. The experimental investigation of the ACTSLHX automotive A/C system with different diameters and lengths of capillary tube, some results were obtained. The discharge temperature, discharge pressure, and compression ratio increased with increased capillary

tube length. The total mass flow rate and vapour mass flow rate decreased with increased capillary tube length, but vapour quality increased with increased capillary tube length. The discharge temperature, discharge pressure, and compression ratio decreased with increased capillary tube diameter; The total mass flow rate increased, but vapour quality decreased with the increased capillary tube diameter.

- With the experiment of an A/C system including two-phase flow measurements, some results were obtained that revealed some unknown aspects of the automotive A/C systems. The quality (of refrigerant and PAG oil) at the inlet of compressor was directly related to the COP and cooling load. Two-phase flow measurement was necessary in analyzing automotive A/C precisely. The total mass flow rate increased with the increase of refrigerant charge, evaporator air inlet temperature, condenser water temperature, and compressor speed. The total quality at accumulator outlet increased with the increase of evaporator air inlet temperature, but decreased with the increase of refrigerant charge, condenser water temperature, and compressor speed. The cooling capacity did not change with the variation of refrigerant charge, but increased with increased evaporator air inlet temperature and compressor speed, and decreased with increased condenser water temperature. The Coefficient of performance (COP) of the system decreased with the increase of refrigerant charge, condenser water temperature and compressor speed, and increased with increased evaporator air inlet temperature. The compression ratio decreased with increased refrigerant

charge and evaporator air inlet temperature, and increased with increased condenser water temperature and compressor speed. The change of compressor volumetric efficiency was opposite to that of compression ratio.

6.2 Recommendations

In order to improve the analysis on automotive A/C system proposed in this work, the following future study and action were recommended:

- Improve the Automotive Air Conditioning Test System (AACTS). The existing data acquisition system plays a very important role in the experiments, but there are still some disadvantages; for example, vapour quality (two-phase flow) can only measured at compressor inlet, and oil concentration can only measured at condenser outlet. For further investigation, a recommendation is to make the AACTS capable of measuring vapour quality at the inlet and outlet of the compressor and the inlet of the evaporator, and measuring the oil concentration in the whole system.
- Improve the theoretical analysis model. The existing computational model plays a very important role in the theoretical analysis process, and the performance of accumulator can be described by this model. But it is a steady state model, and still not perfect; several updates should be applied to it in the future. The future research work is to develop a transient model and a system model including this accumulator model to predict the effects of accumulators on the whole automotive A/C systems.

- The present research is for the A/C system with R-134a as refrigerant and PAG oil as the lubricant; the future work should be transferred to the system with carbon dioxide as refrigerant, the next generation A/C system.

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Appendix A: The computational programs

A.1 The accumulator simulation program (Matlab)

% Simulation of Performance of Accumulator

% in Automotive Air Conditioning System

format short

% INPUT VALUE

% input values of parameters

Pin=260000; % pressure at inlet (Pa)

Mrate=130/3600; % total mass flow rate in kg/s (130kg/h)

Roil=0.045; % ratio of oil to total mass flow

Hliq=0.22; % height of liquid (m)

% Geometry and dimensions of accumulator

Dout=0.0175; % internal diameter of 3/4" inlet pipe (m)

Dj=0.01; % internal diameter of 1/2" j-tube (m)

Lin=0.0254; % length of inlet pipe (m)

Href=0.0254; % height from top to reflector(m)

L1=0.2; % length of first part of j-tube(m)

L2=0.2; % length of second part of j-tube(m)

L3=0.03; % length of third part of j-tube(m)

Lout=0.05; % length of outlet pipe(m)

Hacc=0.3; % internal height of accumulator (m)

Dacc=0.1; % internal diameter of accumulator(m)

Doil=0.0014; % diameter of oil bleedding hole(m)

Dsiph=0.002; % diameter of anti-siphon hole(m)

Dref=0.096; % diameter of reflector(m)

% CALCULATION

% calculation of temperature,density and viscosity

$$Pin0 = Pin / 100000;$$

$$Tin = 0.3271 * Pin0^3 - 4.3957 * Pin0^2 + 26.713 * Pin0 - 48.618;$$

% density of liquid, vapor and oil at inlet (kg/m³)

$$PHliq = -0.9649 * Pin0^3 + 12.667 * Pin0^2 - 80.183 * Pin0 + 1444.9;$$

$$PHvap = 0.0077 * Pin0^3 - 0.0717 * Pin0^2 + 4.9742 * Pin0 + 0.2885;$$

$$PHoil = 1016.8 - 0.71695 * Tin;$$

% viscosity of liquid, vapor and oil (N*s/m²)

$$MUliq = (-0.0002191 * Tin^3 + 0.039304 * Tin^2 - 3.6494 * Tin + 267.67) * 1E-6;$$

$$MUvap = (11.021 + 0.038599 * Tin) * 1E-6;$$

$$MUoil = \exp(-12.99498 + 3124.094 / (Tin + 273.15));$$

% Calculation of quality

$$F0 = 100000000000;$$

for i=70000:95000

$$Rvap = i / 100000;$$

for j=1:200

Rsiph=j/10000;

% stream 1

PHa=1/(Rvap/PHvap+Roil/PHoil+(1-Rvap-Roil)/PHliq);

Va=4*Mrate/(pi*PHa*Dout^2);

P11=0.24*0.5*PHa*Va^2;

MUa=1/(Rvap/MUvap+Roil/MUoil+(1-Rvap-Roil)/MULiq);

RE12=PHa*Va*Dout/MUa;

if RE12<2320

F12=64/RE12;

else

F12=0.3164*RE12^(-0.25);

end

P12=F12*0.5*PHa*Va^2*Lin/Dout;

K13=(1-Dout/2/Href)^2;

P13=K13*0.5*PHa*Va^2;

P14=0.24*0.5*PHa*Va^2;

$$V15=4*Mrate/PHa/pi/(Dacc^2-Dref^2);$$

$$P15=1.5*0.5*PHa*V15^2;$$

$$Pb=Pin+0.5*PHa*Va^2-P11-P12-P13-P14-P15;$$

% stream 2

$$V2=4*(Rvap-Rsiph)*Mrate/pi/PHvap/Dj^2;$$

$$P21=0.5*0.5*PHvap*V2^2;$$

$$RE22=PHvap*V2*Dj/MUvap;$$

if RE22<2320

$$F22=64/RE22;$$

else

$$F22=0.3164*RE22^{(-0.25)};$$

end

$$P22=F22*0.5*PHvap*V2^2*L1/Dj;$$

$$P23=0.15*0.5*PHvap*V2^2;$$

$$P2dyna=0.5*PHvap*V2^2;$$

% stream 3

$$PH_{liqoil} = (1 - R_{vap}) / ((1 - R_{vap} - R_{oil}) / PH_{liq} + R_{oil} / PH_{oil});$$

$$V_3 = 4 * (1 - R_{vap}) * M_{rate} / (\pi * PH_{liqoil} * D_{oil}^2);$$

$$P_{31} = -PH_{liqoil} * 9.81 * H_{liq};$$

$$P_{32} = 1.5 * 0.5 * PH_{liqoil} * V_3^2;$$

%stream 4

$$V_4 = 4 * R_{siph} * M_{rate} / \pi / PH_{vap} / D_{siph}^2;$$

$$P_{41} = 1.5 * 0.5 * PH_{vap} * V_4^2;$$

% stream 5

$$PH_5 = (1 - R_{siph}) / ((R_{vap} - R_{siph}) / PH_{vap} + R_{oil} / PH_{oil} + (1 - R_{vap} - R_{oil}) / PH_{liq});$$

$$MU_5 = (1 - R_{siph}) / ((R_{vap} - R_{siph}) / MU_{vap} + R_{oil} / MU_{oil} + (1 - R_{vap} - R_{oil}) / MU_{liq});$$

$$V_5 = 4 * (1 - R_{siph}) * M_{rate} / \pi / PH_5 / D_j^2;$$

$$P_{51} = 0.15 * 0.5 * PH_5 * V_5^2;$$

RE52=PH5*V5*Dj/MU5;

if RE52<2320

F52=64/RE52;

else

F52=0.3164*RE52^(-0.25);

end

P52=F52*0.5*PH5*V5^2*L2/Dj;

P5dyna=0.5*PH5*V5^2;

% combined equation

F=abs(P2dyna+P21+P22+P23-P31-
P32)+abs(P5dyna+P21+P22+P23+P51+P52-P41);

if F<F0

F0=F;

Rvap0=Rvap;

Rsiph0=Rsiph;

deltC=P2dyna+P21+P22+P23-P31-P32;

deltD=P5dyna+P21+P22+P23+P51+P52-P41;

i0=i;

j0=j;

end

end

end

% RESULTS

F=F0

Rvap=Rvap0

Rsiph=Rsiph0

deltC

deltD

i0

j0

Pa=Pin;

$T_{in}=0.3271*Pin0^3-4.3957*Pin0^2+26.713*Pin0-48.618;$

% density of liquid, vapor and oil at inlet (kg/m³)

$PH_{liq}=-0.9649*Pin0^3+12.667*Pin0^2-80.183*Pin0+1444.9;$

$PH_{vap}=0.0077*Pin0^3-0.0717*Pin0^2+4.9742*Pin0+0.2885;$

$PH_{oil}=1016.8-0.71695*T_{in};$

% viscosity of liquid,vapor and oil (N*s/m²)

$MU_{liq}=(-0.0002191*T_{in}^3+0.039304*T_{in}^2-3.6494*T_{in}+267.67)*1E-6;$

$MU_{vap}=(11.021+0.038599*T_{in})*1E-6;$

$MU_{oil}=\exp(-12.99498+3124.094/(T_{in}+273.15));$

% stream 1

$PH_a=1/(R_{vap}/PH_{vap}+R_{oil}/PH_{oil}+(1-R_{vap}-R_{oil})/PH_{liq});$

$$Va=4*Mrate/(pi*PHa*Dout^2);$$

$$P11=0.24*0.5*PHa*Va^2;$$

$$MUa=1/(Rvap/MUvap+Roil/MUoil+(1-Rvap-Roil)/MUliq);$$

$$RE12=PHa*Va*Dout/MUa;$$

if RE12<2320

$$F12=64/RE12;$$

else

$$F12=0.3164*RE12^{(-0.25)};$$

end

$$P12=F12*0.5*PHa*Va^2*Lin/Dout;$$

$$K13=(1-Dout/2/Href)^2;$$

$$P13=K13*0.5*PHa*Va^2;$$

$$P14=0.24*0.5*PHa*Va^2;$$

$$V15=4*Mrate/PHa/pi/(Dacc^2-Dref^2);$$

$$P15=1.5*0.5*PHa*V15^2;$$

$$Pb=Pin+0.5*PHa*Va^2-P11-P12-P13-P14-P15;$$

```
% stream 2

V2=4*Rvap*Mrate/pi/PHvap/Dj^2;

P21=0.5*0.5*PHvap*V2^2;

RE22=PHvap*V2*Dj/MUvap;

if RE12<2320;

    F22=64/RE22;

else

    F22=0.3164*RE22^(-0.25);

end

P22=F22*0.5*PHvap*V2^2*L1/Dj;

P23=0.15*0.5*PHvap*V2^2;

P2dyna=0.5*PHvap*V2^2;

deltP2=P21+P22+P23+P2dyna;


% stream 3
```

$$PH_{liqoil} = (1 - R_{vap}) / ((1 - R_{vap} - R_{oil}) / PH_{liq} + R_{oil} / PH_{oil});$$

$$V_3 = 4 * (1 - R_{vap}) * M_{rate} / (\pi * PH_{liqoil} * D_{oil}^2);$$

$$P_{31} = -PH_{liqoil} * 9.81 * H_{liq};$$

$$P_{32} = 1.5 * 0.5 * PH_{liqoil} * V_3^2;$$

$$\Delta P_3 = P_{31} + P_{32};$$

$$F = P_{2dyna} + P_{21} + P_{22} + P_{23} - P_{31} - P_{32};$$

$$P_{bott} = P_b - P_{31};$$

$$P_c = P_b - P_{31} - P_{32};$$

$$P_{cstag} = P_c + 0.5 * PH_{vap} * V_2^2;$$

% stream 5

$$PH_5 = 1 / (R_{vap} / PH_{vap} + R_{oil} / PH_{oil} + (1 - R_{vap} - R_{oil}) / PH_{liq});$$

$$MU_5 = 1 / (R_{vap} / MU_{vap} + R_{oil} / MU_{oil} + (1 - R_{vap} - R_{oil}) / MU_{liq});$$

$$V_{51} = 4 * M_{rate} / (\pi * PH_5 * D_j^2);$$

$$P_{51} = 0.15 * 0.5 * PH_5 * V_{51}^2;$$

$$RE_{52} = PH_5 * V_{51} * D_j / MU_5;$$

if RE52<2320

F52=64/RE52;

else

F52=0.3164*RE52^(-0.25);

end

P52=F52*0.5*PH5*V51²*(L2+L3)/Dj;

V53=4*Mrate/pi/PH5/Dout²;

RE53=PH5*V53*Dout/MUa;

if RE53<2320

F53=64/RE53;

else

F53=0.3164*RE53^(-0.25);

end

P53=F53*0.5*PH5*V53²*Lout/Dout;

P54=0.24*0.5*PH5*V53²;

Pestag=Pcstag-P51-P52-P53-P54;

$$Pe = Pestag - 0.5 * PH5 * V53^2;$$

%

$$Rvap = Rvap * 100$$

$$Pa = Pa / 1000$$

$$Pb = Pb / 1000$$

$$Pbott = Pbott / 1000$$

$$Pc = Pc / 1000$$

$$Pe = Pe / 1000$$

$$Pa = Pa / 100;$$

$$Pb = Pb / 100;$$

$$Pbott = Pbott / 100;$$

$$Pc = Pc / 100;$$

$$Pe = Pe / 100;$$

$$Ta = 0.3271 * Pa^3 - 4.3957 * Pa^2 + 26.713 * Pa - 48.618$$

$$Tb = 0.3271 * Pb^3 - 4.3957 * Pb^2 + 26.713 * Pb - 48.618$$

$$T_{bott}=0.3271*P_{bott}^3-4.3957*P_{bott}^2+26.713*P_{bott}-48.618$$

$$T_c=0.3271*P_c^3-4.3957*P_c^2+26.713*P_c-48.618$$

$$T_e=0.3271*P_e^3-4.3957*P_e^2+26.713*P_e-48.618$$

A.2 Determination of point 2s in isentropic compression process

Explanation:

When calculating isentropic efficiency of compressor with consideration of two-phase flow and oil concentration, we need to know the states of compressor outlet in isentropic compression. At compressor inlet, pressure, temperature, vapour quality and oil concentration are measured. At point 2s, the pressure and oil concentration are the same as that of compressor inlet. The temperature and vapour quality can be calculated by solving two equations. One is the entropy constant equation, and another is the Grebner and Crawford (1992) model. This program is for solving these two equations.

```
%get mixture temperature T (C) and quality x from mixture pressure P (kPa),entropy s
%calculation liquid refrigerant fraction w from mixture pressure P, mixture
%temperature T and saturated temperature t
%w=0.069
%P=19.1
%T=526.5
```

%t=455.3

%y=(1-w)*((-7.1524667E-02+5.9851602E-02/w^0.5)+(1.7133122E-03-2.9748674E-03/w^0.5+1.7002776E-03/w-3.7562142E-04/w^1.5+2.9984011E-05/w^2)*P)+1-T/t

%P=((T-t)/t/(1-w)-(-7.1524667E-02+5.9851602E-02/w^0.5))/(1.7133122E-03-2.9748674E-03/w^0.5+1.7002776E-03/w-3.7562142E-04/w^1.5+2.9984011E-05/w^2)

%w=0.7583;

P=[1741.21

1435.15

1582.34

1847.20

1786.24

1760.53

1526.76

1472.71

1453.47

1590.69];

%s1=1.611

s1=[1.6114

1.6142

1.6022

1.6051

1.6163

1.6181

1.6010

1.6169

1.6141

1.6443];

[m,n]=size(P);

for j=1:m

```

F0=1000000;
for i=1:9999
w(j)=i/10000;
A(j)=-7.1524667E-02+5.9851602E-02/w(j)^0.5;
B(j)=1.7133122E-03-2.9748674E-03/w(j)^0.5+1.7002776E-03/w(j)-3.7562142E-
04/w(j)^1.5+2.9984011E-05/w(j)^2;
t(j)=6.8404E-09*P(j)^3 - 4.0403E-05*P(j)^2 + 9.9768E-02*P(j) - 2.6071E+01;%saturated
temperature
%at point 2s (C)
tR(j)=t(j)*1.8+32;%saturated temperature at point 2s (R)
TR(j)=tR(j)+tR(j)*(1-w(j))*(A(j)+B(j)*(P(j)/6.8948));%mixture temperature (R)
T(j)=(TR(j)-32)/1.8;%mixture temperature (C)
svs=-9.6551E-08*t(j)^3 + 8.4047E-06*t(j)^2 - 5.6635E-04*t(j) + 1.7268E+00;%vapour entropy
at saturated temperature
sv=0.004*(T(j)-t(j))+svs;%vapour entropy at 2s mixture temperature
sl=6.1100E-08*T(j)^3 - 6.4137E-06*T(j)^2 + 4.9104E-03*T(j) + 1.0001E+00;%liquid entropy at
2s mixture temperature
so=1.2256*sl;%oil entropy at 2s mixture temperature
x(j)=1-0.04/(1-w(j)); %vapour quality at point 2s
s2(j)=x(j)*sv+(1-x(j)-0.04)*sl+0.04*so;%total entropy at 2s mixture temperature
F(j)=s2(j)-s1(j);

%F=(1-w(j))*((-7.1524667E-02+5.9851602E-02/w(j)^0.5)+(1.7133122E-03-2.9748674E-
03/w(j)^0.5+1.7002776E-03/w(j)-3.7562142E-04/w(j)^1.5+2.9984011E-05/w(j)^2)*P(j)+(1-
T(j)/t(j));
    if abs(F(j))<abs(F0);
        F0=F(j);
        w0=w(j);
    end
end
w(j)=w0;%get refrigerant liquid fraction
A(j)=-7.1524667E-02+5.9851602E-02/w(j)^0.5;

```

```

B(j)=1.7133122E-03-2.9748674E-03/w(j)^0.5+1.7002776E-03/w(j)-3.7562142E-
04/w(j)^1.5+2.9984011E-05/w(j)^2;
t(j)=6.8404E-09*P(j)^3 - 4.0403E-05*P(j)^2 + 9.9768E-02*P(j) - 2.6071E+01;%saturated
temperature
%at point 2s (C)
tR(j)=t(j)*1.8+32;%saturated temperature at point 2s (R)
TR(j)=tR(j)+tR(j)*(1-w(j))*(A(j)+B(j)*(P(j)/6.8948));%mixture temperature (R)
T(j)=(TR(j)-32)/1.8;%mixture temperature (C)
svs=-9.6551E-08*t(j)^3 + 8.4047E-06*t(j)^2 - 5.6635E-04*t(j) + 1.7268E+00;%vapour entropy
at saturated temperature
sv=0.004*(T(j)-t(j))+svs;%vapour entropy at 2s mixture temperature
s1=6.1100E-08*T(j)^3 - 6.4137E-06*T(j)^2 + 4.9104E-03*T(j) + 1.0001E+00;%liquid entropy at
2s mixture temperature
so=1.2256*s1;%oil entropy at 2s mixture temperature
x(j)=1-0.04/(1-w(j)); %vapour quality at point 2s
s2(j)=x(j)*sv+(1-x(j)-0.04)*s1+0.04*so;%total entropy at 2s mixture temperature
F(j)=s2(j)-s1(j);
end
disp(w')
disp(T')

```

A.3 Calculation of liquid refrigerant fraction

Explanation:

Calculation of liquid refrigerant fraction w from mixture pressure P (psi), mixture temperature T (R) and saturated temperature t (R)

%w=0.069

%P=19.1

%T=526.5

%t=455.3

%y=(1-w)*((-7.1524667E-02+5.9851602E-02/w^0.5)+(1.7133122E-03-2.9748674E-03/w^0.5+1.7002776E-03/w-3.7562142E-04/w^1.5+2.9984011E-05/w^2)*P)+1-T/t

%P=((T-t)/t/(1-w)-(-7.1524667E-02+5.9851602E-02/w^0.5))/(1.7133122E-03-2.9748674E-03/w^0.5+1.7002776E-03/w-3.7562142E-04/w^1.5+2.9984011E-05/w^2)

%w=0.7583;

P=[207.52

268.58

295.62

195.39

245.81

262.17

174.66

213.68

228.09];

T=[593.21

625.08

639.00

589.35

622.92

639.34

585.65

624.46

640.77];

t=[588.6

608.094
 615.69
 584.19
 601.254
 606.204
 576.198
 590.742
 595.566];

```
[m,n]=size(P);
for j=1:m
F0=1000000;
for i=1:1000
w(j)=i/1000;
F=(1-w(j))*((-7.1524667E-02+5.9851602E-02/w(j)^0.5)+(1.7133122E-03-2.9748674E-
03/w(j)^0.5+1.7002776E-03/w(j)-3.7562142E-04/w(j)^1.5+2.9984011E-05/w(j)^2)*P(j))+(1-
T(j)/t(j));
    if F^2<F0^2;
        F0=F;
        w0=w(j);
    end
end
w(j)=w0;
F=F0;

end
disp(w')
```

Appendix B: User guide of the accumulator simulation program

In this MATLAB program, we define a set of inputs that fully describes the problem. At first, these are the geometry and dimensions of the accumulator. Then, we take the total mass flow rate, pressure at inlet of accumulator, the ratio of oil flow rate to total mass flow and the liquid height inside accumulator as inputs. Outputs are quality, and other parameters, such pressures and temperature at various positions in accumulator.

1. Description of input values of parameters

Pin: pressure at inlet of accumulator (Pa)

Mrate: total mass flow rate (kg/s)

Roil: ratio of oil flow rate to total mass flow rate

Hliq: height of liquid in accumulator (m)

2. Description of geometry and dimensions of accumulator

Dout: internal diameter of inlet pipe (m)

Dj: internal diameter of J-tube (m)

Lin: length of inlet pipe (m)

Href: height from top to reflector (m)

L_1 : length of first part of J-tube (m)

L_2 : length of second part of J-tube (m)

L_3 : length of third part of J-tube (m)

L_{out} : length of outlet pipe (m)

H_{acc} : internal height of accumulator (m)

D_{acc} : internal diameter of accumulator (m)

D_{oil} : diameter of oil bleeding hole (m)

D_{siph} : diameter of anti-siphon hole (m)

D_{ref} : diameter of reflector (m)

Figure B.1 The dimensions of accumulator

Figure B.2 Flow streams in an accumulator

3. Description of results

R_{vap} : the quality of refrigerant

P_a : pressure at point a (kPa)

P_b : pressure at point b (kPa)

P_{bott} : pressure at bottom (kPa)

P_c : pressure at point c (kPa)

P_e : pressure at point e (kPa)

T_a : temperature at point a ($^{\circ}\text{C}$)

T_b: temperature at point b (°C)

T_{bott}: temperature at bottom (°C)

T_c: temperature at point c (°C)

T_e: temperature at point e (°C)

Appendix C: Experimental setup

C.1 AACTS operating instruction

1. Leak testing

This is done after any part is removed/replaced from the system. Charge the stand with Nitrogen to 500 kPa (use the AACTS LabVIEW program to monitor the pressures when charging stand). Use “snoop” to check for leaks around the fittings near any replaced/removed parts. If a leak is found, tighten the fitting – remember the stand is charged so do not loosen part! Check with “Snoop” again and make sure the leak is sealed. Use AACTS Labview program to monitor the pressures in the system and to see if the pressure in the system drops – usually wait 15 minutes and if the pressures remain constant then there are no leaks. If there are no leaks then discharge the system slowly using a highside coupler and a rag. If there is a leak, use “snoop” to find leakage spot and seal it. Vacuum the system for 10 minutes by attaching the highside coupler on the vacuum to the highside port after the condenser (there is a highside port on the line out of the compressor but you can remove oil from the system if you vacuum here). Monitor pressures, they should not rise. If they do re-pressurize the system, repeat pressure test with Nitrogen as explained above and re-check for leaks. Note: Refrigerant can come out of the oil and re-pressurize the system if the system has just been recovered and has not been opened to the atmosphere for a long period of time. It is important to note that

there are gauges reading pressure at the compressor inlet and outlet and these can be used to check system pressures at any time.

2. Charging-up the AACTS

The desired amount of charge in the AACTS is 1.55kg. To charge the AACTS, initially use a charge of 1.80kg, recover the lines and record the actual amount in the system. If it is less than the desired amount use the can from the OIC stand and add in additional amount needed. Refer to the Robinair operating manual for how to charge up a system. Charging must be done through the high side charge port only. After the charging-up is complete, uncouple the red hose and recover the refrigerant left in the hose. Disconnect hoses with extreme caution! All hoses may contain liquid refrigerant under pressure.

3. Start-up Procedure

Turn on the motor and clutch and slowly increase compressor speed to ~700 rpm. Turn on the evaporator heaters [either constant power (no less than 3500 Watts otherwise the evaporator chamber temperature will not rise above ambient temperature and therefore there is no heat load) or constant temperature (this can be adjusted to desired level by using the temperature controller on the evaporator box)]. Turn on the heaters in the accumulator box at a constant temperature (this can be adjusted using the temperature controller in the accumulator box). Set the condenser bath to the desired temperature. Slowly change the rpm to the desired speed for test. Whenever any system conditions are changed, the liquid level in the separator may change. To maintain the same liquid level

in the separator for each test condition the level may need to be adjusted using the large knob and the smaller knob for, fine tuning, located on the line exiting the bottom of the separator. To obtain steady state conditions, let system run at set conditions for at least 30 minutes before recording data (the data should be recorded for at least 15 minutes after this point). The LabVIEW program has a switch which allows you to choose whether you want to record transient data or steady state data.

4. Shutdown Procedure

To turn off the AACTS you must slow down the compressor to ~ 700 rpm. Switch from constant temperature to constant power in the evaporator chamber and set the evaporator heater to 0 Watts, but kept the evaporator mixer on. Set the condenser water bath temperature to 25°C but keep the condenser mixer on. Turn off the heaters in the accumulator box. Wait until the evaporator chamber temperature reaches 20°C and then you can slowly reduce the compressor speed down to 0 rpm and turn off the clutch. The evaporator chamber temperature will slowly heat up but should not go above 60°C , if it does, start up the AACTS again and repeat shutdown procedure. Make sure you leave the mixer on in the evaporator chamber for at least 10 minutes after shutdown to prevent hotspots in the box. After 10 minutes, you can turn the power off to each component in the stand. In case of a power outage while running the AACTS, the evaporator chamber temperature will quickly rise because there is no refrigerant moving through the evaporator to remove the heat in the chamber and you may need to remove the evaporator box lid and remove the foam insulation from the top of the evaporator to cool it down.

5. Oil in circulation test

We use the following conditions for an Oil In Circulation Test:

Compressor: 700 rpm

Condenser Water Bath Temperature: 35°C

Evaporator Chamber Temperature: 60°C.

Four samples are taken from the highside port after the condenser using the oil in circulation (OIC) canisters. For the first 3 samples the amount of refrigerant and oil removed by each canister is estimated and is put back into the system after each sample is taken but for the 4th sample canister no oil or refrigeration is put back. The actual amount of oil and refrigerant removed by the sample canisters is calculated after they are recovered and this is done using an Excel spreadsheet. This amount is subtracted from the known total amount that was put back into the system during testing and these calculated values are the amount of oil and refrigerant that needs to be put back into the system to keep the OIC and refrigerant levels the same as they were prior to testing.

Table C.1 AACTS parameter settings used for validation test runs

	Comp	Heater	Cond water	Comp suction	Comp discharge	Comp discharge	Evap outlet	Flow rate
	rpm	(%)	Bath T (°C)	P (kPa)	P (kPa)	T (°C)	T (°C)	(kg/hr)
Idle	700	35	65	472	2220	86.3	12.7	109
City	1500	67	36	293	1433	87.1	-2.9	138
Hwy	2000	74	25	289	1370	71.8	-2.3	176

C.2 Validation test of refrigerant permeation rate

The AACTS can be considered as a control volume system filled with R-134a, and then the following equation is given:

$$PV = mRT = m \cdot \frac{R^\circ}{M} T \quad (4.1)$$

where, P is the pressure; V is the total volume inside the AACTS; m is the mass of the refrigerant vapour inside the AACTS; R° is the universal gas constant; M is the molecular mass of R-134a and T is the temperature.

At state 1, we have equation:

$$P_1 V = m_1 \cdot \frac{R^\circ}{M} T \quad (4.2)$$

Then the state changes to state 2 because of leaks:

$$P_2 V = m_2 \cdot \frac{R^\circ}{M} T \quad (4.3)$$

$$(P_1 - P_2)V = (m_1 - m_2) \cdot \frac{R^\circ}{M} T \quad (4.4)$$

Take $\Delta P = P_1 - P_2$ and $\Delta m = m_1 - m_2$

$\dot{P} = \frac{\Delta P}{t}$ and $\dot{m} = \frac{\Delta m}{t}$ are pressure drop rate and mass flow rate, respectively.

where, t is time, then,

$$\dot{P}V = \dot{m} \cdot \frac{R^o}{M} T \quad (4.5)$$

$$\dot{m} = \frac{\dot{P}VM}{R^o T} \quad (4.6)$$

For the value of the AACTS, we take the value of V as $1 \times 10^{-3} \text{ m}^3$, $\dot{p} = 0.52 \text{ kPa/day} = 0.52 \times 10^3 \text{ N/m}^2 \text{ per day}$, $M = 102.3 \text{ g/mol}$, $R^o = 8.314 \text{ J/mol} \cdot \text{K}$, $T = 21 + 273 = 294 \text{ K}$.

Through calculation, we have, $\dot{m} = 0.218 \text{ g/day}$

The relationship of R-134a release rate to pressure inside the AACTS is:

$$P - P_{atm} = K \times \frac{1}{2} \rho v^2 = K \times \frac{1}{2} \rho \left(\frac{\dot{m}}{\rho A} \right)^2 \quad (4.7)$$

where, K, ρ, A are constants, Equation (4.7) can be simplified as:

$$\dot{m} = f \sqrt{P - P_{atm}} \quad (4.8)$$

Then, the R-134a release rate can be calculated in the following situation:

When the AACTS is off, the pressure is the saturated pressure at 21°C , $P = 600 \text{ kPa}$

$$\dot{m} = 0.218 \times \sqrt{\frac{600 - 100}{360 - 100}} = 0.3 \text{ g/day} \quad (4.9)$$

The total charged R-134a is $1.8 \times 10^3 \text{ g}$, so the time the R-134a release completely is:

$$t = \frac{1800}{0.3} = 6000 \text{ days} = 16.4 \text{ years} \quad (4.10)$$

When the AACTS is run continuously, the leaks are all located at high-side

The pressure is taken as $P=2000$ kPa

$$\dot{m} = 0.218 \times \sqrt{\frac{2000 - 100}{360 - 100}} = 0.59 \text{ g/day} \quad (4.11)$$

The time the R-134a release completely is:

$$t = \frac{1800}{0.59} = 3050 \text{ days} = 8.3 \text{ years} \quad (4.12)$$

According the above analysis, the leaks are very fine, which is close to actual situation of an automotive A/C system, and the effect of leaks to the results is small enough and can be ignored.

C.3 AACTS instrumentation wiring table

Table C.2 AACTS instrumentation wiring

Max name:		Instrumentation wiring				
	Function	Wire	From	To	Destination	
Compressor stand						
Prsscompin	Inlet pressure	Pa-c1-red	Pa-c1-1	Tbx-1303-ch4+	1102 ch4+	
		Pa-c1-blk	Pa-c1-8	Tbx-1303-ch4-	1102 ch4-	
		Pa-c1-grn	Pa-c1-4	Phoenix-1	Excitation +24v	
		Pa-c1-wht	Pa-c1-9	Phoenix-16	Excitation gnd	
Tempcompin	Inlet temp	Tk-c1-yel	Tk-c1-v+	Tbx-1303-ch1+	1102 ch1+	
		Tk-c1-red	Tk-c1-v-	Tbx-1303-ch1-	1102 ch1-	
Presscompout	Outlet pressure	Pa-c2-red	Pa-c2-1	Tbx-1303-ch5+	1102 ch5+	
		Pa-c2-blk	Pa-c2-8	Tbx-1303-ch5-	1102 ch5-	
		Pa-c2-grn	Pa-c2-4	Phoenix-2	Excitation +24v	
		Pa-c2-wht	Pa-c2-9	Phoenix-17	Excitation gnd	
Tempcompout	Outlet temp	Tk-c2-yel	Tk-c2-v+	Tbx-1303-ch2+	1102 ch2+	
		Tk-c2-red	Tk-c2-v-	Tbx-1303-ch2-	1102 ch2-	
Tempcompbody	Body temp	Tk-c3-yel	Tk-c3-v+	Tbx-1303-ch3+	1102 ch3+	
		Tk-c3-red	Tk-c3-v-	Tbx-1303-ch3-	1102 ch3-	
Rpmcomp	Speed sensor	Ss-c1-grn	Ss-c1-4	Tbx-1303-ch6+	1102 ch6+	Resistor required
		Ss-c1-wht	Ss-c1-5	Tbx-1303-ch6-	1102 ch6-	
		Ss-c1-red	Ss-c1-1	Phoenix-12	5v	
		Ss-c1-blk	Ss-c1-2	Phoenix-13	Gnd	
	Torque	Proximity Switch 24v red	Proximity switch 24v	Phoenix-3	24v	
		Proximity Switch 24v black	Proximity switch gnd	Phoenix-18	Gnd	
		Wt-c1-grn	Wt-c1-5	Ultra slimpak -		
		Wt-c1-wht	Wt-c1-4	Ultra slimpak +		
Trqcomp			Ultra slimpak +	Tbx-1303-ch7+	1102 ch7+	
			Ultra slimpak -	Tbx-1303-ch7-	1102 ch7-	
		Wt-c1-red	Wt-c1-1	Phoenix-26	Excitation +12v	Excitation supply
			R1+, 47k	Phoenix 26		
			R1-, 47k	Phoenix-27		
			R2+, 47k	Phoenix-27		Excitation monitor
			Phoenix-27	Tbx1303-ch22+	1102 ch22+	

			R2-, 47k	Phoenix-28		
		Wt-c1-blk	Wt-c1-2	Phoenix-28	Excitation 0v	Excitation return
Trqexc			Phoenix-23	Tbx1303-ch22-	1102 ch22-	
Speed control	Speed control	Sc-c1-blk	Sc-c1-2	Tbx1325-supply 0	1124 ch0 i+	4-20ma
		Sc-c1-wht	Sc-c1-4	Tbx1325-isink 0	1124 ch0 i-	
Motor stop/run	Motor stop/run	Sc-c2-red	Sc-c2-1	Rssd 25b-1	1161 ch 0 n.o.	
			Phoenix-4	Rssd 25b-2	1161 ch 0 com	
			Rssd 25b-2	Phoenix-4	Excitation +24v	
		Sc-c2-grn	Sc-c2-5	Phoenix-19	Excitation gnd	
Clutch on/off	Clutch on/off	Cc-c1-blk	Cc-c1-1	Rssd 25b-3	1161 ch 1 n.o.	
			Phoenix-5	Rssd 25b-4	1161 ch 1 com	
			Rssd 25b-4	Phoenix-5	Excitation +24v	
		Cc-c1-wht	Cc-c1-2	Phoenix-20	Excitation gnd	
Condenser stand						
Tempcndwtr	Bath temp	Tk-d1-yel	Tk-d1-v+	Tbx-1303-ch8+	1102 ch8+	
		Tk-d1-red	Tk-d1-v-	Tbx-1303-ch8-	1102 ch8-	
Swevplvl	Lo level switch	Ls-d1-wht	Ls-d1-1	Tbx-1303-ch9+	1102 ch9+	
		Ls-d1-grn	Ls-d1-4	Phoenix 30	Excitation +5v	
			Tbx-1303-ch9-	Phoenix 29	Excitation gnd	
			Phoenix 29	Tbx-1303-ch9-	1102 ch9-	
	Valve on	Vc-d1-blk	Vc-d1-1	Rssd 25b-7	1161 ch 3 n.o.	
			Phoenix-7	Rssd 25b-8	1161 ch 3 com	
			Rssd 25b-8	Phoenix-7	Excitation +24v	
		Vc-d1-wht	Vc-d1-2	Phoenix-22	Excitation gnd	
	Mixer on	Mc-d1-blk	Mc-d1-1	Rssd 25b-5	1161 ch 2 n.o.	
			Phoenix-6	Rssd 25b-6	1161 ch 2 com	
			Rssd 25b-6	Phoenix-6	Excitation +24v	
		Mc-d1-wht	Mc-d1-2	Phoenix-16	Excitation gnd	
Evaporator stand						
Tempevpwtr	Chamber temp	Tk-e1-yel	Tk-e1-v+	Tbx-1303-ch10+	1102c ch10+	
		Tk-e1-red	Tk-e1-v-	Tbx-1303-ch10-	1102c ch10-	
	Heater control	Hc/lc-e1-red	Hc/lc-e1-1	Tbx1325-supply 1	1124 ch1 i+	4-20ma
		Hc/lc-e1-blk	Hc/lc-e1-2	Tbx1325-isink 1	1124 ch1 i-	
Not hooked up	Lo level alarm	Hc/lc-e1-grn	Hc/lc-e1-5	Tbx-1303-ch11+	1102 ch11+	
		Hc/lc-e1-wht	Hc/lc-e1-4	Tbx-1303-ch11-	1102 ch11-	
	Mixer on	Mc-e1-blk	Mc-e1-1	Rssd 25b-9	1161 ch 4 n.o.	
			Phoenix-8	Rssd 25b-10	1161 ch 4 com	
			Rssd 25b-10	Phoenix-8	Excitation +24v	
		Mc-e1-wht	Mc-e1-2	Phoenix-23	Excitation gnd	
Acc heater power monitor	Heater power monitor	Jr-e1-red	Jr-e1-1	Tbx-1303-ch12+	1102 ch12+	0-10vdc

		Jr-e1-blk	Jr-e1-2	Tbx-1303-ch12-	1102 ch12-
Tempgasin	Gas temp in	Tk-m3-yel	Tk-m3-v+	Tbx-1303-ch27+	1102 ch27+
		Tk-m3-red	Tk-m3-v-	Tbx-1303-ch27-	1102 ch27-
Prssevapin	Gas press in	Pa-m3-red	Pa-m3-1	Tbx-1303-ch30+	1102 ch30+
		Pa-m3-blk	Pa-m3-8	Tbx-1303-ch30-	1102 ch30-
		Pa-m3-grn	Pa-m3-4	Phoenix-7	Excitation +24v
		Pa-m3-wht	Pa-m3-9	Phoenix-22	Excitation gnd
Tempgasout	Gas temp out	Tk-m5-yel	Tk-a2-v+	Tbx-1303-ch26+	1102 ch26+
		Tk-m5-red	Tk-a2-v-	Tbx-1303-ch26-	1102 ch26-
Prssmxrgas	Gas press out	Pa-m5-red	Pa-a2-1	Tbx-1303-ch20+	1102 ch20+
		Pa-m5-blk	Pa-a2-8	Tbx-1303-ch20-	1102 ch20-
		Pa-m5-grn	Pa-a2-4	Phoenix-6	Excitation +24v
		Pa-m5-wht	Pa-a2-9	Phoenix-21	Excitation gnd
Evaporator mixer					
Evap mixer					
Inst power on	Inst power on	Jc-m1-red	Jc-m1-cr-2	Rssd 25b-11	1161 ch 5 n.o.
			Phoenix-9	Rssd 25b-12	1161 ch 5 com
			Rssd 25b-12	Phoenix-9	Excitation +24v
		Jc-m1-blk	Jc-m1-cr-7	Phoenix-24	Excitation gnd
Flowmeter					
Tempmxrlin	Liquid temp in	Tk-m1-yel	Tk-m1-v+	Tbx-1303-ch15+	1102 ch15+
		Tk-m1-red	Tk-m1-v-	Tbx-1303-ch15-	1102 ch15-
Prssmxrlin	Liquid press in	Pa-m1-red	Pa-m1-1	Tbx-1303-ch18+	1102 ch18+
		Pa-m1-blk	Pa-m1-8	Tbx-1303-ch18-	1102 ch18-
		Pa-m1-grn	Pa-m1-4	Phoenix-10	Excitation +24v
		Pa-m1-wht	Pa-m1-9	Phoenix-25	Excitation gnd
Massflow	Total mass flow	Fm-m2-red	Fm-m2-26	Tbx-1303-ch14+	1102 ch14+
			Resistor+, 249	Tbx-1303-ch14+	
			Resistor-, 249	Tbx-1303-ch14-	
		Fm-m2-blk	Fm-m2-27	Tbx-1303-ch14-	1102 ch14-
Separator					
Tempsepgas	Temperature gas	Pa-m2-red	Pa-m2-4	Tbx-1303-ch23+	1102 ch23+
		Pa-m2-blk	Pa-m2-9	Tbx-1303-ch23-	1102 ch23-
Tempsepliquid	Temperature liquid	Tk-f2-yel	Tk-f2-v+	Tbx-1303-ch21+	1102 ch21+
		Tk-f2-red	Tk-f2-v-	Tbx-1303-ch21-	1102 ch21-
Anemometer					
Not hooked up	Velocity	-wht		Tbx-1303-ch19+	1102 ch19+
		-grn		Tbx-1303-ch19-	1102 ch19-
		-blk		Phoenix-14	Excitation gnd
		-red		Phoenix-15	Excitation +16.5v
Gasflow	Gas flow	Fm-m1-red		Tbx-1303-ch13+	1102 ch13+

		Fm-m1-blk		Tbx-1303-ch13-	1102 ch13-
			Resistor+, 249	Tbx-1303-ch13+	
			Resistor-, 249	Tbx-1303-ch13-	
Ambient					
Tempamb	Ambient temp	Tk-m2-yel	Tk-m2-v+	Tbx-1303-ch0+	1102 ch0+
		Tk-m2-red	Tk-m2-v-	Tbx-1303-ch0-	1102 ch0-
Accumulator stand					
	Power on				
Tempaccin	Temp in	Tk-a1-yel	Tk-a1-v+	Tbx-1303-ch25+	1102 ch25+
		Tk-a1-red	Tk-a1-v-	Tbx-1303-ch25-	1102 ch25-
Prssacctop	Press top	Pa-a1-red	Pa-a1-1	Tbx-1303(b)-ch14+	1102c ch14+
		Pa-a1-blk	Pa-a1-8	Tbx-1303(b)-ch14-	1102c ch14-
		Pa-a1-grn	Pa-a1-4	Phoenix-2	Excitation +24v
		Pa-a1-wht	Pa-a1-9	Phoenix-17	Excitation gnd
Tempaccout	Temp out	Tk-a2-yel	Tk-a2-v+	Tbx-1303-ch17+	1102 ch17+
		Tk-a2-red	Tk-a2-v-	Tbx-1303-ch17-	1102 ch17-
Prssaccout	Press out	Pa-a2-red	Pa-a2-1	Tbx-1303-ch29+	1102 ch29+
		Pa-a2-blk	Pa-a2-8	Tbx-1303-ch29-	1102 ch29-
		Pa-a2-grn	Pa-a2-4	Phoenix-5	Excitation +24v
		Pa-a2-wht	Pa-a2-9	Phoenix-20	Excitation gnd
Prssaccbottom	Press bottom	Pa-a3-red	Pa-a3-1	Tbx-1303(b)-ch13+	1102c ch13+
		Pa-a3-blk	Pa-a3-8	Tbx-1303(b)-ch13-	1102c ch13-
		Pa-a3-grn	Pa-a3-4	Phoenix-2	Excitation +24v
		Pa-a3-wht	Pa-a3-9	Phoenix-17	Excitation gnd
Prssaccorifice	Press orifice	Pa-a4-red	Pa-a4-1	Tbx-1303(b)-ch12+	1102c ch12+
		Pa-a4-blk	Pa-a4-8	Tbx-1303(b)-ch12-	1102c ch12-
		Pa-a4-grn	Pa-a4-4	Phoenix-2	Excitation +24v
		Pa-a4-wht	Pa-a4-9	Phoenix-17	Excitation gnd
Diffprssacc	Diff press	Pd-a1-red	Pd-a1-2	Tbx-1303-ch24+	1102 ch24+
			Resistor+, 249	Tbx-1303-ch 24+	
			Resistor-, 249	Tbx-1303-ch 24-	
		Pd-a1-grn	Pd-a1-1	Phoenix 26	Excitation +12v
Acc fan on	Fan on	Fc-f1-red	Fc-f1-?1	Rssd 25b-15	1161 ch 6 n.o.
			Phoenix-10	Rssd 25b-16	1161 ch 6 com
			Rssd 25b-16	Phoenix-10	Excitation +24v
		Fc-f1-blk	Fc-f1-?2	Phoenix-25	Excitation gnd
Acc heater on	Heater on	Hc-f1-red	Hc-f1-?1	Rssd 25b-17	1161 ch 7 n.o.
			Phoenix-1	Rssd 25b-18	1161 ch 7 com
			Rssd 25b-18	Phoenix-1	Excitation +24v
		Hc-f1-blk	Hc-f1-?2	Phoenix-16	Excitation gnd

Excitation
supply

Acc heater power monitor	Heater power monitor	Jr-e2-red	Jr-e2-1	Tbx-1303-ch31+	1102 ch31+	4-20ma
		Jr-e2-blk	Jr-e2-2	Tbx-1303-ch31-	1102 ch31-	Resistor 249 ohms
Tempacctop	Temp top can	Tk-a3-yel	Tk-a3-v+	Tbx-1303-ch16+	1102 ch16+	
		Tk-a3-red	Tk-a3-v-	Tbx-1303-ch16-	1102 ch16-	
Tempaccbottom	Temp bottom can	Tk-a5-yel	Tk-a5-v+	Tbx-1303-ch28+	1102 ch28+	
		Tk-a5-red	Tk-a5-v-	Tbx-1303-ch28-	1102 ch28-	
	Temp acc box grid	Tk-a4-yel	Tk-a4-v+			*not Connected
		Tk-a4-red	Tk-a4-v-			
Tempaccgas	Temp gas	Tk-a6-yel	Tk-a6-v+	Tbx-1303(b)-ch2+	1102c ch2+	
		Tk-a6-red	Tk-a6-v-	Tbx-1303(b)-ch2-	1102c ch2-	
Tempaccliquid	Temp liquid	Tk-a7-yel	Tk-a7-v+	Tbx-1303(b)-ch0+	1102c ch0+	
		Tk-a7-red	Tk-a7-v-	Tbx-1303(b)-ch0-	1102c ch0-	
Tempacc Underdeflector	Temp under deflector	Tk-a8-yel	Tk-a8-v+	Tbx-1303(b)-ch1+	1102c ch1+	
		Tk-a8-red	Tk-a8-v-	Tbx-1303(b)-ch1-	1102c ch1-	
Comp power monitor	Comp power monitor	Red			1102c ch16+	0-8kohms = 0-8hp
		Black			1102c ch16-	
Evap power monitor	Evap power monitor	Red			1102c ch15+	0-10kohms = 0-10hp
		Black			1102c ch15-	