Design, build and test of Ejector Based Multi-Evaporator Refrigeration System

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Summary

This thesis proposed a novel ejector based multi-evaporator refrigeration system (MERS). The energy inefficiency of conventional multi-evaporator refrigeration system was analyzed and this novel ejector based multi-evaporator refrigeration system was introduced. The working principle is described in detail and the possible advantage and disadvantage is discussed.

The experimental platform is built to conduct the comparison study between conventional MERS and ejector based MERS. The system is introduced from architecture to detailed functional programming. Experiments have been done on the experimental platform to compare the energy consumption of these two systems. By analyzing the performance of the novel ejector based multi-evaporator refrigeration system, several improvements and future works are proposed.
Chapter 1 Introduction

1.1 Motivations and Background

The refrigeration industry evolved in response to the pressing need to preserve and transport food for expanding populations. It continued to grow as comfort and industrial air conditioning applications were developed. These applications can be divided into four groups: food production and distribution, chemical and process industry, special application of refrigeration and industrial and comfort air conditioning. The energy consumption of the refrigeration system is quite large in industry as well as domestic usage. For instance, statistical data shows that air conditioner and refrigerator account for 28% of total home energy consumption in US. For hot and humid tropical country such as Singapore, this percentage can even be over 50%. Among all types of cooling systems, refrigerated shipping containers are widely used in transporting perishable goods. It forms an important part of the global food chain, which accounts for over 30% of the world's food supply. The number of refrigerated shipping containers operated throughout the world today is over 100,000 and the growth for refrigerated shipping containers in the world market has been estimated to be about 10% annually. Because refrigerated shipping containers need to carry a huge variety of cargoes under wide climatic conditions variations, they are subjected to very demanding performance requirements. Typically, three evaporating temperatures: –30°C; –5°C and +7°C are required for freezing food, perished food and space cooling, respectively. Generally, the cold areas on the earth have very different climates than those hot areas, the ambient temperature could vary from −40°C to +50°C. In order to address different application needs, the refrigeration system should be designed to be energy efficient and precise temperature control is required in the chill cargo range, especially for temperature sensitive products.

A convenient way to meet the different cooling requirements is to adopt different refrigeration systems to cater for the different loads independently. However, this may not be economically viable due to the high total initial cost. Another alternative is to use multi-evaporator refrigeration systems (MERS) which consists of a single compressor and a single condenser but three evaporators. To reduce the pressure of the higher temperature evaporators to the compressor suction pressure which is the pressure of the lowest temperature evaporator, pressure regulating valves are used to maintain the required pressure in high temperature evaporators. Since refrigerant vapor at higher pressures are first throttled then compressed, and compressor inlet is in superheated region, and the system is energy inefficient as the unrecoverable system pressure loss increases while evaporating pressure differences increase.

To reduce the pressure losses due to the throttle effect of pressure regulating valves, several schemes have been proposed including multistage refrigeration cycles [7, 8], cascade refrigeration cycles [7, 8], expander based MERS [9-11] and ejector based MERS. However, the first three schemes are often used in ultra-low temperature refrigeration (cryogenic refrigeration) or CO2 refrigeration systems where the pressure and temperature of the refrigerant drop greatly after the expansion process. As for MERS applications in
food processing, storage and transportation, the energy saving by such schemes not significant since the temperature and pressure span before and after expansion is small. Therefore, ejector based multi evaporator refrigeration system is considered the only energy saving alternative in such applications.

1.2 Objectives

The main objective of this research work is to design and build an experimental platform for comparison study between the conventional multi-evaporator refrigeration system and the novel ejector based multi evaporator refrigeration system. With this experimental platform, various experiments will be conducted, so as to investigate the performance of ejector based multi evaporator refrigeration system, develop a practical technology for the design and operation of EMERS to recover the pressure losses associated with high temperature evaporators in the conventional ejector based MERS. The target is to achieve 15% COP improvement over the existing MERS. The intended application of this proposed research is for the refrigerated shipping containers. However, the technology developed in this research can be easily extended to other sectors, such as food processing, supermarket, and chemical industries, etc., where MERSs are required.

1.3 Major contributions of the thesis

This thesis presented the working principle of the ejector based multi-evaporator refrigeration system, proposed the hardware and software setup for the experimental platform, conducted comparison experiments between conventional MERS and ejector based MERS, and performed data analysis on the experiment results.

1.4 Organization of the thesis

This thesis is organized as follows. Chapter 2 presents the working principle of ejector based multi evaporator refrigeration system. Chapter 3 introduces design concept detailed information and associated components of this experiment system. In Chapter 4, the experiment results were presented and analyzed. Chapter 5 states the conclusions and presents some possible future works.
Chapter 2 Working Principles of Ejector based MERS

2.1 Working principles of vapor compression refrigeration systems

The working principle of most refrigeration systems are based on the vapor compression refrigeration cycle, where the purpose of this cycle is to use refrigerant to carry heat from a space of lower temperature (cold reservoir) to an environment of high temperature (hot reservoir).

The schematic of common vapor compression refrigeration cycle is as shown in Fig. 1, where there are four basic components in one cycle – Evaporator, Compressor, Condenser, and Expansion Valve. And these four components are connected in a closed loop to ensure the continuous circulating of refrigerant[1].

The \( p-h \) chart reflecting the variation of the refrigerant states for theoretical vapor compression refrigeration cycle is as shown in Fig. 2. The details of this vapor compression cycle are explained below[1].
1. Point 1 is located near the bottom of the condenser where the refrigerant is a saturated liquid at the condensing temperature and pressure. Since the refrigerant at point 1 is always a saturated liquid in this theoretical cycle, the point is located somewhere along the saturated liquid line of the Ph diagram. If either the pressure, temperature or enthalpy of state 1 is known, its location on the diagram is found by following the known variables’ property line to where it intersects the saturated liquid line.

2. In the basic cycle there is no change in the properties of the refrigerant liquid as it flows from the condenser, through the liquid line and into the metering device. Therefore, the state of the liquid approaching the metering device is the same as its condition at point 1. A throttling process occurs within the metering device when the refrigerant’s pressure and temperature are reduced. Since throttling is a constant enthalpy process, point 2 is located on the diagram by drawing a vertical line from state point 1 to the intersection with the horizontal pressure line that represents the saturation pressure in evaporator. When the liquid refrigerant passes through the metering device orifice, it expands as it enters the lower pressure evaporator. During this expansion process, the temperature of the liquid is reduced from its condensing temperature to the evaporating temperature. This adiabatic process occurs by flashing a portion of the warmer liquid refrigerant into its vapor phase. As a consequence of partial vaporization of some of the liquid refrigerant during process 1-2, the refrigerant at point 2 is a liquid–vapor mixture. Therefore, state 2 lies within the mixture region of the ph diagram.

3. The process 2-3 represents the vaporization of the refrigerant in the evaporator. At point 3, the refrigerant is complet vaporized. Since vaporization takes place at a constant temperature and pressure, process 2-3 is an isothermal and isobaric process. Therefore, point 3 is located on the ph diagram by following the line of constant pressure or constant temperature from point 4 to the point where they intersect the saturated vapor line. The refrigerant at state 3 exists as a saturated vapor at the evaporator’s vaporizing temperature and pressure.
4. In the basic saturated cycle, the refrigerant does not undergo a change in its condition while flowing through the suction line. Therefore, process 3-4 takes place within the compressor, as the pressure of vapor is increased from the saturation pressure in evaporator to saturation pressure within the condenser. In a basic saturated cycle, the compression cycle is assumed to be isentropic. Isentropic compression is a special type of adiabatic process that takes place without any losses due to friction. Since this process is isentropic, the entropy of the refrigerant at point 4 is the same as at point 3. State point 4 can be located on the ph diagram by following the line of constant pressure corresponding with the condensing pressure.

5. As a consequence of absorbing the heat of compression, the vapor discharged from the compressor is greater than the saturation temperature corresponding to its current pressure. Before the discharged vapor can condense, the additional superheat must be removed so that the temperature of the vapor can be lowered from discharge temperature at state point 4 to its saturation temperature at state point 5. Process 4-5 represents this cooling process. Process 4-5 occurs to some extent as the refrigerant flows through the discharge line. The remainder of the heat is rejected in the upper part of the condenser. During process 4-5, the pressure of the vapor remains constant. Therefore, point 5 is drawn on the ph diagram by following a line of constant pressure from point 4 to the intersection with the saturated vapor curve. At point 5, the refrigerant is once again a saturated vapor at the condensing temperature and pressure. The quantity of sensible heat removed per unit mass of vapor as it cools from the discharge temperature to the condensing temperature is the difference between the enthalpy of the refrigerant at point 4 and point 5.

6. Process 5-1 represents the phase change of the saturated vapor into a saturated liquid inside the condenser. Since condensation takes place at a constant temperature and pressure, process 5-1 follows along the horizontal line of constant pressure and temperature between points 5-1. Since both process 4-5 and 5-1 occur inside the condenser, the total amount of heat rejected by the refrigerant to the
condensing medium is the sum of the heat quantities rejected during process 4-5 and 5-1.

In the theoretical saturated cycle, the suction vapor is assumed to reach the suction inlet of compressor as a saturated vapor. This rarely occurs in real systems because the cold, saturated vapor inside the evaporator continues to absorb heat from the refrigerated space. Additional heat can be absorbed by the vapor as it travels through the suction line. Consequently, the vapor is superheated before it reaches the compressor. As a result, the real refrigeration cycles are less efficient than theoretical refrigerant cycles. Fig. 7 is the ph chart of vapor compression refrigeration cycle considering superheat and subcool.
2.2 Conventional MERS

The normal vapor compression refrigeration cycle can only provide one target temperature at a time. One way to meet the different cooling requirements is to use a multi-evaporator refrigeration systems (MERS) which consists of a single refrigeration system with one compressor and three-evaporators\[12\]. The schematic of conventional MERS is shown in Fig. 8 and its corresponding operating cycle on $P-h$ diagram is as shown in Fig. 9. The system consists of a single compressor and a single condenser but three evaporators working at evaporating temperature $-30^\circ\text{C}; -5^\circ\text{C}$ and $+7^\circ\text{C}$, respectively. To reduce the pressure of the higher temperature evaporators (EVAP1 and EVAP2) to the compressor suction pressure which is the pressure of the lowest temperature evaporator (EVAP3), pressure regulating valves (PRV1 and PRV2) are used to maintain the required pressure in high temperature evaporators.

As illustrated in Fig. 9, the working process of the MERS can be described as follows:

1. The refrigerant enters the compressor at low pressure $P_L$ at state (1) and is compressed isentropically to the high pressure $P_H$ at state (2).

2. The fluid enters the condenser where it condenses to state (3) by rejecting heat to the surroundings.

3. The refrigerant is divided into three flows, at states (4), (5) and (6), and enters the EVAP1 (state 7), EVAP2 (state 8) and EVAP3 (state 9) after pressure reductions by EV1, EV2, and EV3, respectively.

4. Through the evaporators, the refrigerant changes to the corresponding superheated states (10), (11) and (14).

5. The flow at state (10) was throttled to state (12); while the flow at state (11) was throttled to state (13).
6. The flows of states (12), (13), (14) mix up into flow state (1) and return to compressor, finishing the whole refrigeration cycle.

![Figure 8 Schematic of CMERS](image)

During the throttling process, the pressure is exhausted inside the pressure regulating valves, it is unrecoverable. Since refrigerant vapor at higher pressures are first throttled then compressed, and compressor inlet is in superheated region, the system is energy inefficient as the unrecoverable system pressure loss increases with increasing evaporating pressure differences.

In order to recover the pressure loss, the ejector is introduced in the multi-evaporator refrigeration system.

![Figure 9 Pressure-enthalpy chart of CMERS](image)
2.3 Ejector based Multi Evaporator Refrigeration System

The ejector is an old device and has been invented for a long time and applied to wide areas in process industry. The use of ejector in refrigeration system also has a long history, where the ejector based refrigeration cycle was developed, even before the appearance of vapor-compression refrigeration cycle, by Maurice Leblanc in 1910 [13], which utilized low grade thermal energy or waste heat instead of electricity to generate cooling. The main advantage of using an ejector as a pumping device is that it has fewer moving parts (no compressor). It is, therefore, very low in wear and significantly more durable.

An ejector consists of four main parts:

1. The nozzle;
2. The suction chamber
3. The throat
4. The diffuser

Its working principle together with the variations in the stream velocity and pressure as a function of location inside the ejector is illustrated in Fig. 10 and explained below:

The motive steam enters the ejector at point (p) with a subsonic velocity. As the stream flows in the converging part of the ejector, its pressure is reduced and its velocity increases. At the nozzle throat, the stream velocity reaches sonic speed, and its Mach number is equal to 1. In the diverging part of the nozzle, the cross section area increased, thus a decrease in the shock wave pressure and an increasing in its velocity to supersonic conditions take place. At point (2), the nozzle outlet plane, the motive steam pressure becomes lower than the entrained vapor pressure and reaches the highest velocity. The entrained vapor at point (e) enters the ejector, where its velocity increases and its pressure decreases to that of point (3). The motive steam and secondary streams may mix within the suction chamber and the diffuser’s converging section; it may also flow as two separate streams into the constant cross section area of the diffuser, and mix there. In both case, the mixture goes through a shock inside the constant cross section area of the diffuser. The shock will cause an increase in the mixture pressure and reduction to subsonic conditions of the mixture velocity at point (4). As the subsonic mixture emerges from the constant cross section area of the diffuser, another pressure increase process occurs in the diverging section of the diffuser, thus part of the kinetic energy of the mixture is converted into pressure at point (c). Since \( P_c > P_2 \), the pressure loss through the ejector is partially recovered when the flow passes through the diffuser.
The application of ejector for refrigeration system can be traced back to early 1990’s for domestic refrigerators [14-16]. It was later reported that the prototype of a compression-injection hybrid refrigeration cycle system for household refrigerators with an energy consumption reduction of 7.75% has been achieved[17]. An energy efficient three-evaporator refrigeration system with two ejectors EJ1 and EJ2 was also proposed recently by L. Kairouani, et al.[18] where the schematic is as shown in Fig.11. The pressure-enthalpy chart for the ejector based multi evaporator cycle is shown in Fig. 12.
The working process of the ejector based MERS can be described as follows:

1. The refrigerant flows into the compressor at low pressure $P_L$ at state (1).
2. The compressor works isentropically. The flow from the outlet is at state (2) with high pressure $P_H$.
3. The fluid enters the condenser then it condenses to state (3) by releasing heat to the environment.
4. The condensate is divided into three flows, at states (4), (5) and (6), and enters the EVAP1 (state 7), EVAP2 (state 8) and EVAP3 (state 9) after pressure reductions by EV1, EV2, and EV3, respectively.
5. Through the evaporators, the refrigerant changes to the corresponding superheated states (10), (11) and (15).

6. The flow at state (10) enters into the nozzle of EJ1 and expands to a mixture at state (12). The saturated secondary vapor drawn into EJ1 is at pressure Pevap2 (state 11).

7. The two vapors mix in EJ1 to state (13), flows through EJ1 diffuser where its pressure recovers to state (14) and then enters into the primary nozzle of the EJ2, then suck vapor into the EJ2 from evaporator 3 and expands to state (16).

8. The two vapors mix to state (17) and leave EJ2 after a recovery of pressure in the diffuser part at state (1).

The comparison can easily be made between fig. 9 and fig. 12. In fig 9, the compressor inlet pressure (pressure at state 1) is equal to the evaporating pressure of EVAP 3 (pressure at point 9). While in fig 12, the compressor inlet pressure (pressure at point 1) is higher than the evaporating pressure of EVAP 3 (pressure at state 9) because of the ejector, in this case, if the compressor outlet pressure is fixed, then the compression ratio will be affected by the performance of the ejector.

In this ejector based multi evaporator refrigeration system, ejectors are used to reduce the compression ratio of the compressor so that energy saving can be achieved. Pressure recovery ratio (PRR) is defined as[20]:

$$PRR = \frac{p_b - p_s}{p_s} \times 100\%$$

where, $p_b$ is the back pressure and $p_s$ is the secondary flow inlet pressure in ejector, to evaluate the performance of compression energy saving.

Through energy analysis, it is revealed that the system COP can be improved by about 8% and 20%, for two temperatures ejector based MERS and three temperatures ejector based MERS, respectively, as compared to the conventional MERSs[18].

2.4 Novel Ejector based Multi Evaporator Refrigeration System

Considering that the refrigeration and freezing loads are relatively stable, and the air-conditioning load can change in a wide range, we propose a new ejector based MERS configuration to accommodate these facts. The schematic of the new ejector based MERS is as shown in Fig. 13. The main differences of the new ejector based MERS compared with the existing EMERS are

1) When the air-conditioning is running, the primary and secondary inlets of EJ1 are from EVAP2 (refrigeration) and EVAP3 (freeze), respectively, and the output of EJ1 is used as the secondary inlet for EJ2;

2) When the air-conditioning is turned off, the solenoid valve is switched fully open so that the pressure drop in EJ2 is avoided.
The pressure-enthalpy chart using R134a as refrigerant with air-conditioning running is shown in Fig. 14,

In case of air-conditioning system is off, i.e., only EJ1 works, then the pressure-enthalpy chart is shown in Fig. 15.
The main drawback of this EMERS configuration is that it can only work well at the on-design condition. If the air-conditioning load (EVAP1) is reduced to less than 80% of rated value, the primary mass flow rate through the nozzle of EJ1 and EJ2 will be reduced and the flow passing through EVAP2 and EVAP3 will also be reduced. Consequently, both ejectors will not function properly and even more energy will be consumed as the results.

In order to study the power saving characteristics of the prepared ejector based multi evaporator refrigeration systems, as well as to solve the problems like how to expand the on-design working condition range; how to effectively allocate the cooling capacity; how to control these three highly coupled evaporators, etc, it is essential to design and build an experimental testing platform to support our research.

Because this is a completely new area of study, everything in this system is designed and built from scratch, the design of this system is described in the next chapter.
Chapter 3 Experimental Platform Setup

3.1 System Architecture

In order to provide valid experiment results for our novel ejector based multi evaporator refrigeration system study, the experimental platform must have the following features:

1. The system must be able to either run as a conventional MERS or run as a novel ejector based MERS. To conduct the comparison study between conventional and ejector based MERS, it is important that we can obtain the runtime data from both systems. So the experiment platform must provide two kinds of operation mode.

2. All the controllable parameters in the system must be able to be set and modified accurately and efficiently. The comparison study will cover a wide working point range, and we will use different parameter combination to set the system into different working point. It is essential that we keep both systems at the same working point, only in this case, the comparison will make sense.

3. The system must be stable and run continuously without interruption.

4. The system can perform good data saving and report generation.

5. The system should also have a good Human Machine Interface (HMI) which will clearly display all the parameters of interest. With this function, the researchers will be able to know the working state of the system at any time.

To fulfill these requirements, we propose the following system architecture:

1. Two stand-alone system architecture

   For this kind of architecture, we will build two separated system as shown in fig 8 and fig 13, where each system has its own compressor, condenser, EEV and evaporator. Each system should have its own control system, thus these two system can run at the same time. The advantage of this architecture is that they can run a perfect comparison experiment between conventional MERS and ejector based MERS, and they can be set to run at the same cooling capacity (same working states) when they are both running. In this way the researchers can get the best comparison results from these two systems at real time. Also, because each system has its own control system, it is relatively easy for the operators to set the parameters and control these two systems. The disadvantage is that the cost will be very high because of two sets of compressor, condenser and evaporators, and also the space required for these two systems will be doubled, so it is not cost and space efficient.

2. Two systems running simultaneous while using the same compressor and condenser

   For this kind of architecture, we will combine conventional MERS in Fig 8 and ejector based MERS in fig 13 together and build a new system as shown in fig.16.
This architecture is not a simple combination of these two systems, noting that the compressor and condenser is shared between these two sub-systems. All the refrigerant flow is provided by this single compressor, and the flow in each sub branch will be controlled by the expansion valves (EV_{C1-3} and EV_{E1-3}). An essential problem is the pressure balance between these two different sub systems, as is shown in Fig 9 and Fig 14, the compressor inlet pressure for ejector based MERS is higher than that in the conventional MERS, which means that in this architecture, the output refrigerant pressures is different between the two sub systems. Thus in order to keep both sub systems work properly, another pressure regulation valve must be installed between the outlet of the ejector 2 and the inlet of the compressor. Only in this way, the two-sub systems are able to function as required.

The advantage of this system architecture is that the hardware cost will be cheaper than the first architecture, while it can still perform a real time comparison experiment between conventional MERS and ejector based MERS. Both sub systems can run simultaneously, and the working state will be controlled by the system software. The disadvantage is that the control is very difficult. In order to fine tune the 6 evaporators, all the 6 expansion valves and the compressor must be controlled at the same time. Any change on one of these control factors will affect the other working state, so this control system contains more than 10 coupled control parameters. Although these two systems can run simultaneously, it is very difficult to get two sub-systems to run properly as the researchers’ request. The second disadvantage is that the power consumption of the compressor is not easy to obtain for each system, because the two systems are using the same compressor to provide the compressed refrigerant, we can only use the inlet pressure, outlet pressure and mass flow to approximately calculate the power consumption of each sub system. In the first architecture, we can easily get the power consumption by reading the power meter on the compressor.
3. Two systems running in turns while using the same compressor and condenser

In order to solve the problems in the first two system architectures, we proposed another system architecture, it also combines the conventional MERS in fig 8 with the ejector based MERS in fig 13, but the two sub-systems are not working together. They do not work simultaneously, instead they operate in turns. The system architecture is shown in fig.17.

This system architecture uses a set of hand valves to make the whole system switchable between conventional MERS and ejector based MERS, as in fig. 17. When the hand valves 1, 2, 4 are open and hand valves 3, 5, 6, 7 are closed, the system diagram is just the same as that in fig. 8, i.e. it is a conventional MERS, and the ejector part is completely closed. If we close hand valves 1, 2, 4 and open hand valves 3, 5, 6, 7, the system diagram will be the same as that in fig 13, so the system will be working as an ejector based MERS. We can easily switch between these two kinds of system setup by simple operation of the hand valves.

![Figure 17 Switchable multi evaporator refrigeration system](image)

The advantage of this system architecture is that it has the lowest hardware cost as compared with the first and second architectures, although it cannot provide the running data for both systems at the same time. These two systems can operate in
turns. If the system is controlled and debugged, it is not hard for the researchers to keep these experiments under similar conditions, thus we can still have meaningful comparison data using this system setup. Another advantage for this system architecture is that the control system is easier to design and implement compared with the second architecture. Besides the operation of hand valves, which can be done manually, the other system control functionality is essentially the same as that of the first architecture. This will simplify the control system dramatically. For this system architecture, the most considered disadvantage is that it cannot be run simultaneously. Sometimes, when we performing system comparison study, it is essential that the systems under comparison are run together, this is so that we can change the working state at the same time point. In this way, the researchers can monitor and control all the systems on the fly and get the most useful data. But in this system architecture, if one system is running, it is difficult to switch to the other system, then switch back and continue running. But if we have a complete experiment plan for the comparison study, we can finish all the tests on one system, then switch to the other and repeat the test. This can be a little time consuming, but the results will still be acceptable.

Considering these three system architectures, comparing their advantages and disadvantages, the third system architecture is chosen as our experimental platform architecture. This saves us a lot of components cost but still gives us good experiment data.

3.2 Main System Components

3.2.1 Compressor

The compressor is one of the most important components in this refrigeration system.

The refrigeration compressor is a mechanical device that increases the pressure of a refrigerant by reducing its volume, and then transport the fluid into the system pipe. For most refrigerants, at the inlet of the compressor, the state is low temperature, low pressure gas. And at the outlet of the compressor, the refrigerant is compressed into high pressure, high temperature gas. For our experimental system, we have some general requirements for the compressor.

1. The power consumption of the chosen compressor should be big enough. This is because we have three evaporators in our system, one working at +7°C evaporating temperature, while the other two are working at -5°C and -30°C evaporating temperature, respectively. For the general refrigeration systems, the lower the evaporating temperature, the lower is the COP. Thus in order to achieve the same cooling capacity, the evaporator with lower temperature set will cost more power. In our system, we will have two evaporators working at relatively low evaporating temperature, so we will need a powerful compressor to guarantee that three evaporators can work normally.

2. The maximum outlet pressure should be as high as 11bar, and the compression ratio of outlet pressure over inlet pressure will be around 10. Because we have a low temperature evaporator with the evaporating temperature at -30°C, the evaporating
pressure for refrigerant (R134a) is 1 bar, the outlet pressure of the compressor will be up to, depending on the working state, 10 bar.

3. The chosen compressor should be able to work under low temperature. The lowest temperature in this system is -30°C. Although the inlet of the compressor will not be that low, the compressor should be able to work at -30°C.

4. The frequency of this compressor should be adjustable according to our working state. In our experiment, we will change the cooling capacity of the system to different values to verify the system under different specifications. Thus the output power of the compressor must be able to adjust to our requirements. The direct method of changing the compressor power is to change the frequency. That means we will need an open, or semi-hermetic compressor.

Considering the above compressor requirements, we have chosen the product from Bitzer Company. The model is Bitzer 4cc-9.2y. This is a Semi-hermetic Reciprocating Compressor for R134a refrigerant, the lowest evaporating temperature under standard working mode is -30°C, the maximum high pressure is 28bar, the maximum input power is 11.8kw, and the working frequency can vary from 30Hz to 60Hz. The specification of this model can satisfy our demands.

3.2.2 Condenser

In heat transfer systems, a condenser is a device or unit used to condense a substance from its gaseous to its liquid state, by cooling it. In so doing, the latent heat is given up by the substance, and will transfer to the condenser coolant. Condensers are typically heat exchangers which have various designs. In typical refrigeration systems, a condenser is to get rid of heat extracted from the interior of the unit to the outside air. Condensers are used in air conditioning, industrial chemical processes such as distillation, steam power plants and other heat-exchange systems. Use of cooling water or surrounding air as the coolant is common in many condensers.

In our experiment, the condensing coolant is air at room temperature, which is around 25°C in our Lab. We used a finned condenser with 20m² of effective heat exchange area. The compressed high temperature (above 50°C), high pressure (above 9 bar) refrigerant gas enters from the inlet, goes through heat transfer process, becomes high pressure, medium temperature (around 28°C) refrigerant liquid, exits from outlet and proceeds into next cycle step.

3.2.3 Expansion Valve

The expansion valve removes pressure from the liquid refrigerant to allow expansion or change of state from a liquid to a vapor in the evaporator. In order for the higher temperature fluid to cool, the flow must be limited into the evaporator to keep the pressure low and allow expansion back into the gas phase.

The high-pressure liquid refrigerant entering the expansion valve is quite warm. This may be verified by feeling the liquid line at its connection to the expansion valve. The liquid refrigerant leaving the expansion valve is quite cold. The orifice within the valve does not remove heat, but only reduces pressure. Heat molecules contained in the liquid refrigerant are thus allowed to spread as the refrigerant moves out of the orifice. Under a greatly
reduced pressure the liquid refrigerant is at its coldest as it leaves the expansion valve and enters the evaporator.

Pressures at the inlet and outlet of the expansion valve will closely approximate gauge pressures at the inlet and outlet of the compressor in most systems. The similarity of pressures is caused by the closeness of the components to each other. The slight variation in pressure readings of a very few pounds is due to resistance, causing a pressure drop in the lines and coils of the evaporator and condenser.

There are two possible expansion valve types we can choose from:

1. **Thermal expansion valve**
   
   A thermal expansion valve (often abbreviated as TEV, TXV, or TX valve) is a component in refrigeration systems that controls the amount of refrigerant flow into the evaporator thereby controlling the superheating at the outlet of the evaporator. Thermal expansion valves are often referred to generically as "metering devices". The thermal expansion valve has sensing bulbs connected to the suction line of the refrigerant piping. The sensing bulbs give temperature readings to the TXV to adjust flow of refrigerant.

   The conventional thermal expansion valve is controlled by springs, bellows, and push rods. The spring force is a closing force on the thermal expansion valve. The evaporator pressure, which acts under the thermostatic element's diaphragm, is also a closing force. An opening force is the remote bulb force, which acts on top of the thermostatic element's diaphragm. There is also a liquid force from the liquid line, which has a tendency to open the valve. Working together, these forces maintain a constant evaporator superheat in a refrigeration system.

   Generally speaking, the thermal expansion valve dose not need real time control, it will automatically sensing the refrigerant temperature and adjust the opening of the valve. There are no electronic devices associated with a conventional thermal expansion valve.

2. **Electronic Expansion Valve**

   The electronic expansion valve (often abbreviated as EEV) controls the flow of refrigerant entering a direct expansion evaporator. It does this in response to signals sent to them by an electronic controller. A small motor is used to open and close the valve port. The motor is called a stepper motor. Stepper motors do not rotate continuously. They are controlled by an electronic controller and rotate a fraction of a revolution for each signal sent to them by the electronic controller. The stepper motor is driven by a gear train, which positions a pin in a port in which refrigerant flows.

   Stepper motors can run at 200 steps per second and can return to their exact position very quickly. The controller remembers the number of step signals sent by the controller. This makes it possible for the controller to return the valve to
any previous position at any time. This gives the valve very accurate control of refrigerant that flows through it.

In our experimental platform, the most important thing is to keep two systems running under the same cooling capacity. For each evaporator, it is essential that each evaporator’s expansion valve can be adjusted freely to give us the best flexibility. If we are using the thermal expansion valve, what we can control is only the evaporating temperature, we cannot directly control the opening of the valve. This will not be helpful in our case, so we decided to use the electronic expansion valve as our expansion device.

### 3.2.4 Evaporator

An evaporator is used in the refrigeration system to allow a compressed cooling refrigerant to evaporate from liquid to gas while absorbing heat in the process.

Typically, there are coils or tubes in the evaporator. A fan (blender) circulates warm air (water) in the enclosed space across the coils or tubes carrying the cold refrigerant liquid and vapor mixture from the expansion valve. That warm air (water) evaporates the liquid part of the cold refrigerant mixture. At the same time, the circulating air (water) is cooled and thus lowers the temperature of the enclosed space to the desired temperature. The evaporator is where the circulating refrigerant absorbs and removes heat which is subsequently rejected in the condenser and transferred elsewhere.

In our experimental platform, we have three different target temperatures to achieve, thus we will need three different testing environments to run the experiments. The best solution is to design a customized testing chamber which contains a built-in evaporator while can also act as the chilled environment.

The functional diagram of the testing chamber is as shown in fig. 18.
The three testing chambers are designed especially for the system performance test. It has four main components:

1. A liquid-cooled heat exchanger as evaporator, which is the copper cube part in the fig. 18. The whole chamber will be the chilled space. The space in the testing chamber is relatively small compared to a building room. If we continue to use air as the heat transfer medium, the cooling capacity will be very small. In order to increase the potential cooling capacity, we will use liquid with higher specific heat as the heat transfer medium. Because we have three different evaporating temperatures, the heat transfer medium should be different as well.

For the air-conditioner (room temperature, testing chamber1), the evaporating temperature is 7℃, the water can be used directly as the heat transfer medium. For the food storage space (testing chamber2) and freezing space (testing chamber3), the evaporating temperatures are -5℃ and -30℃, respectively, water cannot be used anymore because the freezing point of pure water is 0℃, it will freeze into ice when temperature goes under 0℃.

The ethylene glycol solution was chosen to replace water in testing chamber 2 and 3. Different ethylene glycol solution has different freezing point, to choose a proper solution, one can refer to table 1.
Table 1 Freezing points of ethylene glycol based water solutions

<table>
<thead>
<tr>
<th>Ethylene Glycol Solution Freezing Point (%) by volume</th>
<th>0</th>
<th>10</th>
<th>20</th>
<th>30</th>
<th>40</th>
<th>50</th>
<th>60</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature (°F)</td>
<td>32</td>
<td>25.9</td>
<td>17.8</td>
<td>7.3</td>
<td>-10.3</td>
<td>-34.2</td>
<td>-63</td>
</tr>
<tr>
<td>Temperature (°C)</td>
<td>0</td>
<td>-3.4</td>
<td>-7.9</td>
<td>-13.7</td>
<td>-23.5</td>
<td>-36.8</td>
<td>-52.8</td>
</tr>
</tbody>
</table>

Another factor we should consider for ethylene glycol solution is that the dynamic viscosity will also increase when volume concentration increases, as is shown in table 2. If the viscosity is too big, the blender will be hard to operate properly, and it may cause system failure. In order to choose the best solution for the test, both freezing point and dynamic viscosity should be taken under consideration. The solution in table 1 with the closest freezing point compared with designed evaporating temperature will be the solution that can satisfy the temperature requirements while still has the smallest dynamic viscosity.

Table 2 Dynamic viscosity of ethylene glycol based water solutions

| Dynamic Viscosity –(centiPoise) |
|-------------------------------|------------------|------------------|------------------|------------------|------------------|------------------|------------------|------------------|
| Temperature                   | Ethylene Glycol Solution(%) by volume | 25   | 30   | 40   | 50   | 60   | 65   | 100  |
| (°C)                          | (-17.8)          | 1)   | 1)   | 15   | 22   | 35   | 45   | 310  |
|                               | 4.4              | 3    | 3.5  | 4.8  | 6.5  | 9    | 10.2 | 48   |
|                               | 26.7             | 1.5  | 1.7  | 2.2  | 2.8  | 3.8  | 4.5  | 14   |

For food storage testing chamber (chamber 2), we used ethylene glycol solution with a volume fraction of 20%, and its freezing point is -8°C. In the freezing testing chamber (chamber 3), we will use ethylene glycol solution with a volume fraction of 50%, and its freezing point is -37°C.
2. Each testing chamber is equipped with a blender driven by electrical motor. It can help increase the heat transfer rate in the testing chambers.

3. A controllable heater is also installed in the testing chamber. The heater operates as a controllable cooling load, dynamically generate heat to balance the current cooling capacity. It can keep the temperature of the heat transfer medium at a set value. When the temperature is stable, we can read the power consumption of the heater, which can be taken as the actual cooling capacity of the refrigeration system.

4. Two PT 100 temperature sensors are also mounted on the testing chamber. The temperature in the testing chamber is essential for the system control as well as data analysis.

The whole testing chamber will be thermal isolated from the room environment using insulation materials, as is shown in fig 19.

### 3.2.5 Variable area ratio ejector

Ejector is another key component in this experimental platform for the ejector based MERS as is shown in Fig. 13. Ejector will be designed according to the working condition, and the working condition will not vary too far from the designed working point. However, conventional fixed nozzle ejector cannot maintain the performance once the primary load changes. In a real process control system, varying load may be drastic. It means that as the loads vary, conventional ejector may be inefficient in the system. To overcome this
drawback, a variable area ratio ejector is introduced into the experimental platform. It will work as EJ2 in Fig. 13. So when air-conditioning load changes, the EJ2 can also be changed to adjust the real time system cooling capacity.

The configuration of the variable area ratio ejector is shown in Fig. 20. Like a typical ejector, this new ejector consists of four principal parts, which are the primary nozzle, the suction chamber, the ejector throat and the diffuser. But a spindle was added as a new feature. The function of the spindle is to provide a fine tuning for the ejector operation. The primary flow rate can be adjusted using the spindle in order to provide more flexible operation [20]. The spindle changes the primary nozzle throat area and therefore changes the area ratio (defined as throat section area to primary nozzle throat area) of the ejector. As the spindle tip travels forward, the primary nozzle throat area decreases, and consequently the mass flow rate decreases. Therefore, the primary flow rate can be adjusted in order to provide the largest entrainment ratio and COP (the entrainment ratio and COP are effected by the area ratio) possible as well as a more flexible operation.

In the ejector based MERS case, the variable area ratio ejector will be used to maintain the evaporating pressure in EVAP1 as the air-conditioning load changes. If the air-conditioning load increases, the nozzle area should also be increased while the air-conditioning load decreases. The nozzle area should be decreased too to keep the evaporating pressure constant.

We developed two methods to implement this variable area ratio ejector. One is to retrofit an electrical valve, another one is to design a new welded metal bellow based variable area ratio ejector.
An electrical valve retrofitted variable area ratio ejector is as shown in Fig. 21:

![Figure 21 Variable area ratio ejector 1](image)

1. Ejector Body
2. Electrical Valve Motor
3. Spindle

Part 2 is the motor of an electrical valve, we replaced the valve body with a customized ejector at part 1. The spindle is mounted at part 3. The purpose of this design is to use the air tight structure of the electrical valve, which is from the flange to the mounted spindle. The air tightness is guaranteed by the filler inside the right side flange. And because of the filler, to move the spindle we need a relative big power to push in and pull out. That’s why this structure requires a high power rated motor.

Now we have this kind ejector installed in our experimental platform, it can efficiently change the working state of the ejector.

Another ejector is welded metal bellow based variable area ratio ejector, as shown in Fig. 22.

All the components of bellow based adjustable ejector are mounted on the Mounting base (9). The Front Mounting Bracket (2), Rear Mounting Bracket (5) and Sliding Chute (8) are fixed on the Mounting base.

The Ejector (1) should have an open back end with flange, so it can be mounted on the Front Mounting Bracket. The Bellow will have flanges on both ends, one side is mounted on the Front Mounting Bracket; another side is mounted on the Sliding base. Thus the Sliding base, bellow and the ejector should make up a common gas tightness space.

The Spindle (10) is fixed on the Sliding base, and it must be concentric with the nozzle of the ejector. The Sliding base is placed on the Sliding Chute (8) in order to hold its position and make sure the spindle is concentric with the nozzle.

The Worm (6) is mounted through the Rear Mounting Bracket (5) and fixed on the Sliding base. The Motor and Gear (7) will be installed to drive the worm.
For both variable area ratio ejector designs, by adding moving parts to the ejectors, there are two problems to solve. 1) How to maintain the gas tightness of the ejector, 2) the ejector should be easy to control and adjust. The electrical valve retrofit method maintains the gas tightness of the moving spindle by adding filler to possible gas leakage point. But the moving of the spindle will require much power. So we tried to use bellows as is shown in Fig 2. Using this method, we can move the spindle with less power while still maintain the gas tightness of the ejector.

### 3.3 Measurement and Control System

Besides the main components we have discussed above, measurement and control systems are key elements in this experimental platform. We need to keep an eye on all the interested measurement points, including pressure, temperature, and flow rate at different points of the system, while controlling the compressor, condenser, electronic expansion valve and so on. Good measurement and control systems are required to achieve all these targets.

#### 3.3.1 Development platform for Measurement and Control system

We use the CompactDAQ and LabVIEW (laboratory virtual instrumentation engineering workbench) from National Instruments [www.ni.com] as our development platform. CompactDAQ is a portable, rugged data acquisition platform that integrates connectivity and signal conditioning into modular I/O for directly interfacing to sensor or signal. We
use CompactDAQ with LabVIEW software to customize data acquire, analyze, present, and manage the measurement data.

The CompactDAQ system consist of two parts, the chassis and the function modules. The chassis we used in the experimental platform is NI cDAQ-9178 CompactDAQ controller and USB chassis, as is shown in Fig. 23.

The NI cDAQ-9178 is an 8-slot NI CompactDAQ USB chassis designed for small, portable, mixed-measurement test systems. Combine the cDAQ-9178 with up to eight NI C Series I/O modules for a custom analog input, analog output, digital I/O, and counter/timer measurement system. CompactDAQ chassis and controllers control the timing, synchronization, and data transfer between C Series I/O modules and a host computer. The USB chassis provides the plug-and-play simplicity to sensor and electrical measurements.

In order to take different measurement, the data acquisition modules are also required. NI cDAQ-9178 can provide slots for up to 8 NI C series DAQ modules. C Series modules provide high-accuracy measurements for advanced data acquisition applications. Each module contains measurement-specific signal conditioning to connect to an array of sensors and signals, and support for wide temperature ranges to meet a variety of application and environmental needs, all in a single rugged package, as is shown in Fig. 24.
The module types we used in the experimental system are NI 9205, 8 channel voltage input module; NI 9219, 4 channel universal input module and NI 9401, Bidirectional digital Input/Output module.

NI 9205 is the simplest voltage measurement module. It can measure voltage range from -10V to +10V. In our application, we use two NI 9205 to measure the outputs from RTD transducer, pressure sensor and flow meter.

The NI 9219 is a 4-channel universal C Series module designed for multipurpose testing. With NI 9219, we can measure several signals from sensors such as strain gages, resistance temperature detectors (RTDs), thermocouples, and other powered sensors. The channels are individually selectable, so we can perform a different measurement type on each of the four channels. In this experimental platform, the NI 9219 is used to measure the key temperature RTD data of the system. There are the temperature before evaporator, the evaporating temperature and the temperature after evaporator.

Besides analog data acquisition, we also need digital signal output for this application. We have three electronic expansion valves in the system, each one will require at least three digital signals to operate. Thus the NI 9401 is used to fulfill the digital DAQ requirements.

### 3.3.2 HMI (Human Machine Interface) software development

We use the LabVIEW to develop our HMI software.

LabVIEW is a development environment designed specifically that can help us quickly develop our testing software. With a graphical programming syntax that makes it simple to visualize, create, and code engineering systems, LabVIEW dramatically reduce the time to develop and retrofit system software.

Although the programming method of LabVIEW is quite different from other programming language such as C, C++, etc. The ideas of programming is all the same. For our program, we used the programming technique named state machine to provide proper organization of all these codes.

The state machine is one of the fundamental architectures frequently used in NI LabVIEW to help us build applications quickly. We can use state machines in applications where distinguishable states exist. Each state can lead to one or multiple states and can end the process flow. A state machine relies on user input or in-state calculation to determine which state to go to next. Many applications require an “initialize” state followed by a default
state, where we can perform many different actions. These actions depend on previous and current inputs as well as states.

In our application, we defined 10 states and they are:

- **Initial:** Initiate the system, set all the parameter to default value (if any), and prepare the system to accept command from operator
- **Idle:** No command received, No task to finish
- **HMI Input Check:** Check if there is any input from HMI
- **Sampling Time Check:** Check if it is time to perform a data acquisition operation
- **Data Acquisition:** Read data from all the sensors and transducers
- **Data Processing:** Process the raw data from DAQ system and calculate the result
- **Data Recording:** Record the data according to the timestamp
- **Command Output:** Output data like compressor frequency and EEV open/close signal.

Figure 25 State diagram of experimental platform software
Error Check: Check if there is any error message from the system
End: shutdown the system.

The state diagram is as shown in Fig. 25, and the sample front panel is as shown in Fig. 26. Once we have this state diagram, we can program the LabVIEW block diagram according to the state transfer, which makes the programming easier and clearer.

Figure 26 Front Panel of experimental platform software

The details of the whole system programming will not be discussed here, and only some special data processing will be discussed.

1. Compressor frequency control
   As it is discussed previously, the compressor frequency must be changeable according to set working state. The Working frequency for the compressor could vary from 30Hz to 60Hz. In order to change the frequency of the compressor, an inverter must be used to connect to the power input of the compressor. The inverter used in the experiment platform is VLT Micro Drive FC 51 with rated power at 7.5kW. This inverter can be controlled by the front panel on it. The application also requires the control of the inverter via a program, thus a voltage output module is introduced to output a 0-10V DC voltage to the inverter DC input terminal. Then we can easily control the output DC voltage value to control the compressor frequency. One important thing is to make sure that the program has the input value limitation of 30 to 60Hz, because any frequency under 30Hz or over 60Hz will damage the compressor.
2. Testing chamber temperature control
The testing chamber has an integrated heater to balance the cooling capacity of the refrigeration system. It is strange that for a refrigeration system, we will control the heater to balance the temperature. The reason is that to control the cooling capacity in each evaporator is very difficult. As the three evaporators has coupled relationships, any change made to any evaporator will affect the other two. So direct control of the cooling capacity is hard to achieve. But if we assume the inlet and outlet refrigerants are at saturation state, and if we can keep the inlet temperature, pressure, outlet temperature, pressure, and flow rate in the two systems at the same or very close value, the same or nearly the same cooling capacity can be achieved, and the comparison study will be reasonable. In this case, it is essential that we are able to keep the testing chamber temperature stable.
The heater is powered by SCR based heater controller, the controller can accept PWM input, and the output power is adjusted according to the duty ratio of the input PWM. We use the classical PID control method to control the actual temperature of each testing chamber. Use the setting temperature, current chamber temperature as the input variables, calculate the output and obtain the required PWM duty ratio. Then use the digital output module on the programmable controller to generate the PWM. So we can control the real time testing chamber temperature.

3. Cooling capacity calculation
There are two ways to calculate cooling capacity.
   a. When the system is running at steady state, the power of heater should be equal to the cooling capacity, because they are at thermal equilibrium state. So one way to get the cooling capacity is to calculate the heater’s power consumption. But this method will introduce other errors as energy loss, circuit loss, etc. And we will need new meters to measure the voltage and current of the heater.
   b. Assume the inlet and outlet refrigerants are at saturation state, then the cooling capacity can be approximately calculated using the following equation:
$$\left(h_{sat\_gas} - h_{sat\_liquid}\right) \times \left(flowrate_{liquid} \times density_{liquid}\right)$$
where $h_{sat\_gas}$ is the saturation Enthalpy of the evaporator outlet gas, $h_{sat\_liquid}$ is the saturation Enthalpy of the evaporator inlet liquid, $flowrate_{liquid}$ is the inlet flowrate of refrigerant by volume, $density_{liquid}$ is the density of the inlet refrigerant at the current pressure and temperature.
When the refrigerant is under saturation state at specified temperature and pressure, all the parameters above can be obtained from the physical characteristics table. Table 3 is part of R134a characteristics table.

Table 3 R134a saturation characteristics table (part)

<table>
<thead>
<tr>
<th>Sat Pressure</th>
<th>Sat Temp</th>
<th>Sat h_liquid</th>
<th>Sat h_gas</th>
<th>Sat density</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.420520521</td>
<td>-43.78638769</td>
<td>143.4048793</td>
<td>371.5767119</td>
<td>1428.604797</td>
</tr>
<tr>
<td>0.543643644</td>
<td>-38.94375628</td>
<td>149.4308671</td>
<td>374.675996</td>
<td>1414.615768</td>
</tr>
<tr>
<td>1.036136136</td>
<td>-25.63345745</td>
<td>166.3891745</td>
<td>383.0072564</td>
<td>1375.300372</td>
</tr>
<tr>
<td>1.4998999</td>
<td>-17.23486062</td>
<td>177.1946812</td>
<td>388.2590836</td>
<td>1349.728068</td>
</tr>
</tbody>
</table>

As is shown in table 3, the provided reference value is not a continuous curve, it is not possible to locate the exact value for actual working state, only most suitable ones can be find. So we will read the real time temperature of the inlet liquid and outlet gas, find the closest value in the table, put the value in the above equation and calculate the actual cooling capacity.

4. Data record

A good experiment system must have powerful data recording functions, so that it can continuously record the system running state even if there is no operator standby. The ejector based MERS experimental system has a complete data recording function, without the need for help from operator. It can automatically record the compressor power, compressor inlet pressure, outlet pressure, Opening of three electronic expansion valves, refrigerant flow rate, evaporator temperature, pressure, ejector pressure, etc. All the key parameter values will be recorded together with experimental data, data taken time, and calculated cooling capacity. So it is very easy for the researchers to just open a file and find the data they need, and do any post data analysis he wants.

The above are some details information about the system architecture, components, and control system platform and software functions of the designed experimental platform.

A schematic view and Photographs of the experimental setup are shown in Figs. 27 and 28, respectively.
Figure 27 Schematic of the experimental setup

Figure 28 Photographs: the experimental setup
Chapter 4 Experiment results

4.1 Conventional MERS cycle

In order to take the measurements of conventional PRV MERS, the hand valves V9, V10, V11 are switched off while V5, V6, V7 in Fig.10, are switched on; the system schematic is shown as in Fig.29:

Figure 29 Schematic of conventional MERS setup

The key parameters we need to measure are:

1. P6: Compressor inlet pressure
2. P1: Compressor outlet pressure
3. P3/P4/P5: Three evaporating pressures
4. F1/F2/F3: Three evaporator flow rates
5. Compressor frequency
6. Compressor power consumption

As mentioned earlier, the refrigeration and freezing loads are relatively stable, and the air-conditioning load can change in a wide range, we will change the air-conditioning loads to investigate the system performance. While keeping the evaporating pressure relatively constant, we will adjust the EEV, change the flow rate, and control the compressor and pressure regulating valves to achieve the following working condition:

Flow rate (L/H): EVAP1: 40/50/60/80/90, EVAP2: 30, EVAP3: 10
Evaporating Pressure (Bar): EVAP1: 4.2, EVAP2: 2.1, EVAP3: 1.7
Because the three evaporators are highly coupled, any change on one of the evaporators will affect the other two. The measuring data may be slightly different from the setting value, but it will not affect the analysis. The measurement data is listed in Table 4:

<table>
<thead>
<tr>
<th>Experiment No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>F1:(EVAP1 Flow)(L/H)</td>
<td>41</td>
<td>50.7</td>
<td>63.2</td>
<td>83.4</td>
<td>92</td>
</tr>
<tr>
<td>F2:(EVAP2 Flow)(L/H)</td>
<td>27.6</td>
<td>30.3</td>
<td>31.6</td>
<td>29.6</td>
<td>27.8</td>
</tr>
<tr>
<td>F3:(EVAP3 Flow)(L/H)</td>
<td>8.1</td>
<td>8.2</td>
<td>8.3</td>
<td>9.2</td>
<td>8.4</td>
</tr>
<tr>
<td>P3:(EVAP1 Pressure)(Bar)</td>
<td>4.13</td>
<td>4.23</td>
<td>4.15</td>
<td>4.18</td>
<td>4.19</td>
</tr>
<tr>
<td>P4:(EVAP2 Pressure)(Bar)</td>
<td>2.13</td>
<td>2.1</td>
<td>2.15</td>
<td>2.18</td>
<td>2.24</td>
</tr>
<tr>
<td>P5:(EVAP3 Pressure)(Bar)</td>
<td>1.5</td>
<td>1.58</td>
<td>1.62</td>
<td>1.86</td>
<td>1.93</td>
</tr>
<tr>
<td>P6:(COMP Inlet)(Bar)</td>
<td>1.14</td>
<td>1.2</td>
<td>1.24</td>
<td>1.4</td>
<td>1.46</td>
</tr>
<tr>
<td>Compressor Frequency(Hz)</td>
<td>45.3</td>
<td>48</td>
<td>49.5</td>
<td>51.1</td>
<td>51.1</td>
</tr>
<tr>
<td>Compressor Power(kW)</td>
<td>4.92</td>
<td>5.185</td>
<td>5.624</td>
<td>6.12</td>
<td>6.34</td>
</tr>
</tbody>
</table>

Using the measured data, we can calculate the cooling load and COP of conventional MERS as listed in Table 5:

<table>
<thead>
<tr>
<th>Experiment No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>cooling load</td>
<td>3.591</td>
<td>4.2</td>
<td>4.86</td>
<td>5.785</td>
<td>6.132</td>
</tr>
<tr>
<td>COP</td>
<td>0.73</td>
<td>0.81</td>
<td>0.86</td>
<td>0.94</td>
<td>0.96</td>
</tr>
</tbody>
</table>

![Figure 30 Effect of EVAP1 flow rate on COP in conventional MERS](image)

As shown in Fig.30, with the cooling loads of EVAP2 and EVAP3 relatively stable, when the flow rate of EVAP1 increases, the cooling load also increases, and the COP increases.
So the cooling load of EVAP1 will affect the whole system’s COP greatly. Generally, the air-conditioning’s COP is higher than refrigeration and freezing, it will have such effect on the system.

Our experiment is not only for COP investigation, but also for ejector based MERS power saving analysis.

4.2 Ejector base MERS cycle

In order to take the measurements of ejector based MERS, we will switch off the hand valves V5, V6, V7 while switch on V9, V10, V11 in Fig.10. The system schematic is shown in Fig.31:

![System Schematic](image)

Figure 31 Schematic of ejector based MERS setup

The key parameters we need to measure are:

1. P6: Compressor inlet pressure
2. P1: Compressor outlet pressure
3. P3/P4/P5: Three evaporating pressures
4. Back pressure of EJ1 and EJ2
5. F1/F2/F3: Three evaporator flow rates
6. Compressor frequency
7. Compressor power consumption

In order to compare with the conventional MERS, the working condition is also set as below:

- Flow rate (L/H): EVAP1: 40/50/60/80/90, EVAP2: 30, EVAP3: 10
- Evaporating Pressure (Bar): EVAP1: 4.2, EVAP2: 2.1, EVAP3: 1.7
For the pressure regulating valves are no longer used in the system, the controllable items are compressor frequency, 3 EEVs and the variable area ratio ejector. When adjusting EVAP1 flow rate, we must control the EEV, compressor and ejector together to get the target working condition.

We found that when the EVAP1 flow rate drops to 40L/H, the working condition cannot be maintained any more. This is because the EVAP1 flow provides the EJ1 primary flow, when the flow rate drops to 40L/H, it is too small to make the ejector working normally. If the EJ1 cannot work, the EJ2 cannot work either. So we did not manage to make the measurement when EVAP flow rate is 40L/H. The measuring data of other working condition is listed in Table.3:

<table>
<thead>
<tr>
<th>Experiment No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>F1:(EVAP1 Flow)(L/H)</td>
<td>51.2</td>
<td>63</td>
<td>83</td>
<td>92</td>
<td></td>
</tr>
<tr>
<td>F2:(EVAP2 Flow)(L/H)</td>
<td>27.6</td>
<td>31.5</td>
<td>29.5</td>
<td>28.9</td>
<td></td>
</tr>
<tr>
<td>F3:(EVAP3 Flow)(L/H)</td>
<td>8.5</td>
<td>8.2</td>
<td>9.54</td>
<td>8.5</td>
<td></td>
</tr>
<tr>
<td>P3:(EVAP1 Pressure)(Bar)</td>
<td>4.21</td>
<td>4.18</td>
<td>4.13</td>
<td>4.18</td>
<td></td>
</tr>
<tr>
<td>P4:(EVAP2 Pressure)(Bar)</td>
<td>2.07</td>
<td>2.12</td>
<td>2.17</td>
<td>2.23</td>
<td></td>
</tr>
<tr>
<td>EJ1 Outlet Pressure</td>
<td>1.9</td>
<td>1.87</td>
<td>1.98</td>
<td>1.98</td>
<td></td>
</tr>
<tr>
<td>P5:(EVAP3 Pressure)(Bar)</td>
<td>1.72</td>
<td>1.58</td>
<td>1.81</td>
<td>1.84</td>
<td></td>
</tr>
<tr>
<td>EJ2 Outlet Pressure</td>
<td>1.83</td>
<td>1.78</td>
<td>1.91</td>
<td>1.92</td>
<td></td>
</tr>
<tr>
<td>P6:(COMP Inlet)(Bar)</td>
<td>1.45</td>
<td>1.39</td>
<td>1.59</td>
<td>1.57</td>
<td></td>
</tr>
<tr>
<td>P1:(COMP Outlet)(Bar)</td>
<td>9.34</td>
<td>9.58</td>
<td>10.06</td>
<td>10.27</td>
<td></td>
</tr>
<tr>
<td>Compressor Freq(Hz)</td>
<td>43</td>
<td>43.7</td>
<td>44.6</td>
<td>47.4</td>
<td></td>
</tr>
<tr>
<td>Compressor Power(kW)</td>
<td>4.757</td>
<td>4.968</td>
<td>5.676</td>
<td>6.232</td>
<td></td>
</tr>
</tbody>
</table>

Using the measured data, we can calculate the pressure recovery ratio, cooling load and COP of conventional MERS as listed in Table.4:

<table>
<thead>
<tr>
<th>Experiment No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>EJ1 Pressure recovery ratio</td>
<td>3.83%</td>
<td>5.05%</td>
<td>3.66%</td>
<td>3.12%</td>
<td></td>
</tr>
<tr>
<td>EJ2 Pressure recovery ratio</td>
<td>6.40%</td>
<td>12.66%</td>
<td>5.52%</td>
<td>4.34%</td>
<td></td>
</tr>
<tr>
<td>cooling Load</td>
<td>4.12</td>
<td>4.84</td>
<td>5.784</td>
<td>6.164</td>
<td></td>
</tr>
<tr>
<td>COP</td>
<td>0.866</td>
<td>0.974</td>
<td>1.019</td>
<td>0.98</td>
<td></td>
</tr>
</tbody>
</table>
As shown in Fig. 32 we can see that the EJ1 and EJ2 have the highest pressure recovery ratio when EVAP1 flow rate is around 60L/H. At this time, the working condition of the system fits the ejector’s design working state very well. If we change the working condition of the system, the change in flow rate will affect the ejector’s working state, although we can control the variable area ratio ejector, it can only improve the performance in a certain degree. How to control the variable area ratio ejector is an important future work.

We can also find that the PRR of EJ2 is higher than the PRR of EJ1. This is because we made a lot of changes to our system, for ejectors, some working condition cannot be achieved, the set point shifting is much more serious on EJ1 than that on the EJ2. Thus the EJ2’s working condition is better than EJ1. So the next step we have to redesign the ejectors, so that the PRR can be improved again.

As shown in Fig. 33, generally, the COP of ejector based MERS is higher than the conventional MERS, the difference is: in conventional MERS, when the EVAP1 flow rate increases, the COP will also increase, but in the ejector based MERS, because the pressure
recovery rate may decrease while EVAP1 flow rate increases, so COP doesn’t have a monotone increasing relationship with the EVAP1 flow rate.

Table 8 Power saving of ejector based MERS

<table>
<thead>
<tr>
<th>Experiment No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>PRV cycle power consumption (kW)</td>
<td>5.185</td>
<td>5.624</td>
<td>6.12</td>
<td>6.34</td>
<td></td>
</tr>
<tr>
<td>Ejector cycle power consumption (kW)</td>
<td>4.757</td>
<td>4.968</td>
<td>5.676</td>
<td>6.232</td>
<td></td>
</tr>
<tr>
<td>Power saving</td>
<td>8.254%</td>
<td>11.664%</td>
<td>7.254%</td>
<td>1.703%</td>
<td></td>
</tr>
</tbody>
</table>

![Figure 34 Effect of EVAP1 flow rate on power saving rate](image)

Also as shown in Table 8 and Fig.34, where the PRR of ejectors have the maximum value, the power saving rate is the highest. In our setup, the max power saving is 11.664%. If we optimize the system parameter correctly, the power saving rate can still be improved. It is possible to achieve a 20% power saving.

4.3 Two Evaporator Testing

In practical test, if the experiments actually involves three evaporators, it will be very difficult to tune it and get the reasonable data. In order to simplify the test structure, and to reduce the coupling effect between evaporators, we decided to perform test on just two evaporators. Thus the comparison study would be easier and simpler.

We are using two evaporators for the experiment, evaporator 1 and evaporator 2, their evaporating pressures are at 2.4bar and 1.1bar, respectively. Also the system is equipped with variable area ratio ejector.

The energy consumption will increase while target temperature decrease. Therefore it is not correct to just simply add two evaporators’ cooling capacity together and do system
level analysis. The cooling capacity of each evaporator should be taken into consideration separately.

Because the PRV based conventional MERS cycle is relatively easier to control and adjust. During the comparative experiments, ejector based MERS cycle will first be tested. By adjusting the compressor frequency, opening of two EEVs and the position of spindle in the variable area-ratio ejector, we can tune this system to run at one fixed working state. Then we will switch the system into PRV based conventional MERS. Again by adjusting the compressor, two EEVs and the PRV to make the system running under the same working state.

In this experiment, by keeping the flow rates and evaporating pressures in both evaporators the same value when running under different system configuration, the cooling capacity can be controlled separately.

Table 9 shows the separated cooling capacity extracted from testing data for PRV MERS and ejector based MERS when $P_{EVAP1} = 2.4\text{bar}$ and $P_{EVAP2} = 1.1\text{bar}$.

<table>
<thead>
<tr>
<th>PRV Conventional MERS</th>
<th>Ejector Based MERS</th>
</tr>
</thead>
<tbody>
<tr>
<td>EVAP1 Flow(L/h)</td>
<td>EVAP1 Cooling Capacity (kW)</td>
</tr>
<tr>
<td>21.449</td>
<td>1.013</td>
</tr>
<tr>
<td>27.298</td>
<td>1.327</td>
</tr>
<tr>
<td>30.873</td>
<td>1.497</td>
</tr>
<tr>
<td>33.899</td>
<td>1.628</td>
</tr>
<tr>
<td>34.009</td>
<td>1.649</td>
</tr>
<tr>
<td>34.632</td>
<td>1.591</td>
</tr>
<tr>
<td>34.676</td>
<td>1.593</td>
</tr>
<tr>
<td>37.009</td>
<td>1.781</td>
</tr>
<tr>
<td>37.706</td>
<td>1.764</td>
</tr>
</tbody>
</table>
Figure 35 Cooling Capacity in evaporator 1 (Pressure=2.4bar)
Fig. 35 and Fig. 36 are the cooling capacity curves for evaporator 1 and evaporator 2, respectively. These two curves show that at all the corresponding working states, the cooling capacities of PRV based MERS group are very close to the cooling capacities in ejector based MERS group. In our experiment case, the working state refers to the refrigerant flow rate and evaporating pressure. Only if the working state and cooling capacity of these two system configurations are all the same, comparative data analysis will make sense. As mentioned before, the total cooling capacity or overall COP can only be considered as a related parameter; they do not have an obvious varying trend throughout the whole experiment, so they cannot be used as an analysis reference variable. Referring to table 9, the flow rate of evaporator 1/evaporator 2 is a monotonically increasing/decreasing variable, it can be used as a good reference variable when doing data analysis. In the following discussion, we will use flow rate of evaporator 1 or the evaporator 1 cooling capacity as the reference variable. All of our testing data are under the same evaporating pressure in evaporator 1 and evaporator 2. In order to keep these two pressures constant, we have to change the EEV,
compressor and ejector/PRV to adjust the flow rate in both loops and make sure the pressures do not change. Take ejector cycle for example, when it is necessary to decrease the flow rate of evaporator 1, first, EEV1 will be closed to a certain degree, because the flow rate of the evaporator decreases, the evaporating pressure will also drop. To keep the pressure unchanged, the spindle on the variable-area ratio ejector must be pushed in, decrease the nozzle area, thus it can throttle the flow to fit the current flow rate and make primary pressure increase to the setting value. And for the secondary flow, which is from evaporator 2, when primary flow decrease, the secondary pressure will decrease accordingly, to keep the pressure unchanged, the EEV2 has to be open to a certain degree, thus increase the flow rate and pressure in evaporator 2. This is why in table 9, the flow rate of evaporator 1 and evaporator 2 has a completely different varying trend. Fig. 37 shows the relationship between flow 1 and flow 2. Fig. 38 gives the relationship between evaporator 1 and evaporator 2’s cooling capacity. Because the evaporating pressures have been kept constant, the cooling capacity will be mainly affected by the flow rate, which is the reason why the curves in Fig. 35 and Fig. 36 are almost the same.

![Figure 37 Flow rate relationship between evaporator 1 and evaporator 2](image-url)
Ejector is implemented in the ejector based MERS to recover the pressure loss during the throttle process. In order to evaluate the performance of the ejector, Pressure Recovery Ratio is defined to represent the percentage of recovered pressure. The pressure recovery ratio will be significantly affected by the entrainment ratio. Fig. 39 shows the relationship between pressure recovery ratio and entrainment ratio. This curve shows that when entrainment ratio is relatively low, that means flow 1 is much high than flow 2, and the cooling capacity in evaporator 1 is much higher than that in evaporator 2, higher pressure recovery ratio can be achieved. This is a reasonable result. Inside in the ejector, when primary flow have larger flow rate, there will be more refrigerant with higher kinetic energy. When primary flow and secondary flow mix together, the total kinetic energy will be even larger. Thus the back pressure would be higher. While if the secondary flow have larger flow rate, there will be less refrigerant with high kinetic energy. Therefore when primary flow and secondary flow mix together, the total kinetic energy will also be smaller, thus the back pressure will be lower. For the situation that the flow 1 reaches 0, the recovered pressure will approach 0 respectively.
The energy consumption for both PRV based conventional MERS and ejector based MERS is plotted as shown in Fig. 40. As mentioned previously, the x-axis in Fig. 39 is the reference variable evaporator 1 cooling capacity.
Obviously, in Fig. 40, the compressor power consumption for PRV based conventional MERS is higher than that for ejector based MERS. And the higher evaporator 1 cooling capacity, the higher is the achieved compressor energy saving. Analysing Fig. 40 together with Fig. 37 and Fig. 39, when evaporator 1 cooling capacity is high, that means the flow rate of evaporator 1 is high. According to Fig. 37, the flow rate of evaporator 2 is relatively low, thus the entrainment ratio is also low. According to Fig. 39, the lower entrainment ratio, the higher is pressure recovery ratio and the higher is the compressor energy saving is. The reason for this result is that the power saved in ejector based MERS is by recovering the pressure loss from evaporator 1 to evaporator 2, thus if the flow rate in evaporator 1 is high, the throttle pressure loss will be high. In this experiment, the lowest energy saving is obtained when evaporator 1 flow rate= 21.86L/h, evaporator 2 flow rate= 34.07L/h, where the power consumption in PRV based conventional MERS is 1.99kW, and power consumption in ejector based MERS is 1.96kW. So, energy saving is 1.5%. The highest energy saving is obtained when evaporator 1 flow rate= 48.16L/h, evaporator 2 flow rate= 12.54L/h, where the power consumption in PRV based
conventional MERS is 2.38kW, and power consumption in ejector based MERS is 2.09kW. So energy saving is 12.2%.
Chapter 5 Conclusions and Future works

5.1 Conclusions

The ejector based MERS can recover the pressure loss during the throttle process, which happens in the PRV based conventional MERS, and achieve energy saving. The highest energy saving in the experiment is 12.2%, with improved ejector parameter setting; this value may be increased to 15% with high pressure recovery ratio.

The cooling capacity distribution between high temperature evaporator and low temperature evaporator has a significant effect on the energy saving result. Typically, a higher cooling capacity in high temperature evaporator will result in higher energy saving.

5.2 Future Work

Based on the results reported in this thesis, some future work extending on this research effort should focus on modeling, control and optimize of the system:

5.2.1 Modeling of Ejector based MERS

To develop real-time optimal operation strategy, simple and effective components models which can fit into different kinds of operation schemes, such as operations with and without air-conditioning, are essential. For both the ejector and vapor compressor cycles, there have been many investigations concerning elaborate models for system design and simulation of the system behaviors. However, most of these models are very complicated and not suitable for real time optimization purposes. Certain components such as compressors, fans, pumps and valves have been well established by polynomial methods and have been proven to be simple and accurate; the objective is to use the existing models for these components. Based on energy balance and heat and mass transfer analyses, the hybrid ejectors model which only require few characteristic parameters to predict the performance can be developed. Linear or non-linear least squares methods can be used for the parameters determining via curve fitting existing catalog data or real time experimental data[21, 22]. In future research work, this modeling approach can be used to model condensers, evaporators and ejectors together with the variable area ratio ejectors.

5.2.2 Optimizing the ejector based MERS

Normalized Integration Error Based Input-Output (IO) Selection: Due to the complicated relations among the system variables and the ambient environment conditions, the key trust for the system to be economically viable lies with reliable and optimal operation under different working conditions and cooling load demands. Sensible IO selection is primarily important for this system since: 1) the improper IO set and location may limit the performance, this cannot be solved by advanced controller design; 2) the IO set determines aspects such as reliability and the cost of hardware, implementation, operation, and maintenance; and 3) IO selection technique can be used to identify a particular way to improve the system controllability and flexibility. So far, there has been no systematic approach to this issue because of the size, complexity, and objectives that are associated with the development of the systems. An effective solution will include prioritization among competing or disparate objectives, reduction in the dimensionality of the problem and consistent means for the evaluation of alternatives, such that the optimization system
can be reliably implemented with the least cost. For this purpose, an approach for control structure synthesis will be developed to determine the types, numbers and locations of sensors and actuators such that the system variables can be effectively and economically measured and/or controlled. The multivariable system synthesis techniques [23] will be further developed to fit this particular application for the best sensor and actuator placement and control system configurations.

Model Based Optimization: Since the heats from the air-conditioned space are removed by several loops in a cascade manner (space → evaporators → compressor → condenser), the output of one loop becomes the set point or constraint of the subsequent loop; their operations are therefore affected by each other. Hence the physical limitations of each device, its operation window and the interactions of each loop have to be considered in the model based performance optimization. In this research, we can adopt two layers optimization structure, i.e. the overall optimization layer and the evaporator optimization layer. The overall optimization layer will determine the operating status of each evaporator according to the cooling demand and outdoor environment. In the evaporator optimization layer, each evaporator will determine its own optimal set-points while taking into consideration the couplings from other evaporators and following the instructions given by the top layer. Some new issues with real time optimization such as reliability, safety and robustness will also be addressed.

5.2.3 Control of Ejector based MERS

Relative Energy Based Advanced Controls: Since the system has multi-evaporator loops and each evaporator has to operate in a wide range of load conditions, the system is Multi-Input and Multi-Output (MIMO) in nature with severe nonlinear dynamic characteristics. Thus, the controllers in the system which are used to maintain an optimal operation, especially, pressure distribution have to adapt to the current operating conditions. Several advanced control schemes could be proposed in this research:

MIMO Control: For a given operating condition, depending on the open or closed loop status, and the requirement of the control system performance, several empirical dynamic modeling methods has been developed and can be readily adopted to identify the system parameters[24, 25]. The nonlinear models developed for optimization can also help to provide some prior knowledge such as the range of process gain so that the control system can use it to set step or relay size in test experiment; and also to validate the resulting linear model from the test by comparing its parameters values with the limits obtained from the nonlinear model. For controller design, relative energy based multi-input and multi-output controller design techniques will be devised for the system. Depending on loop characteristics and the requirements on system performance, decentralized, sparse or decoupling control schemes will be implemented into the control loops. Comparisons will be made on different control schemes to find the best solution for control.

Multi-model Control: In previous studied, a multi-model control strategy for HVAC processes [26, 27] has been proposed where linear local models that describe the process at nominal operating points are firstly developed and their weightings on the outputs are determined by fuzzy combination rules. An additional local regime is added and boundaries of fuzzy functions for partitioning are redistributed only when an unsatisfactory controller performance caused by model mismatch is detected. An additional local controller as well
as model is to be added to the global multi-model control system only when unsatisfied performance occurs. Once the multi-model structure has been determined, the overall controller output will be obtained by combining the local controller outputs using fuzzy combination. Consequently, the control performance based division will provide the proper structure, i.e. a minimum number of local models, for the whole operating range. Through membership of fuzzy sets, the combination of operating regimes can be described more smoothly and naturally. These topics are currently under study.
Reference