Investigation of Vapor Ejectors in Heat Driven Ejector Refrigeration Systems

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Abstract

Refrigeration systems, air-conditioning units and heat pumps have been recognized as indispensable machines in human life, and are used for e.g. food storage, provision of thermal comfort. These machines are dominated by the vapor compression refrigeration system and consume a large percentage of world-wide electricity output. Moreover, CO₂ emissions related to the heating and cooling processes contribute significantly to the total amount of CO₂ emission from energy use. The ejector refrigeration system (ERS) has been considered as a quite interesting system that can be driven by sustainable and renewable thermal energy, like solar energy, and low-grade waste heat, consequently, reducing the electricity use. The system has some other remarkable merits, such as being simple and reliable, having low initial and running cost with long lifetime, and providing the possibility of using environmentally-friendly refrigerants, which make it very attractive. The ERS has received extensive attention theoretically and experimentally.

This thesis describes in-depth investigations of vapor ejectors in the ERS to discover more details. An ejector model is proposed to determine the system performance and obtain the required area ratio of the ejector by introducing three ejector efficiencies. Based on this ejector model, the characteristics of the vapor ejector and the ERS are investigated from different perspectives.

The working fluid significantly influences the ejector behavior and system performance as well as the ejector design. No perfect working fluid that satisfies all the criteria of the ERS can be found. The performance of nine refrigerants has been parametrically compared in the ERS. Based on the slope of the vapor saturation curve in a $T-s$ diagram, the working fluids can be divided into three categories: wet, dry and isentropic. A wet fluid has a negative slope of the vapor saturation curve in the $T-s$ diagram. An isentropic expansion process from a saturated vapor state will make the state after the expansion to fall inside the liquid-vapor area of the $T-s$ diagram which will result in droplet formation. Generally, an isentropic expansion for a dry fluid will not occur inside the liquid-vapor area, and consequently no droplets will form. An isentropic fluid has a vertical slope of the vapor saturation curve in the $T-s$ diagram and an isentropic expansion process will hence follow the vapor saturation curve in the $T-s$ diagram, ideally without any droplet formation. However, when the saturation condition is close to the critical point, it is possible that the isentropic expansion process of a dry fluid and an isentropic fluid occurs inside the
liquid-vapor area of the $T$-$s$ diagram, resulting in formation of droplets. In order to avoid droplet formation during the expansion, a minimum required superheat of the primary flow has been introduced before the nozzle inlet. Results show that the dry fluids have generally better performance than the wet fluids and the isentropic fluid. Hence the thesis mostly focuses on the features of vapor ejectors and the ERS using dry fluids.

Exergy analysis has been proven to be very useful to identify the location, magnitude, and sources of exergy destruction and exergy loss, and to determine the possibilities of system performance improvement. This method is applied to the ejector and the ERS. The ejector parameters are closely interacting. The operating condition and the ejector area ratio have a great impact on the ejector overall efficiency and system COP. The ejector efficiencies are sensitive to the operating conditions, and they significantly influence the system performance. A so-called advanced exergy analysis is adopted to quantify the interactions among the ERS components and to evaluate the realistic potential of improvement. The results indicate that, at the studied operating condition, the ejector should have the highest priority to be improved, followed by the condenser, and then the generator.

Thermoeconomics, which combines the thermodynamic analysis and economic principles, is applied to reveal new terms of interest of the ERS. The economic costs of the brine side fluids (fluids that supply heat to the generator and evaporator and remove heat from the condenser) play very essential roles in the thermoeconomic optimization of the ERS. Depending on different economic conditions, the system improvement from a thermodynamic point of view could be quite different from the thermoeconomic optimization. The ERS is economically sound when using free heat sources and heat sink.

An ejector test bench has been built to test the entrainment ratio of different ejectors. Although the experiments do not achieve the desired results, they could still be discussed. The insignificant effect of the superheat of the secondary flow found in the theoretical study is validated. The assumption of neglecting the velocities at the ejector inlets and outlet are confirmed. The quantification of the ejector efficiencies shows that they largely depend on the operating conditions and the ejector dimensions.

**Keywords**

Ejector Refrigeration System; Ejector; Efficiency; Performance; Exergy Analysis; Experiment.
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Jianyong Chen
Stockholm, November 2014
Publications

This thesis is based on Papers I-V, which are enclosed at the end.

Paper I


Paper II


Paper III


Paper IV


Paper V

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5.3.2 Conclusions

5.4 Paper IV
1 Introduction

The development of industry and increasing living quality have caused a great and continually increasing demand of energy, and consequently, energy shortage and high energy prices. Associated with the increasing energy usage, serious environmental problems have been reported, like global warming, air pollution, production of toxic gases and ozone depletion. Refrigeration systems and air-conditioning units as well as heat pumps are dominated by the vapor compression refrigeration systems and are mainly driven by high-grade electricity. They account for between 10-20% of the world-wide electricity consumption and the demand is still increasing (Lucas, 1998). The latest percentage is about 17% (Lundqvist, 2014). The energy usage for refrigeration and heat pumping systems is projected to increase rapidly in the 21st century, from close to 300 terawatt hours (TWh) in 2000, to about 4000 TWh in 2050 and more than 10000 TWh in 2100. The associated CO₂ emissions from both heating and cooling increase from 0.8 gigatons of carbon (Gt C) in 2000 to 2.2 Gt C in 2100, which is about 12% of the total CO₂ emissions from energy use (Isaac and Vuuren, 2009). Meanwhile, the electrical peak demand from air conditioners during hot summer days can lead to escalating costs, blackouts and brownouts. There is therefore a global movement to reduce the primary energy use by means of decreasing the consumption of electricity and fossil fuels.

In the refrigeration field, researchers and engineers have been making continuous efforts to seek alternative approaches of utilizing renewable and sustainable energy, for example, solar radiation and geothermal energy, and to utilize waste heat, to drive refrigeration systems, air-conditioning units and heat pumps. Solar-driven air conditioning has a very interesting feature since solar radiation is generally in phase with cooling demands. Moreover, using solar energy to drive air conditioners will alleviate the problems of peak load in the electrical grid in summer.

Utilizing waste heat to produce a refrigerating effect enables a high recovery of energy from industrial processes and helps to mitigate the problems related to CO₂ emission from the combustion of fossil fuels as well as to reduce the cost. Of the heat-driven refrigeration systems, the absorption refrigeration system is now the most available on the market. Its main advantage over other heat-driven systems is a high COP. However, it is relatively complicated, and the corrosion and erosion inherent to working fluid pairs may cause failures, forcing the requirement of periodic maintenance. It is marketed only in the multi-kilowatt capacity ranges.
Another type, the adsorption refrigeration system, has abilities of using green refrigerants and low temperature heat sources as well as not involving any moving parts. But this system requires multiple adsorbent vessels to provide approximately continuous cooling capacity, and needs high design requirements to maintain a high vacuum. Moreover, the poor thermal conductivity in the adsorbent makes it unsuitable for large capacities. Both absorption and adsorption refrigeration systems are equipped with heavy weight components and are rather expensive.

The ejector refrigeration system, abbreviated ERS, is a very attractive alternative. Besides its ability of utilizing renewable and sustainable energy and the possibility of recovering waste heat, the ERS has many advantages, for instance, various options of refrigerant with a special interest of using environmentally-friendly ones. Other advantages include simple system construction, long lifespan, flexible system capacities, low capital cost and low maintenance. The main disadvantages are the relatively low COP and the difficulties ejector design, which greatly limit the wide spread use of such a system. A number of investigations on the ERS have been performed to gain a better understanding of its characteristics and to promote its applications.

1.1 The ejector and ejector refrigeration system

The editorial paper by Groll (2011) in a special issue of International Journal of Refrigeration, entitled *Ejector Technology*, started with *Ejectors are fascinating mechanical devices* and closed with *...very tricky but truly amazing flow devices*, which cannot be agreed with more.

The ejector is a flow device that allows a high pressure fluid, termed the primary fluid, to entrain a low pressure fluid (the secondary fluid) into the flow path, and discharge the mixed flow at an intermediate pressure that is higher than the secondary flow pressure. Thus the ejector acts like a compressor or a pump, but without any moving parts, lubricants and maintenance. The ejector is evaluated by the entrainment ratio, which is defined as the ratio of the mass flow rates between the secondary flow and the primary flow, and the pressure lift ratio, which is the ratio of the ejector outlet pressure to the secondary flow pressure at the ejector inlet.

Generally, the ejector has four parts: a nozzle which could be convergent shape or convergent-divergent shape, a suction chamber, a mixing chamber (a convergent part and a constant-area part) and a diffuser, as illustrated in Figure 1-1. The main geometry is characterized by the area ratio, which is
defined as the area of the constant-area part in the mixing chamber divided by the nozzle throat area.

Figure 1-1 Schematic drawing of the ejector with a convergent-divergent nozzle.

The primary flow acts as the driving, motive, or energizing fluid for the ejector, while the secondary flow is the driven, passive, or energized fluid. Both the primary and secondary flows can be in any flowing state, like liquid, vapor and two-phase. They can also be pure fluids or mixtures of non-identical fluids. The common ejector types are summarized in Table 1-1 (Elbel, 2011). Depending on its applications, the ejector has also been named synonymously as e.g. injector, eductor, thermal-compressor, jet pump, diffusion pump or aspirator.

<table>
<thead>
<tr>
<th>Primary flow</th>
<th>Secondary flow</th>
<th>Outlet flow</th>
<th>Remarks</th>
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<tbody>
<tr>
<td>Vapor ejector</td>
<td>Vapor</td>
<td>Vapor</td>
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<tr>
<td>Liquid ejector</td>
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<tr>
<td>Condensing ejector</td>
<td>Vapor</td>
<td>Liquid</td>
<td>Liquid</td>
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<tr>
<td>Two-phase ejector</td>
<td>Liquid</td>
<td>Vapor</td>
<td>Two-phase</td>
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</table>

The ejector has been known for a very long time. It was invented by Henry Giffard in 1858. The application was to pump liquid water to the reservoir of steam engine boilers replacing a mechanical liquid pump. In 1869, Schau firstly introduced a convergent-divergent nozzle to get a larger suction effect. The ejector gained a great popularity in steam locomotives to replace the pump. In 1901, Sir Charles Parsons used an ejector to remove air from the condenser of a steam engine. Ever since then the ejectors have been widely used in various processes, especially in the conditions of
requiring high reliability, harsh environments, and hazardous or combustible substances. Nowadays, they are very common in petrochemical industry, vacuum systems, power stations and nuclear reactors. Elbel (2011) briefly reviewed the history of ejectors, and it was later supplemented by Fischer (2013) with some important original sources.

Ejectors have two main applications in refrigeration systems: the ejector refrigeration system (ERS), using a vapor ejector to fulfill the function of a compressor, which is the theme of this thesis, and the ejector enhanced vapor compression refrigeration system, which applies a two-phase ejector as the expansion device for system performance improvement. The two systems are schematically illustrated in Figure 1-2 and 1-3, respectively.

The ejector was firstly introduced to refrigeration technology in 1910 by a French engineer, Maurice Leblanc, who introduced a cycle with a vapor ejector. It allowed producing a refrigerating effect by using low-grade energy like process steam from the plants or engines to drive the system. Thus it was called a steam jet refrigeration system and was mainly used in breweries, chemical plants, warships, etc. In 1926, Follain introduced a multi-stage steam jet refrigeration machine. There was much interest in this type of system to condition the air for large buildings in the early 1930s. Around 1955, a Russian engineer, Badylyes, was the first to develop the closed vapor jet refrigeration systems and used refrigerants other than water (Arora, 2010). However, it was gradually replaced by the more efficient vapor compression refrigeration systems. The development and refinement of the ERS have been almost at standstill since most efforts have been devoted to vapor compression refrigeration systems (Chunnanond and Aphornratana, 2004). As the energy and environmental issues arise during the last two decades, the ERS has attracted extensive and renewed attention by using various working fluids and different methods.

The ERS is driven by thermal energy, and consists of a generator, a condenser, an evaporator, a vapor ejector, a pump and a throttling device, as shown in Figure 1-2 with its thermodynamic diagrams. The working principle of the system is generalized as: low-grade heat \( Q_g \) is delivered to the generator for vaporization of the refrigerant \((g,i\rightarrow g,o)\). The high-pressure vapor out from the generator, i.e. the primary flow, enters into the ejector nozzle and draws low-pressure vapor from the evaporator, i.e. the secondary flow. The two flows undergo mixing and pressure recovery inside the ejector \((g,o\text{ and } e,o\rightarrow c,i)\). The mixed flow is then fed into the condenser, where condensation takes place by rejecting heat \( Q_c \) to the heat sink \((c,i\rightarrow c,o)\). The liquid from the condenser is divided into two parts. One
goes through the expansion device (c,o→e,i) to the evaporator (e,i→e,o), where it evaporates and hence produces a cooling effect ($Q_e$). The remaining liquid is pumped back to the generator by the pump (c,o→g,i), and completes the cycle. It is also recognized that the ERS can be considered as two loops, a power loop (an organic rankine cycle), and a refrigerating loop (a vapor compression refrigeration cycle).

Another appealing ejector application is to use a two-phase ejector in vapor compression refrigeration systems to reduce the exergy destruction associated with the throttling process and to enhance the system performance. It was firstly patented by Gay (1931). The ejector has been more popularly considered in CO$_2$ transcritical refrigeration systems. Layouts of the subcritical cycle and the CO$_2$ transcritical cycle are very similar. The only difference is the heat exchanger after the compressor, which is a condenser in the subcritical cycle, and a gas cooler in the CO$_2$ transcritical cycle, as shown in Figure 1-3. This type of system is however not the objective of this thesis. More details about the ejector enhanced vapor compression refrigeration systems can be found in Elbel (2011), Sarkar (2012) and Sumeru et al. (2012).
The other two types of ejectors (liquid ejectors and condensing ejectors) in Table 1-1 have also been applied in refrigeration systems. As this study only focuses on the vapor ejector and the ERS, the other types of ejectors are not discussed here.

1.2 Literature survey

The application of ejectors in refrigeration technology has been studied theoretically and experimentally for a long time. The related work has been summarized in several review papers with focus on different perspectives. For instance, Chunnanond and Aphornratana (2004) gave a general introduction on ejector applications in refrigeration systems; Riffat et al. (2005) presented a review of the main studies on ejectors and their applications during the period of 1995 to 2005; Chen et al. (2013b) overviewed the ejector modeling progress; Abdulateef et al. (2009) presented a survey on solar-driven ejector refrigeration systems; Sarkar (2012) and Sumeru et al. (2012) reported the status of the two-phase ejector as an expansion device to enhance vapor compression refrigeration systems and heat pumps; Elbel (2011) focused on ejector enhanced CO₂ transcritical refrigeration systems. A literature review on the state-of-art various refrigeration systems integrated with ejectors (Paper VI) has also been conducted during the course of this project. The literature survey presented here concerns only typical vapor ejector models and experimental aspects of vapor ejectors.

1.2.1 Vapor ejector models

The flows inside the vapor ejector are very complex, involving expansion, supersonic flow, choked phenomena, flow mixing, flow interaction, shock trains, etc. The detailed flow information is not yet quite clear. The ejector processes are not easily formulated mathematically. However, by using appropriate assumptions and simplifications, the ejector can be modeled with the assistance of the conservation laws of mass, momentum and energy, gas dynamic equations, equations of state and isentropic relations.

A very early profound analysis of the ejector was conducted by Keenan et al. (1950), who proposed two feasible methods to describe the ejector mixing process: the constant-pressure mixing model, in which the pressure of the mixing process was assumed constant, and the constant-area mixing model, where the mixing process was assumed to occur in the constant area part of the ejector. These two ejector models have become the basis of
many other developed ejector models and the fundamental theories for ejector design ever since. However, they cannot predict the constant entrainment ratio of a fixed-geometry ejector. This was later explained by Munday and Bagster (1977), who made an important assumption that two chokes occurred in the ejector. One was the primary flow at the nozzle throat, the other one was the secondary flow at the “hypothetical throat” which resulted from the assumption that the primary flow out from the nozzle was not mixing with the secondary flow immediately, introducing a convergent duct where the secondary flow was choked. To include the irreversibilities in the ejector, Eames et al. (1995) modified Keenan’s model (Keenan et al., 1950) by introducing three efficiencies in the nozzle, mixing chamber, and diffuser, respectively. Huang et al. (1999) developed a more sophisticated one-dimensional ejector model from Munday and Bagster’s theory (Munday and Bagster, 1977) to predict the ejector performance at critical mode operating condition. The double choked flow in the ejector was well derived in the model. Chou et al. (2001) proposed that there was a third choked flow in the fully mixed flow that took place before the diffuser. They focused on the development of a shear mixing layer to formulate the secondary choked flow at the “hypothetical throat”. This model was able to determine the maximum entrainment ratio of the ejector. However, the processes were assumed to be isentropic. A developed model from Huang et al. (1999) was presented by Kumar and Ooi (2014) in a way of applying the Fanno flow concept to capture frictional compressible flow in the mixing chamber and averaging the specific heat capacity ratio to improve the accuracy of predictions. Chen et al. (2013a) proposed an ejector model to predict the ejector performance at critical and sub-critical operation modes.

The models mentioned above are the constant-pressure mixing model. The constant-area mixing model has not been widely studied. Grazzini and Mariani (1998) used the constant-area mixing model in a two-stage ejector to achieve a larger compression ratio. Addy et al. (1981) proposed a one-dimensional method to analyze a constant-area ejector without diffuser. Yapıcı and Ersoy (2005) incorporated a diffuser in Addy’s model (Addy et al., 1981) to determine the optimum system COP and area ratio. The nozzle efficiency and the diffuser efficiency were thus taken into consideration in the model.

Although the shocks contribute to the compression effect, they also introduce large irreversibilities. Eames (2002) presented the constant rate of momentum change (CRMC) method to produce a diffuser that allowed the momentum to change at a constant rate when passing through the diffuser.
As a result, the static pressure could rise gradually from entry to exit in the diffuser, instead of the sudden increase by the normal shock, and the total pressure loss related to the shock process could be decreased.

The one-dimensional ejector models have shown their effectiveness in predicting the ejector performance. Multi-dimensional models have been developed as well. Zhu et al. (2007) introduced a “shock circle” at the entrance of the constant-area chamber and adopted a 2D exponential expression for velocity distribution near the ejector inner walls to calculate the ejector performance. This 2D model was independent of the flow in the constant area chamber and diffuser. The ejectors with two and three dimensions have been extensively investigated using Computational Fluid Dynamics (CFD), such as by Varga et al. (2009) and Gagan et al. (2014).

To eliminate the errors introduced by the assumption of ideal gas behavior of the fluid, Yu et al. (2007) used the properties of refrigerant from the NIST database to formulate the ejector working processes with irreversibilities in terms of efficiencies. The real gas properties of R245fa from the same database were used by Grazzini et al. (2012) in their ejector model with the aim of designing the ejector. The real properties of R134a were used by Khalil et al. (2011) with a concern on the friction on the wall surface of the constant area section, and defined two isentropic efficiencies in the nozzle and diffuser. Valle et al. (2012) claimed that the results from the ideal gas assumption were close to those obtained from real gas behavior, and confirmed the validity of assuming the refrigerant R141b as an ideal gas. Abdel-Aal et al. (1990) stated that the behavior of steam as an ideal gas was very similar to that of its real gas behavior. However, Grazzini et al. (2011) used steam as the working fluid to check the ideal gas assumption in the ejector processes, and it was shown that the results from the ideal gas law were far away from reality due to the appearance of condensation inside the ejector, and the effects of condensation could not be eliminated by a slight superheat of the primary flow.

Depending on the slope of the vapor saturation curve in the $T$-$s$ diagram, the working fluids are categorized as wet fluids, dry fluids and isentropic fluids. For the wet fluids, droplets may be formed during an isentropic expansion, like in steam power cycles. The dry and isentropic fluids normally do not have droplet formation during expansion. Therefore, the boundary conditions for the wet fluids differ from those of the dry fluids and isentropic fluids. Zhu and Li (2009) proposed a model to evaluate the ejector performance working with the dry and wet fluids. They used the ideal gas law to calculate the velocity of the primary flow at the entrance of
the constant area section for the dry fluid, while for wet fluids the velocity was obtained by simply applying an energy balance. Cardemil and Colle (2012) used real gas equations of state for both dry and wet fluids. For wet fluids a relaxation model was employed to calculate the sound speed in two-phase mixtures. Sharifi et al. (2013) numerically simulated a steam ejector with the assumption of homogenous nucleation inside and analyzed the effects of steam condensation on the ejector aerodynamics and thermodynamic performance. The obtained results were found to be quite different from those of assuming steam as a dry fluid.

The main findings of these studies can be concluded as: (1) for a fixed-geometry ejector, a critical back pressure exists and limits the ejector flow rate as well as its performance. The entrainment ratio decreases dramatically beyond this critical condition. A variable-geometry ejector enables to smooth the sharp drop of the entrainment ratio in a fixed-geometry ejector and maximize the performance; (2) the ideal gas assumption has been extensively used in the ejector models. It has been proven that it is simple and effective, and its predictions of the ejector performance are acceptable for some refrigerants; (3) the droplet formation during the ejector process could largely influence the ejector behavior and mathematical ejector model formulation as well as the ideal gas assumption; (4) most of these models are either used for ejector design, where the ERS operating conditions and the entrainment ratio are predefined, or applied to performance evaluation, where operating conditions and the ejector geometries are known, and mainly dealing with fixed-geometry ejectors; (5) both the ejector behavior and whole system performance are significantly dependent on the ejector geometries, operating conditions and working fluid properties.

### 1.2.2 Experimental aspects

Experimental studies are always irreplaceable. They are used to validate the ejector models and to investigate the operating characteristics of the ejector and the ERS.

Water has been used in the ERS since 1910. Until now, it is still widely used in the studies. Eames et al. (1995) built a small-scale steam ERS to study its performance. They used a fixed-geometry ejector that had an area ratio $Ar$ of 90 with a 2 mm throat diameter ($D_t$). Temperatures in the generator $T_g$, the condenser $T_c$ and the evaporator $T_e$ were in the range of 120-140 °C, 26.5-26.3 °C, and 5-10 °C, respectively. The measured COP was from 0.18 to 0.59. Pollerberg et al. (2008) tested a solar-driven steam ERS with a cooling capacity of 1 kW and claimed the COP was in the range of
0.38-0.86 with \( T_g=120-135 \, ^\circ\text{C} \), \( T_c=24-36 \, ^\circ\text{C} \) and \( T_e=7 \, ^\circ\text{C} \). Chandra and Ahmed (2014) experimented with two steam ejectors. One was a conventional ejector, and another was designed based on CRMC proposed by Eames (2002). Both ejectors had the same \( D_{th}=3.5 \, \text{mm} \) and \( Ar=32.7 \). Results confirmed that the later ejector had higher entrainment ratios in the range of 0.23-0.28 when \( T_g=90-120 \, ^\circ\text{C} \), \( P_c=3.5-5.8 \, \text{kPa} \) (\( T_c=26.6-35.5 \, ^\circ\text{C} \)) and \( T_e=10 \, ^\circ\text{C} \).

The main drawback of using water as a refrigerant is the limit of refrigerating temperature to be above 0 \(^\circ\text{C}\). Since halocarbon refrigerants emerged in the 1930s, they have been widely used in the ERS. Yapıcı and Yetisen (2007) experimentally studied the effects of operating conditions on system performance with R11 as the working fluid in a wide range: \( T_g=90-102 \, ^\circ\text{C} \), \( P_c=114-143 \, \text{kPa} \) (\( T_c=27.1-33.8 \, ^\circ\text{C} \)) and \( T_e=0-16 \, ^\circ\text{C} \). The used ejector had a \( D_{th}=2.6 \, \text{mm} \) with \( Ar=10.94 \). A COP of 0.25 was achieved. Al-Khalidy (1998) tested the performance of R113 in the ERS, the maximum cooling capacity was obtained with a COP of 0.42 when \( P_g=320 \, \text{kPa} \) (\( T_g=87.3 \, ^\circ\text{C} \)), \( P_c=85 \, \text{kPa} \) (\( T_c=42.4 \, ^\circ\text{C} \)) and \( T_e=18 \, ^\circ\text{C} \). Yapıcı et al. (2008) examined six ejectors with a range of \( Ar \) from 6.5 to 11.5 and used R123 as the working fluid in the ERS. The measured COP increased from 0.29 to 0.41 in an approximately linear trend as the optimum generator temperature \( T_g \) increased from 83 to 103 \(^\circ\text{C}\) and \( P_c=125\, \text{kPa} \) (\( T_c=33.7 \, ^\circ\text{C} \)), \( T_e=10 \, ^\circ\text{C} \). For a given ejector area ratio, there existed an optimum generator temperature at which a maximum COP was obtained. Huang et al. (1999) carried out an experiment using 11 ejectors with \( Ar \) in the range of 6.44-10.64 and R141b as the working fluid to verify the ejector model and to determine the coefficients introduced in the model. The ejector entrainment ratios were obtained from 0.18-0.61, depending on the operating conditions (\( T_g=78-94 \, ^\circ\text{C} \), \( T_c=24.4-42.5 \, ^\circ\text{C} \), \( T_e=8 \) and 12 \(^\circ\text{C}\)) and the used ejector. Scott et al. (2011) carried out an experiment with R245fa, and the entrainment ratio was in the range of 0.1-0.58 at the following conditions: \( T_g=80-120 \, ^\circ\text{C} \), \( T_c=15-40 \, ^\circ\text{C} \), \( T_e=0-20 \, ^\circ\text{C} \) and the ejector had \( D_{th}=2.98 \, \text{mm} \). Wang et al. (2009) replaced the pump by a multi-function generator as a thermal pumping device and tested its performance. For an ejector having a \( D_{th} \) of 2.8 mm and \( Ar \) of 11.03 working with R365mfc, the system COP varied from 0.102 to 0.446 as \( T_g=90 \, ^\circ\text{C} \), \( T_c=39.6 \, ^\circ\text{C} \) and \( T_e=7.6-25.7 \, ^\circ\text{C} \). Yan and Cai (2012) employed R134a as the working fluid and six ejector area ratios from 2.74 to 5.37, and tested them at following conditions: \( T_g \) varying from 72 to 78 \(^\circ\text{C}\), a fixed \( T_g \) at 31 \(^\circ\text{C}\) and three \( T_c \) (8 \(^\circ\text{C}\), 10 \(^\circ\text{C}\) and 12 \(^\circ\text{C}\)). It was claimed that the optimum area ratios increased almost linearly with the increasing of the primary flow pressure. At a given operating condition, the cooling capacity
related to the area ratios and nozzle diameters, while the COP depended only on the area ratio. Due to environmental concerns, halocarbon refrigerants that have been the most prominent are facing a possible ban. Therefore, researchers have to turn to the hydrocarbon and natural alternatives. Except water, other environmentally-friendly refrigerants, such as R717, R290 and R600a, have been used in the experimental studies in the ERS. Sankarlal and Mani (2007) constructed a R717 ERS test bench operating at $T_g=62-72 \, ^\circ C$, $T_c=30-36 \, ^\circ C$ and $T_e=5-15 \, ^\circ C$. Three ejectors with different area ratios of 4.0, 5.76 and 8.16 were used. The entrainment ratio was obtained from 0.168 to 0.289. Chen et al. (2013a) tested the performance of R290 in the ERS in the following operating range: $T_g=60-95 \, ^\circ C$, $T_c=25-45 \, ^\circ C$ and $T_e=10-15 \, ^\circ C$ with an ejector having a $D_{th}$ of 2.19 mm and $Ar$ of 4.11. The measured ejector entrainment ratio varied from 0.2 to 0.4. The ERS working with R600a was experimentally studied by Butrymowicz et al. (2014) with an ejector ($D_{th}=3.5 \, \text{mm}$ and $Ar=2.94$). It was found that a maximum entrainment ratio of 0.23 was reached at $T_g=55 \, ^\circ C$, $T_c=24 \, ^\circ C$ and $T_e=3.5 \, ^\circ C$.

Most aforementioned experimental studies were focused on one or several ejectors with fixed geometries. It has been generally confirmed that the ejector performance significantly depends on its operating conditions and geometrical characteristics. To make the ejector more adaptable to different operating conditions and to optimize its performance, some interesting ideas have been proposed to realize a variable-geometry ejector. Ma et al. (2011) introduced a moveable spindle in the nozzle to change nozzle throat area and control the primary flow rate for a steam ejector. It was found that the spindle position had a significant impact on the cooling capacity and the ejector critical back pressure, but only a slight influence on the ejector entrainment ratio and system COP. Eames et al. (2013) tested two R245fa ejectors with 8.6 and 13.2 in $Ar$, respectively, and both were equipped with a movable primary nozzle to study the dependency of system performance on the nozzle exit position (NXP). At a given operating condition, there existed an optimum NXP leading to a maximum COP for both ejectors. The obtained COP was in the range of 0.08-0.25 over the operating conditions ($T_g=100-110 \, ^\circ C$, $T_c=31.8-37.3 \, ^\circ C$ and $T_e=7-13 \, ^\circ C$).

Instead of testing an entire ERS, some open test bench has been built to only focus on the ejector itself. Gagan et al. (2014) used an air compressor to lift the primary flow pressure to 743 kPa, while pressures of the secondary flow and at ejector outlet were kept at 86.3 kPa and 137 kPa, respectively, by control valves. The ejector had a $D_{th}$ of 3.5 mm and $Ar$ of
8.16. The results were used to guide the selection of the turbulence model for a CFD simulation. Zhu and Jiang (2014) built an open system to experiment on a N₂ ejector ($D_{th}=3.6$ mm). The test bench consisted of two N₂ storage tanks and several valves to control the pressures for the primary flow (450-650 kPa) and the secondary flow (50-70 kPa), while the ejector outlet was open to the ambient. The entrainment ratio was in the range of 0.16-0.45. Moreover, Schlieren flow visualization was used to observe the shock wave structure inside the ejector mixing chamber.

Conclusively, various refrigerants are applicable in the ERS. The ejector geometries are critical and have significant influence on the system performance. The design of the ejector area ratio $Ar$ is well established, and it depends on the chosen refrigerant and the operating conditions of the ERS. However, there are few universally accepted guideline for designing the remaining geometrical parameters, like the nozzle area ratio (nozzle exit area divided by nozzle throat area), nozzle exit position (NXP), the selection of convergent and divergent angles, the length of the constant area mixing chamber, etc. These geometries also significantly influence the ejector performance. There is an absolute lack of details of geometrical design for an ejector in the ERS, especially for the variable-geometry ejector.

1.3 Aims of the study and structure of the thesis

The Division of Applied Thermodynamics and Refrigeration in Department of Energy Technology at KTH has since many years been performing research on the ejector refrigeration system. A previous doctoral dissertation by Pridasawas (2006), entitled “Solar-Driven Refrigeration Systems with Focus on the Ejector Cycle”, has given excellent interpretations of the ERS. A very comprehensive overview of solar-driven refrigeration systems was presented. Studies of both steady-state and dynamic features of the solar-driven ERS were carried out.

The present research is a continuation with an in-depth investigation of the ERS. The overall objective of this study is to gain a better understanding and knowledge of the vapor ejectors as well as the ERS. It aims to reveal more details of the ejector behavior and the ERS working characteristics, and to optimize the system as well as to promote a wide usage of such system.

The thesis is divided into 7 chapters and includes theoretical investigations of the ejector and the ERS, which are presented in Chapters 2,
3, and 5, and reported in Paper I-V, as well as an experimental study of the ejectors which is presented in Chapters 4 and 6.

Chapter 1 gives the background and general information of the ERS. A compact literature review on various ejector mathematical models and experiments concerning the vapor ejector and the ERS is presented with related details, such as the operating conditions, main geometries of the used ejectors, and the obtained results, to give an overview of the development and status of research activities. The objectives and limitations of the thesis are presented as well.

Chapter 2-4 are the methodology parts of the thesis. Chapter 2 introduces the basis of the ejector process and ejector modeling, largely based on Paper I. A validation of using the ideal gas law in the proposed ejector model is of interest to present as an important supplement. Based on the proposed ejector model, Chapter 3 extends the theoretical research from different perspectives, including how to select the working fluid in the ERS, an exergy analysis of the ejector itself, an advanced exergy analysis of the ERS and a thermoeconomic optimization of the ERS. Paper II-V correspond to each specific topic. Chapter 4 describes the test bench and the experimental instrumentations as well as the uncertainty analysis.

Chapter 5 and Chapter 6 are the results and discussion part of the thesis. Chapter 5 is the summary of the five papers included in the thesis and presents the main results from the theoretical investigations. An extended study of Paper V is carried out to show the sensitivity of different economic variables to the objective functions. Chapter 6 shows the experimental findings. Some theoretical results and assumptions in the proposed ejector model are validated.

Finally, conclusions and suggestions for future studies are presented in Chapter 7.

1.4 Limitations

This study investigates the vapor ejectors and the ERS. Attempts to optimize the system have been carried out. However, it is subjected to some limitations, which are addressed as following:

The study focuses on a system that is at the steady state, dynamic operating conditions are not considered. When modeling the ERS, it assumes that the refrigerating capacity is constant, and heat exchanger
models are not included, except for the thermoeconomic analysis. In addition, only pure refrigerants are used.

The results of the prediction from the proposed ejector model in Paper I is very sensitive to the ejector efficiencies, i.e. the nozzle efficiency, mixing efficiency and diffuser efficiency, which determine the prediction accuracy. Their values can be determined by experimental data, thus the proposed model is an empirical ejector model. The ejector efficiencies are influenced by e.g. the ejector design, manufacturing methods, operating conditions and the working fluids. Although some correlations of the ejector efficiencies are established in Paper III based on test data from Huang et al. (1999), they are only validated for R141b at the experimental working ranges. The experimental part gives very limited knowledge on how to choose the three ejector efficiencies.

The proposed ejector model only deals with one geometrical parameter: the area ratio. Other geometrical parameters are not involved, which are also very crucial for the ejector design.

Regarding the experiments, the major problem comes from the ejector design. The performance of the ejectors is not as good as desired. The pressure of the primary flow and the mixed flow are able to meet the required values. However, the secondary flow cannot achieve the low required pressure. This may also be related to the ineffective pressure controls, which are achieved through needle valves and adjusted manually.

In the proposed ejector model, the nozzle efficiency is defined as an isentropic efficiency, Eq. (2-5), and its value is considered as an input data in the model. In the experimental investigation, an attempt to quantify the nozzle efficiency has been made. Ideally, real gas properties (REFPROP Lemmon et al., 2010) should be used. However, in the present experimental setup, there is no way of determining the state after the nozzle. Instead, the nozzle efficiency is evaluated by assuming ideal gas behavior together with experimental data, as is given in Eq. (4-14). Thus errors are introduced.
2 Ejector Modeling

The ejector is the key component in the ejector refrigeration system. The processes in the ejector are very complex in reality. However, based on some assumptions, the ejector processes can be simplified and modeled mathematically. This chapter presents the general information of typical ejector processes, and introduces the ejector model proposed in Paper I.

2.1 Ejector processes

The ejector processes involves many complicated flow phenomena. The changes in pressure and velocity along the ejector are schematically shown in Figure 2-1.

To facilitate the formulation of ejector processes, the following assumptions are made to simplify the analysis:

1. The flow inside the ejector is steady state and one-dimensional. The ejector walls are adiabatic.
2. Velocities of the primary flow and the secondary flow before entering the ejector are negligible, i.e. states g,o and e,o in Figure 2-1, respectively. The velocity of the mixed flow leaving the ejector (state c,i in Figure 2-1) is also neglected.

3. Irreversibilities in the ejector are taken into account by using isentropic efficiencies in the nozzle ($\eta_n$), in the diffuser ($\eta_d$), as well as a mixing efficiency in the mixing chamber ($\eta_m$), which is a kinetic energy ratio, which are the same as in Yu et al. (2007).

Based on these assumptions, the ejector processes in the $h$-$s$ diagram is shown in Figure 2-2, in which the numbers are corresponding to those in Figure 2-1. The dashed line $g,o \rightarrow 2'$ is the ideal process in the nozzle (isentropic expansion), while the line $g,o \rightarrow 2$ represents the real nozzle process by taking into account the nozzle efficiency $\eta_n$. State 3 represents a state at which the primary flow and the secondary flow reach the same velocity. State 4' is the end of the ideal process in the mixing chamber, and state 4 is the actual process end with the mixing efficiency $\eta_m$. The dashed line $4 \rightarrow c,i'$ is the isentropic process in the diffuser, and the line $4 \rightarrow c,i$ is the real process with the diffuser efficiency $\eta_d$. State 5 is after the shock. The pressure $P'$ is the mixing pressure, which is lower than the secondary flow pressure $P_e$. The line $e,o \rightarrow 0$ represents the process of entraining the secondary flow from the evaporator into the ejector.

![Figure 2-2 h-s diagram of the ejector working processes (Paper I).](image)

The working processes in the ejector are divided into four parts in the proposed ejector model (Paper I), and they are described as follows with assistance of Figure 2-1 and Figure 2-2.
2.1.1 Primary flow through nozzle to 2 (g,o→2)

Before describing the process of the primary flow through nozzle to 2, a very important phenomenon has to be introduced, i.e. the choked flow at the nozzle throat. Choked flow occurs under a specific condition when a fluid with a high pressure enters into a low pressure region through a constricted flow area where the velocity becomes equal to the speed of sound (Mach number is one, \( M=1 \)). As a result, the mass flow rate becomes independent of the low pressure at the downstream. In other words, the mass flow rate of a choked flow is constant as the downstream pressure falls below a certain level, and a further decrease in the downstream pressure does not increase the flow rate. The mass flow rate of a choked flow depends on the flow area, and the upstream state (pressure and temperature).

The primary flow through nozzle to 2 can be described as: the high pressure and temperature vapor (\( P_g \) and \( T_g \)), leaving the generator (g,o), enters into the convergent-divergent nozzle and undergoes an expansion process. The pressure decreases and velocity increases in the convergent part. The velocity reaches sonic speed at the nozzle throat (1), where the flow is choked. At the divergent part, the velocity continues increasing to supersonic velocity as the pressure decreases. A low pressure region is created at the nozzle exit. The isentropic expansion and the real expansion are used during this process.

Conservation of energy of a steady state flow passing a control volume between the inlet and outlet is generally given as (Munson et al., 2006):

\[
\dot{m} \left[ (h_{\text{out}} - h_{\text{in}}) + \frac{1}{2} \left( v_{\text{out}}^2 - v_{\text{in}}^2 \right) + g \left( z_{\text{out}} - z_{\text{in}} \right) \right] = W_{\text{shaft net in}} + Q_{\text{net in}} \quad (2-1)
\]

The inlet velocity \( u_{g,o} \) is negligible and the heat loss \( Q_{\text{net in}} \) is zero according to the assumptions. No shaft work is exchanged in the nozzle, and \( W_{\text{shaft net in}} \) can therefore be omitted. Moreover, the potential energy change is negligible. Thus, referring to Figure 2-2, Eq. (2-1) can be simplified as:

\[
u_2 = \sqrt{2 \cdot (h_{g,o} - h_2)} \quad (2-2)
\]

By introducing the nozzle isentropic efficiency, Eq. (2-2) is rewritten as (Yu et al., 2007, Paper I):
2.1.2 Secondary flow from inlet to 0 (e,o→0)

Since the mixing pressure $P'$ is lower than the secondary flow pressure $P_e$, the secondary flow ($P_e$ and $T_e$) from the evaporator (e,o) is entrained into the ejector (0), resulting in a pressure reduction from $P_e$ to $P'$ and increasing of the velocity to $u_0$.

Using the assumptions and recalling that no heat or work is exchanged and that the change in potential energy in the flow is negligible, the velocity of the secondary flow at the state 0 (see Figure 2-2) is written as Eq. (2-6) from the conservation of energy equation Eq. (2-1)(Paper I):

\[ u_2 = \sqrt{2 \cdot \eta_n \cdot (h_{g,o} - h_2)} \] (2-3)

\[ h_2 = f(P', s_{g,o}) \] (2-4)

\[ \eta_n = \frac{h_{g,o} - h_2}{h_{g,o} - h_2'} \] (2-5)

This process also introduces irreversibilities. However, the process has much smaller pressure difference compared to the expansion in the nozzle and a moderate velocity increase; the process is therefore assumed to be isentropic.

2.1.3 Mixing process before shock (0 and 2→4)

The primary flow and the secondary flow begin to mix at state 2. At the beginning, the primary flow is retarded while the secondary flow continues accelerating. They are completely mixed at state 3 at which they have the same velocity and pressure. The mixed flow still develops in the constant-area mixing chamber until the end of this chamber (state 4). The ideal mixing process and the real mixing process are adopted.

Conservation of momentum for a control volume between the inlet and outlet in the mixing chamber can be simply written as:

\[ P_{in} \cdot A_{in} + \dot{m}_g \cdot u_2 + \dot{m}_e \cdot u_0 = P_{out} \cdot A_{out} + (\dot{m}_g + \dot{m}_e) \cdot u_4' \] (2 - 8)
where at the constant-area of the mixing chamber, the mixing pressure is constant and the areas are the same. For an ideal mixing process, Eq. (2-8) is transferred as:

\[ \dot{m}_g u_2 + \dot{m}_e u_0 = (\dot{m}_g + \dot{m}_e) u_4 \]  

(2-9)

Using the definition, the ejector entrainment ratio \( \mu \) is

\[ \mu = \frac{\dot{m}_e}{\dot{m}_g} \]  

(2-10)

Then Eq. (2-9) is rewritten as (Paper I):

\[ u_4' = \frac{u_2 + \mu u_0}{1 + \mu} \]  

(2-11)

Defining the mixing efficiency as a kinetic energy ratio between the real mixing and ideal mixing process, the velocity of the real mixing process is obtained as (Paper I):

\[ \eta_m = \frac{u_4^2}{u_4'^2} \]  

(2-12)

\[ u_4 = \frac{u_2 + \mu u_0}{1 + \mu} \sqrt{\eta_m} \]  

(2-13)

2.1.4 Mixed flow through the diffuser (4→c,i)

Due to the supersonic flow at the constant-area mixing chamber, a shock, which, for simplicity of modeling purposes, is assumed as a normal shock, occurs just before the diffuser. The normal shock is common in supersonic compressible flow. It is a thin transitive area propagating with supersonic flow and results in almost instantaneous changes of the properties, like increases in density and pressure, and a dramatic decrease in velocity from supersonic to subsonic. The shock ends at state 5.

The mixed flow then passes through the diffuser and converts kinetic energy into static pressure rise, leading to further pressure increase and velocity decrease. Finally, the flow fans out to the condenser (c,i) at the condenser pressure \( P_c \). The ideal process and the real process are considered. An isentropic efficiency for the diffuser is used.
Applying Eq. (2-1) and neglecting the velocity at the ejector outlet in this process, the following equations can be obtained (Paper I):

\[ h_{c,i} = h_4 + \frac{1}{2} \eta_4^2 \]  

(2-14)

\[ \eta_d = \frac{h_{c,i} - h_4}{h_{c,i} - h_4} \]  

(2-15)

\[ h_{c,i} = f(P_c, \delta_4) \]  

(2-16)

It should be noted that the shock process has been integrated in the energy equation, i.e. Eq. (2-14).

The mixing pressure \( P' \) is coupled by gas dynamic equations by using the ideal gas law, which can be found in Paper I. Iteration processes are needed to calculate the ejector entrainment ratio \( \mu \) and the ejector outlet temperature. Furthermore, a correlation to calculate the ejector area ratio \( Ar \) is adopted for geometrical analysis.

The proposed model can be used to evaluate the system performance and to obtain the corresponding ejector area ratios. The ejector geometries (area ratio \( Ar \), to be specific) needs to vary accordingly along with changes of working fluids and operating conditions, and for each condition, there exists an area ratio \( Ar \) to ensure the ejector behavior and system performance. The detailed mathematical formulations of the ejector processes can be found in Paper I.

2.2 Feasibility of using the ideal gas law in the proposed ejector model

The ideal gas assumption has been widely used in ejector models to formulate the ejector processes (Keenan et al., 1950; Huang et al., 1999; Yapıcı and Ersoy, 2005). The proposed ejector model in Paper I also applies the ideal gas law, allowing analytical expressions to be derived for the ejector entrainment ratio \( \mu \) and the area ratio \( Ar \). Although it has been claimed that the results from the ideal gas law are very close to the predictions from real gas properties (Abdel-Aal et al., 1990; Valle et al., 2012), the feasibility of using the ideal gas law in the proposed ejector model is further studied.

The compressibility factor \( Z_r \) measures the deviation of real gas behavior from ideal gas law. It is defined as the ratio of the volume occupied by a real
gas to the volume occupied by an ideal gas under the same temperature and pressure conditions.

\[ Z_r = \frac{P \cdot V}{(R \cdot T)} \]  

(2-17)

where \( R \) is the gas constant. If \( Z_r \) is less than one, the deviation is largely due to attractive forces between molecules, while if \( Z_r \) is greater than one, the deviation can be ascribed to the volume that is taken up by individual molecules treated as hard spheres or to repulsive forces, or both (Rogers, 2010). For an ideal gas the compressibility factor \( Z_r \) by definition equals to one. The closer the value \( Z_r \) of a real gas at a specific condition is to one, the less its behavior deviates from the behavior of an ideal gas.

The compressibility factor \( Z_r \) at the typical conditions inside the ejector according to the proposed ejector model (Paper I) is calculated from REFPROP (Lemmon et al., 2010), and presented in Table 2-1. Four working fluids, R123, R141b, R245fa and R600a, which are mostly used in the thesis, are selected. The ERS is operating at the following conditions: \( T_g = 95 \, ^\circ C, \, T_c = 35 \, ^\circ C, \, T_e = 10 \, ^\circ C \), and with two sets of fixed ejector efficiencies, Case 1: \( \eta_n = 0.95, \, \eta_m = \eta_d = 0.85 \), which is close to the test data based on Paper III, and Case 2: \( \eta_n = \eta_m = \eta_d = 0.9 \), which is used in the parametric studies in Paper II-IV.

### Table 2-1 Compressibility factors inside the ejector.

<table>
<thead>
<tr>
<th>Working fluids</th>
<th>Typical locations in the ejector in Figure 2-1 and Figure 2-2</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( g,o )</td>
</tr>
<tr>
<td><strong>Case 1</strong></td>
<td></td>
</tr>
<tr>
<td>R123</td>
<td>0.837</td>
</tr>
<tr>
<td>R141b</td>
<td>0.871</td>
</tr>
<tr>
<td>R245fa</td>
<td>0.769</td>
</tr>
<tr>
<td>R600a</td>
<td>0.681</td>
</tr>
<tr>
<td><strong>Case 2</strong></td>
<td></td>
</tr>
<tr>
<td>R123</td>
<td>0.837</td>
</tr>
<tr>
<td>R141b</td>
<td>0.871</td>
</tr>
<tr>
<td>R245fa</td>
<td>0.769</td>
</tr>
<tr>
<td>R600a</td>
<td>0.681</td>
</tr>
</tbody>
</table>

As seen in Table 2-1, the values of the compressibility factor \( Z_r \) at the inlet of the ejector nozzle (\( g,o \)) are the lowest. Hence, at the ejector nozzle entrance the refrigerants have the largest deviation from ideal gas behavior. To avoid large modeling errors, real gas data (REFPROP Lemmon et al.,
2010) has been used for modeling the flow through the nozzle in the model (see Eq. (2-2) through Eq. (2-5)). For the rest of the ejector, the ideal gas assumption is applicable since from state 2 and onwards, the compressibility factor is sufficiently close to one. However, Eq. (2-18) has been used for simplifying the iteration process to obtain the mixing pressure (Paper I). As Eq. (2-18) is derived by using the ideal gas assumption, errors are introduced in the model.

\[ M_{g2} = \sqrt[\frac{k-1}{k-1}]{\frac{2 \eta_{n}}{k-1}} \left[ \left( \frac{P_{g}}{P} \right)^{\frac{k-1}{k}} - 1 \right] \]  

(2-18)

Furthermore, low values of the compressibility factor \( Z_r \) are also found at positions 5, c,i and c,i' for R600a, which are corresponding to the states after the shock, the real, and ideal processes at the diffuser exit, respectively. The reason for the low \( Z_r \) values at this state is due to the relatively high pressure of R600a compared to the other three refrigerants at the assumed condenser temperature.

Conclusively, most of the states of the ejector are at low pressure, at which the working fluid exhibits behavior close to an ideal gas. Thus the use of the ideal gas law in the proposed ejector model is acceptable if calculation of the velocity through the nozzle is done by using real gas data.

![Figure 2-3 Processes in the R245fa ejector produced from REFPROP (Lemmon et al., 2010).](image-url)
Figure 2-3 gives a view of the ejector processes with R245fa as the working fluid for Case 1. It is clearly seen that most states inside the ejector are superheated, except for the two ejector inlets which are assumed to be saturated. Another way to determine if the gas in the state is close to ideal gas behavior is to look at the lines of constant enthalpy. For an ideal gas, the isenthalpic lines are horizontal in a $T$-$s$ diagram. As can be seen in Figure 2-3, the isenthalpic lines at typical states inside the ejector are relatively flat, except the inlet of the ejector nozzle ($g_o$), this confirms that the ideal gas assumption can be used for most parts of the ejector.

2.3 Fixed-geometry ejector

Practically, two choked flows exist in the ejector. One occurs in the primary flow through the nozzle throat, the second choke is the secondary flow at the "hypothetical throat" (Munday and Bagster, 1977; Huang et al., 1999) in the mixing chamber. The appearance of choking phenomena directly depends on the ejector back pressure $P_b$. If $P_b$ is below the critical back pressure $P_b^*$, the primary flow and the secondary flow are both choked, causing a constant mass flow rate. As a result, the entrainment ratio stays constant as $P_b$ changes in this range, known as critical or double choking mode. If $P_b$ increases beyond $P_b^*$, entering into the single-choking or subcritical mode, the entrainment ratio will drop sharply. A further increase in $P_b$ will lead to the back-flow or malfunction mode, where the entrainment ratio is zero, and a reverse flow might occur. Figure 2-4 depicts the variations of the entrainment ratio with the back pressure $P_b$.

![Figure 2-4 Operational modes for a fixed-geometry ejector (Huang et al., 1999).](image)

Clearly, for a fixed-geometry ejector, the back pressure $P_b$ should be practically kept at the critical pressure $P_b^*$ to ensure the system performance...
at a high condenser pressure. A variable-geometry ejector is a promising solution to enhance the system performance by avoiding this sharp drop of the entrainment ratio when the back pressure $P_b$ is higher than the critical value. It should be reemphasized that the proposed ejector model in Paper I is to predict the ejector performance with a suitable area ratio under the given operating condition, rather than to give the performance assessment of a fixed-geometry ejector over a range of operating conditions.

2.4 Ideal ejector

It is of interest to introduce the ideal entrainment ratio, which helps to determine the thermodynamic limit of the ejector performance. It is also very important for the advanced exergy analysis in Paper IV. It has been used as a logical criterion and can be obtained by evaluating the entrainment ratio of a reversible ejector at a given operating condition (McGovern et al., 2012), as schematically shown in Figure 2-5. State “i” is the reversible ejector outlet, which is the crossover point of the line $(g,o-e,o)$ and the condenser pressure line $P_c$ in the $h$-$s$ diagram.

![Figure 2-5 Schematic interpretation of the ideal entrainment ratio (McGovern et al., 2012).](image)

2.5 Performance evaluation

2.5.1 Ejector entrainment ratio

The ejector capacity is evaluated by the entrainment ratio $\mu$, which is defined as the ratio between the secondary mass flow rate $(m_e)$ and primary mass flow rate $(m_g)$. Using all the ejector processes and combining Eqns. (2-3), (2-
6), (2-14) and (2-15) into Eq. (2-13), the expression of the entrainment ratio is given as (Paper I):

$$\mu = \frac{\dot{m}_s}{m_g} = \frac{\sqrt{2\eta_n \cdot (b_{g,o} - b_2)} - \sqrt{2(b_{c,i'} - b_4)/\eta_m \cdot \eta_d}}{\sqrt{2(b_{c,i'} - b_4)/\eta_m \cdot \eta_d} - \sqrt{2(b_{c,o} - b_0)}}$$  \hspace{1cm} (2-19)

where $b_{2'}$ and $b_{c,i'}$ are the end states of the isentropic processes in the nozzle and diffuser, respectively. Definitions of the three ejector efficiencies are found in Eqns. (2-5), (2-12) and (2-15).

For the ideal ejector (see Figure 2-5), by applying an energy balance and assuming a reversible process, as well as recalling that the velocities in the considered states have been assumed to be negligible, the following equations therefore can be obtained as (McGovern et al., 2012):

$$b_{g,o} + \mu_i \cdot b_{c,o} = (1 + \mu_i) \cdot b_i \hspace{1cm} (2-20)$$

$$s_{g,o} + \mu_i \cdot s_{c,o} = (1 + \mu_i) \cdot s_i \hspace{1cm} (2-21)$$

The ideal entrainment ratio $\mu_i$ is the secondary mass flow rate divided by the primary mass flow rate in the reversible ejector (McGovern et al., 2012):

$$\mu_i = \frac{\dot{m}_s}{m_g}$$  \hspace{1cm} (2-22)

2.5.2 System coefficient of performance

The coefficient of performance (COP) is the ratio of the refrigerating capacity produced in the evaporator to the sum of the heat supplied to the generator and the electricity consumed by the pump:

$$\text{COP} = \frac{Q_e}{Q_g + W_{PU}}$$  \hspace{1cm} (2-23)

For calculating the COP, the pump work $W_{PU}$ is typically less than 1% of the generator heat input and hence negligible (Chen et al., 2013b). Thus the COP is rewritten as:
The ERS can be considered as the combination of a power loop and a refrigeration loop, as shown in Figure 1-2(a). Therefore, the Carnot coefficient of performance ($\text{COP}_{\text{Carnot}}$) of an ERS is given as the coefficient of a Carnot engine operating between the generator temperature $T_g$ and the condenser temperature $T_c$ multiplied by the COP of a Carnot refrigerator operating between the condenser temperature $T_c$ and the evaporator temperature $T_e$, in which the temperature unit has to be expressed in Kelvin:

$$\text{COP}_{\text{Carnot}} = \frac{T_g - T_c}{T_g} \cdot \frac{T_c - T_e}{T_c - T_e}$$  \hspace{1cm} (2-25)

To evaluate the refrigeration efficiency of the ERS and reflect how much it deviates from the ideal cycle, a Carnot efficiency is defined as the ratio of the COP of the ejector cycle to the COP$_{\text{Carnot}}$ of the Carnot cycle under the same operating temperatures. It is then written as:

$$\eta_{\text{Carnot}} = \frac{\text{COP}}{\text{COP}_{\text{Carnot}}}$$  \hspace{1cm} (2-26)

### 2.5.3 Ejector area ratio

The main ejector geometry is characterized by the area ratio $A_r$, which is a very important parameter and influences the system COP (Yan and Cai, 2012). The area ratio $A_r$ is defined as the area ratio of the constant-area mixing chamber to the nozzle throat ($A_3/A_1$). For a specific condition, it is expressed by this relation (Paper I):

$$A_r = \frac{A_3}{A_1} = \frac{P_g \cdot (1 + \mu)^{1/2} \cdot (1 + \mu \cdot T_c / T_g)^{1/2} \cdot [2/(k+1)]^{1/(k-1)} \cdot [1 - 2/(k+1)]^{1/2}}{P_c \cdot (P_5 / P_c)^{1/k} \cdot [1 - (P_5 / P_c)^{(k-1)/k}]^{1/2}}$$  \hspace{1cm} (2-27)
3 Extended Theoretical Method

Based on the proposed ejector model in Paper I, different theoretical approaches are applied to study the ejector behavior and to analyze the working characteristics of the ejector refrigeration system (ERS). Paper II-V reveals various features of the ERS from different perspectives. The working fluid is essential to the ejector and the ERS operation. Paper II deals with the selection of refrigerant for the ERS. Special attention is paid to the superheat of the primary flow before entering the ejector nozzle so that the droplet formation during expansion in the ejector can be effectively eliminated. To gain a better understanding of the behavior of the ejector itself under different operating conditions, an exergy analysis is employed to study interaction and relationships between the ejector’s internal and external parameters (Paper III). Thereafter, the exergy-based investigations focusing on the whole ERS are carried out. Paper IV employs a so-called advanced exergy analysis, which is a recently developed technique, to quantify the interactions among the components of the ERS, and evaluate the realistic system improvement potential as well as to achieve more detailed working characteristics on how to thermodynamically optimize the ERS. Finally, Paper V demonstrates the application of thermoeconomics to the ERS, and how to further optimize the ERS from an economic point of view. The results of the theoretical part are summarized in the Chapter 5.

3.1 Working fluids in the ejector refrigeration system

As previously mentioned, one of the merits of the ERS is that various refrigerants can be used. The selection of an appropriate working fluid plays an indispensable role in the ERS as different refrigerants perform differently. A suitable refrigerant should not only provide good system performance, but also bring less system failures, remove or reduce other problems, like environmental issues, safety concerns, etc. Table 3-1 summarizes the working fluids found in literature that have been investigated for use in vapor ejectors in ERS.
Table 3-1 Working fluid used in the ERS.

<table>
<thead>
<tr>
<th>Pure Refrigerants</th>
<th>Halocarbon</th>
<th>Hydrocarbon</th>
<th>Others Refrigerants</th>
</tr>
</thead>
<tbody>
<tr>
<td>R11, R12, R113, R114, R123, R134a, R141b, R142b, R143a, R152a, R21, R227ea, R236fa, R245ca, R245fa, RC318, R365mfc</td>
<td>Propane (R290), Butane (R600), Isobutane (R600a)</td>
<td>Water (R718), Ammonia (R717), Dimethylether (RE170) Acetaldheyde, Methanol, Methylamine</td>
<td></td>
</tr>
<tr>
<td>Refrigerant Mixtures</td>
<td>R500, R22/RC318, R22/R142b, R22/R124, R22/R152a, R22/R134a, R134a/R142b, R152a/R142b, R134a/R152a, R32/R134a, R32/R152a, R290/R600a, R600a/R600; R600a/DME(RE170), R290/DME</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

3.1.1 Criteria of working fluid selection for the ejector refrigeration system

Pridasawas (2006) addressed some general requirements when choosing a working fluid for the refrigeration system.

- **Environmental effect:** Ozone Depletion Potential (ODP) is the relative amount of degradation to the ozone layer, in relation to R11 which is being fixed at an ODP of 1.0. Global Warming Potential (GWP) is a measure of the global warming impact relative to CO2 over a set time period (usually a hundred years). According to Montreal protocol and other international regulations, some suggested refrigerant in Table 3-1 are now banned, such as R11, R12, R113, R114, R141b, R21. The HCFCs are to be totally phased out in 2030 in developed countries and in 2040 in developing countries.

- **Safety issues:** Safety is mainly concerning the toxicity, flammability, and physical hazards. Most of the refrigerants in Table 3-1 have been identified as non-toxic, except R717. Some refrigerants are flammable, like R152a, R32, R365mfc, R290, R600, R600a and R717. In addition, the material compatibility could lead to some problems, such as electrical short-circuiting if the motor insulation is deteriorated, and leakage due to destruction of the seals. These refrigerants need extra attention regarding safety issues.

- **Economics and availability:** The working fluid should be low cost and available on the market. R134a is the most common refrigerant on market. R600a and R717 are the common refrigerants in domestic refrigerators and industrial refrigeration system, respectively. R290 is expected to be more popular in the near future. Many working fluids in Table 3-1 are presently not being widely used as refrigerants, they have other applications.
More specific considerations should be taken into account when choosing a working fluid for the ERS. According to Chunnanond and Aphornratana (2004), the desirable thermodynamic properties should have:

- a large latent heat of vaporization in order to minimize circulation rate cooling capacity per unit mass;
- a relatively high critical temperature to adapt to possible large variations in the generator temperature, but without a too high saturation pressure in the generator to avoid too robust construction of the pressure vessel and to minimize the power required by the pump. The pressure in the evaporator should not be too low and preferably should be above the ambient pressure in order to minimize the leakage of air into the system;
- favorable viscosity and thermal conductivity for good heat transfer;

More importantly, the working fluids used in the ERS have been categorized into three types: wet fluids, dry fluids and isentropic fluids, based on the slope of the vapor saturation curve of the refrigerant in a $T$-$s$ diagram (Pridasawas, 2006), as shown in Figure 3-1.

For the wet fluids, the vapor saturation curve has a negative slope in a $T$-$s$ diagram. If we consider an isentropic process in the nozzle, an isentropic expansion starting at a state of saturated vapor will fall inside the two-phase region, see Figure 3-1(a). The dry fluids have a positive slope of the vapor saturation curve, while the vapor saturation curve of the isentropic fluids is vertical. Normally, there is no phase change during an isentropic expansion for the dry and isentropic fluids, see Figure 3-1 (b) and (c). However, when the temperature is very high, close to the critical temperature, the isentropic expansion of the dry and isentropic fluids could occur inside the two phase region as well, as shown by the blue lines in Figure 3-1(b) and (c), which may lead to the droplet formation that is inherent to the wet fluid.

![Figure 3-1 T-s diagrams of three types of refrigerants: (a) the wet fluid, (b) the dry fluid and (c) the isentropic fluid (Paper II).](image-url)
3.1.2 Working fluid screening

Although a number of refrigerants can be used in the ERS as shown in Table 3-1, some of them have been prohibited due to international regulations, and some are rarely being used in refrigeration systems. Table 3-2 presents some potential working fluids with zero ODP for the ERS, including the common refrigerants and some new refrigerants.

Table 3-2 Refrigerants for the ERS (Lemmon et al., 2010, Calm and Hourahan, 2011).

<table>
<thead>
<tr>
<th>Refringerant-No</th>
<th>Molar mass</th>
<th>Physical data</th>
<th>Safety data</th>
<th>Environment</th>
<th>Type</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>NBP (°C)</td>
<td>$T_{cri}$ (°C)</td>
<td>$P_{cri}$ (kPa)</td>
<td>LFL (%)</td>
</tr>
<tr>
<td>Pure refrigerants</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1234yf</td>
<td>114.04</td>
<td>-29.5</td>
<td>94.7</td>
<td>3382</td>
<td>6.2</td>
</tr>
<tr>
<td>R1234ze</td>
<td>114.04</td>
<td>-19.0</td>
<td>109.4</td>
<td>3636</td>
<td>7.6</td>
</tr>
<tr>
<td>R1270</td>
<td>42.08</td>
<td>-47.6</td>
<td>91.1</td>
<td>4555</td>
<td>2.7</td>
</tr>
<tr>
<td>R134a</td>
<td>102.03</td>
<td>-26.1</td>
<td>101.1</td>
<td>4059</td>
<td>none</td>
</tr>
<tr>
<td>R143a</td>
<td>84.04</td>
<td>-47.2</td>
<td>72.7</td>
<td>3761</td>
<td>8.2</td>
</tr>
<tr>
<td>R152a</td>
<td>66.05</td>
<td>-24.0</td>
<td>113.3</td>
<td>4517</td>
<td>4.8</td>
</tr>
<tr>
<td>R290</td>
<td>44.10</td>
<td>-42.1</td>
<td>96.7</td>
<td>4247</td>
<td>2.1</td>
</tr>
<tr>
<td>R227ea</td>
<td>170.03</td>
<td>-16.3</td>
<td>101.8</td>
<td>2925</td>
<td>none</td>
</tr>
<tr>
<td>R236fa</td>
<td>150.04</td>
<td>-1.4</td>
<td>124.9</td>
<td>3200</td>
<td>none</td>
</tr>
<tr>
<td>R245ca</td>
<td>134.05</td>
<td>25.1</td>
<td>174.4</td>
<td>3925</td>
<td>7.1</td>
</tr>
<tr>
<td>R245fa</td>
<td>134.05</td>
<td>15.1</td>
<td>154.0</td>
<td>3650</td>
<td>none</td>
</tr>
<tr>
<td>RC318</td>
<td>200.03</td>
<td>-6.0</td>
<td>115.23</td>
<td>2778</td>
<td>none</td>
</tr>
<tr>
<td>R32</td>
<td>52.02</td>
<td>-51.7</td>
<td>78.1</td>
<td>5782</td>
<td>14.4</td>
</tr>
<tr>
<td>R600</td>
<td>58.12</td>
<td>-0.5</td>
<td>152.0</td>
<td>3796</td>
<td>2.0</td>
</tr>
<tr>
<td>R600a</td>
<td>58.12</td>
<td>-11.7</td>
<td>134.7</td>
<td>3640</td>
<td>1.6</td>
</tr>
<tr>
<td>R718</td>
<td>18.01</td>
<td>100.0</td>
<td>374.0</td>
<td>22064</td>
<td>none</td>
</tr>
<tr>
<td>R717</td>
<td>17.03</td>
<td>33.33</td>
<td>132.3</td>
<td>11333</td>
<td>16.7</td>
</tr>
<tr>
<td>Mixture refrigerant</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R407A</td>
<td>90.11</td>
<td>-49.0</td>
<td>82.3</td>
<td>4515</td>
<td>none</td>
</tr>
<tr>
<td>R407B</td>
<td>102.94</td>
<td>-51.7</td>
<td>75.0</td>
<td>4130</td>
<td>none</td>
</tr>
<tr>
<td>R407C</td>
<td>86.20</td>
<td>-46.5</td>
<td>86.0</td>
<td>4629</td>
<td>none</td>
</tr>
<tr>
<td>R410A</td>
<td>72.58</td>
<td>-51.4</td>
<td>71.4</td>
<td>4902</td>
<td>none</td>
</tr>
<tr>
<td>R410B</td>
<td>75.58</td>
<td>-51.3</td>
<td>70.8</td>
<td>4812</td>
<td>none</td>
</tr>
<tr>
<td>R430A</td>
<td>63.96</td>
<td>-27.6</td>
<td>107.0</td>
<td>4093</td>
<td>-</td>
</tr>
<tr>
<td>R436B</td>
<td>49.87</td>
<td>-33.4</td>
<td>117.4</td>
<td>4250</td>
<td>1.7</td>
</tr>
</tbody>
</table>

NBP=normal boiling point; $T_{cri}$=critical temperature; $P_{cri}$=critical pressure; LFL=lower flammability limit (% by volume in air); *=ASHRAE 34 safety group; $\tau_{atm}$=atmospheric life.
Water (R718) offers advantages of a high heat of vaporization, being inexpensive and environmentally favorable. However, it limits the evaporator temperature to be above 0 °C. Ammonia (R717) is widely used in industrial refrigeration systems and is an environmentally-friendly fluid, but it is toxic and flammable. R143a, R32, R407B, R410A and R410B have a relatively low critical temperature $T_{cri}$ (less than 80 °C), limiting their adaptability in the ERS when using high temperature heat sources in the generator. R134a, R227ea, R236fa, R245ca, R245fa, RC318, R407A and R407C have environmental concerns with values of GWP higher than 725. Although the definition of a low GWP refrigerant is still under discussion, a refrigerant with a GWP higher than 700 is not classified as a low GWP refrigerant. The remaining candidates, namely R1234yf, R1234ze, R1270, R152a, R290, R600, R600a, R430A and R436B, are all considered as low-GWP refrigerants. Unfortunately, they are flammable. Apparently, there is no “perfect” working fluid available for the ERS. None of the candidates in Table 3-2 has all the required criteria; the above requirements are only partially fulfilled. More considerations regarding the ejector behavior and system performance should be taken into account when selecting a working fluid for the ERS.

Several comparative studies have been carried out to investigate the performance of different refrigerants in the ERS. (Selvaraju and Mani, 2004; Pridasawas, 2006; Roman and Hernandez, 2011; Varga et al., 2013; Dahmani et al., 2011). No agreement has been reached in terms of which refrigerant has the best performance. Paper II has also compared different refrigerants in the ERS with focus on the required superheat of the primary flow before entering the ejector nozzle to eliminate droplet formation.

### 3.1.3 Superheat for the primary flow before entering ejector nozzle

As stated, droplets form during the expansion through the ejector nozzle for wet fluids. The dry and isentropic fluids also have possibilities to form droplets during that process. The droplets may block the effective mixing area, and may seriously affect the flow processes in the ejector and the performance of the ejector itself (Huang et al., 1999). More importantly, it may cause periodically oscillating unsteady flow in the ejector and unstable system operation (Grazzini et al., 2011). It is possible to eliminate the droplets by superheating the primary flow before entering the ejector nozzle. However, it is unclear how much superheat for the primary flow that is appropriate. The problems with droplet formation will remain if the superheat is too small. On the contrary, supplying unnecessary superheat is
a waste of energy. A minimum superheat for the primary flow ($\Delta T_{\text{min}}$) was proposed in Paper II to ensure the elimination of droplet formation inside the ejector and to maximize the system performance.

Figure 3-2(a) shows the isentropic expansion of a wet fluid in the nozzle when the primary flow entering the nozzle is saturated (dashed red line). Introducing an isentropic efficiency $\eta_n$ (solid red line) could, somewhat, decreases the problems of droplet formation. The superheat can explicitly be described by moving the expansion line from the two-phase region to the superheated region. The minimum superheat $\Delta T_{\text{min}}$ is obtained when the end of the expansion process, i.e. state 2, is located on the vapor saturation curve (solid blue lines). The $\Delta T_{\text{min}}$ depends not only on the generator saturation temperature $T_g$ but also on the ejector nozzle efficiency $\eta_n$, as shown in Figure 3-2(b) and (c), respectively. Using the same method, the $\Delta T_{\text{min}}$ of dry and isentropic fluids can be obtained if superheat is required.

![Figure 3-2 Schematic interpretation of the minimum superheat for the wet fluid in T-s diagram (reproduced from Paper II).](image)

Four wet fluids (R134a, R152a, R290 and R430A), four dry fluids (R245fa, R600, R600a and R1234ze) and one isentropic fluid (R436B) are selected and their performance and feasibility in the ERS are compared based on the determined $\Delta T_{\text{min}}$ for each refrigerant. These results are presented in Paper II.

### 3.2 Exergy analysis of the ejector refrigeration system

The first law of thermodynamics, expressing conservation of energy, is one of the most fundamental relationships in energy analysis. However, it does not always provide a full assessment. A typical example for this is the process through a throttling valve, which is normally assumed to be isenthalpic and no energy loss is found. However, it is recognized that the
lower pressure after the throttling process reduces the usable work potential. Apparently, the energy analysis alone is inadequate for depicting some important aspects of energy systems.

Exergy analysis, based on the second law of thermodynamics, provides a true measure of the actual performance and is useful to identify the causes, locations, and magnitudes of process inefficiencies. Exergy is defined as the maximum amount of work that can be produced by a process or a system when it comes to equilibrium with a reference environment (Dincer and Rosen, 2013). Exergy analysis acknowledges that, although energy cannot be created or destroyed, it can be degraded in quality, which is expressed as exergy destruction (due to irreversibilities within the system component) and exergy loss (exergy transfer to the environment).

To calculate exergy magnitudes, it is necessary to define an idealized model for the reference environment, which should be in stable equilibrium and acts as an infinite system for the heat sink and heat source as well as materials (Dincer and Rosen, 2013). The magnitude of exergy depends on the states of the system and the reference environment that in Paper III-V has been assumed as $T_{\text{ref}}=25$ °C and $P_{\text{ref}}=101.32$ kPa. It should be noted that when the reference temperature $T_{\text{ref}}$ is involved in the calculation of exergy, it has to be expressed in Kelvin.

### 3.2.1 Exergy

In the absence of magnetic, electrical, nuclear and surface-tension effects, the exergy, symbolized as $\dot{E}$, consists of physical exergy $\dot{E}^{\text{PH}}$, chemical exergy $\dot{E}^{\text{CH}}$, kinetic exergy $\dot{E}^{\text{KN}}$ and potential exergy $\dot{E}^{\text{PT}}$ (Bejan et al., 1996).

$$
\dot{E} = \dot{E}^{\text{PH}} + \dot{E}^{\text{CH}} + \dot{E}^{\text{KN}} + \dot{E}^{\text{PT}} = \dot{m}(\epsilon^{\text{PH}} + \epsilon^{\text{CH}} + \epsilon^{\text{KN}} + \epsilon^{\text{PT}})
$$

(3-1)

where $\epsilon$ is the specific exergy on a mass basis.

The specific physical exergy $\epsilon^{\text{PH}}$ is the maximum useful work obtained by passing the unit of mass of a substance of the generic state $(P, T)$ to the environmental state $(P_{\text{ref}}, T_{\text{ref}})$ through purely physical processes (Querol et al., 2013).

$$
\epsilon^{\text{PH}} = [h(P, T) - h(P_{\text{ref}}, T_{\text{ref}})] - T_{\text{ref}}[\sigma(P, T) - \sigma(P_{\text{ref}}, T_{\text{ref}})]
$$

(3-2)
It could be further split into thermal exergy $e^T$, which is mainly due to temperature, and mechanical exergy $e^M$, which is mainly due to the pressure (Bejan et al., 1996).

$$ e^{PH} = e^T + e^M $$

(3-3)

with the thermal exergy $e^T$ and mechanical exergy $e^M$ expressed as:

$$ e^T = [h(P, T) - h(P, T_{ref})] - T_{ref} [s(P, T) - s(P, T_{ref})] $$

(3-4)

$$ e^M = [h(P, T_{ref}) - h(P_{ref}, T_{ref})] - T_{ref} [s(P, T_{ref}) - s(P_{ref}, T_{ref})] $$

(3-5)

The specific kinetic exergy $e^{KN}$ and specific potential exergy $e^{PT}$ are given as (Bejan et al., 1996):

$$ e^{KN} = \frac{u^2}{2} $$

(3-6)

$$ e^{PT} = g \zeta $$

(3-7)

where the $u$ and $\zeta$ denote the velocity and the relative altitude, respectively.

Chemical exergy is the maximum useful work which would be attained by passing from the environmental state to the dead state, by means of chemical processes involving heat transfer and exchange of substances only with the environment, when the stream composition is not in chemical equilibrium with the environment (Querol et al., 2013). The chemical exergy is not considered in the ERS because chemical reactions are not expected. It thus will not be discussed.

3.2.2 Conventional exergy analysis of the ejector

For the ejector, the exergy destruction in the nozzle ($\dot{E}_{D,n}$), mixing chamber ($\dot{E}_{D,m}$) and diffuser ($\dot{E}_{D,d}$) are given as below, with the nomenclature corresponding to Figure 2-2:

$$ \dot{E}_{D,EJ} = T_{ref} [\dot{m}_c \cdot \Delta s_{c,i} - \dot{m}_g \cdot \Delta s_{g,o} - \dot{m}_e \cdot \Delta s_{e,o}] $$

(3-8)

$$ \dot{E}_{D,n} = T_{ref} \dot{m}_g \cdot (\Delta s_{g,o}) $$

(3-9)

$$ \dot{E}_{D,d} = T_{ref} \dot{m}_e \cdot (\Delta s_{e,i} - \Delta s_4) $$

(3-10)

$$ \dot{E}_{D,m} = \dot{E}_{D,EJ} - \dot{E}_{D,n} - \dot{E}_{D,d} $$

(3-11)
The exergy destruction associated with the normal shock is calculated as:

$$\dot{E}_{D,\text{sh}} = T_{\text{ref}} \dot{m}_c (s_5 - s_4)$$  \hspace{1cm} (3-12)

It should be noted that the exergy destruction of the shock process $\dot{E}_{D,\text{sh}}$ is integrated into the exergy destruction of the diffuser $\dot{E}_{D,d}$. In other words, the exergy destruction in the diffuser is due to the shock, and frictional losses, flow separation, etc. In Paper III, the exergy destruction $\dot{E}_D$ has been synonymously referred to as irreversibility, symbolized by $I$.

![Figure 3-3 Interactions of various ejector parameters (reproduced from Paper III).](image)

The ejector performance and the exergy destruction are influenced by many parameters, e.g. the ejector geometries, the operating conditions, and the selected refrigerant. Moreover, these parameters are closely related and interacting, as shown in Figure 3-3. To design an ejector, the operating temperatures have to be defined based on the applied heat source, heat sink and the cooling purpose, and the ejector efficiencies have to be carefully determined since they influence the calculated ejector area ratio. To evaluate the ejector performance, the ejector geometrical features determine the optimum operating conditions, and influence the ejector efficiencies which also depend on the operating conditions. All external, internal and geometrical parameters eventually impact the ejector performance and the exergy destructions as well as the distribution of the exergy destructions in the ejector. The ejector external and internal parameters are parametrically studied in Paper III.
It should be noted that the mixing efficiency $\eta_m$ is a kinetic energy ratio. Moreover, the process in the mixing chamber is rather complicated; involving supersonic flow, strong flow interactions, turbulent mixing etc. Thus the mixing efficiency might not represent all the exergy destruction of the mixing process. It was defined in this manner for the sake of simplifying the ejector modeling.

3.2.3 Conventional exergy analysis of the ejector refrigeration system

Regarding the exergy analysis for the entire ERS, the concept “fuel-product” is introduced. The exergy balance at steady-state conditions can be written as the following by using proper definitions of the exergy of fuel $\dot{E}_{F,k}$ and the exergy of product $\dot{E}_{P,k}$ (Bejan et al., 1996):

$$\dot{E}_{F,k} = \dot{E}_{P,k} + \dot{E}_{D,k}$$

(3-13)

where the subscript $k$ indicates the $k$th component in the energy conversion system. The exergy of product $\dot{E}_{P,k}$ is the desired result (expressed in exergy terms) achieved by the $k$th component being considered, and the exergy of fuel $\dot{E}_{F,k}$ represents the exergy resources consumed in the $k$th component to generate the exergy of product. It should be noted that the $\dot{E}_{F,k}$ is not necessarily restricted to being an actual fuel such as coal, natural gas, or oil, and the $\dot{E}_{P,k}$ is not specified as the power or cooling effect, either. The concept “fuel-product” is in a general sense of the exergy resource and desired results.

The exergy loss is the transferred exergy to the environment that is not further being used in any systems. The exergy loss ($\dot{E}_{L}$) appears only at the level of the total system, for which the exergy balance becomes (Bejan et al., 1996):

$$\dot{E}_{F,\text{tot}} = \dot{E}_{P,\text{tot}} + \sum \dot{E}_{D,k} + \dot{E}_{L,\text{tot}}$$

(3-14)

Two indexes are used to evaluate the component and the overall system from the exergetic point of view. One is the exergy efficiency $\epsilon$, defined as the ratio between the exergy of product and the exergy of fuel; the other index is the exergy destruction ratio $y$, defined as the exergy destruction within the $k$th component or the overall system divided by the total exergy of fuel for the overall system. They are written as:
\[ \varepsilon_k = \frac{\dot{E}_{p,k}}{\dot{E}_{F,k}} \]  
\[ \varepsilon_{\text{tot}} = \frac{\dot{E}_{P,\text{tot}}}{\dot{E}_{F,\text{tot}}} \]  
\[ \gamma_k = \frac{\dot{E}_{D,k}}{\dot{E}_{F,\text{tot}}} \]  
\[ \gamma_{\text{tot}} = \frac{\dot{E}_{D,\text{tot}}}{\dot{E}_{F,\text{tot}}} \]  

The \( k \)th component in the ERS is the generator (GE), the condenser (CO), the evaporator (EV), the ejector (EJ), the pump (PU) or the throttling valve (TV), which is consistent with those abbreviations in Paper IV and V.

### 3.2.4 Advanced exergy analysis of the ejector refrigeration system

The advanced exergy analysis is based on the conventional exergy analysis, and splits the exergy destruction into endogenous/exogenous parts and avoidable/unavoidable parts to gain more detailed insights. Therefore, more comprehensive, practical and not just rigorous information on how and to what extent the components can be improved is provided with much higher accuracy than the conventional exergy analysis.

The split parts of exergy destruction within the \( k \)th component are explained as follows (Morosuk and Tsatsaronis 2008, 2009):

- **Endogenous exergy destruction** \( \dot{E}_{D,k}^{\text{EN}} \): Relates only to the \( k \)th component’s own irreversibility and is irrelevant to the irreversibilities in the remaining system components. It is obtained when all other components operate ideally and the \( k \)th component under consideration operates with the real efficiency.

- **Exogenous exergy destruction** \( \dot{E}_{D,k}^{\text{EX}} \): This is the remaining part of the exergy destruction in the \( k \)th component found by excluding the endogenous part. It is the exergy destruction imposed on the \( k \)th component but caused by irreversibilities in the remaining system components.

- **Unavoidable exergy destruction** \( \dot{E}_{D,k}^{\text{UN}} \): Cannot be eliminated even if the best available technology would be applied, and always exists as long as the \( k \)th component is being used. Irreversibilities are due to technical limitations (Morosuk and Tsatsaronis, 2009).

- **Avoidable exergy destruction** \( \dot{E}_{D,k}^{\text{AV}} \): This is the difference between the exergy destruction and the unavoidable part in the \( k \)th component, and is recoverable. This part represents the real potential for improving the system component, thus it should be paid more attention to.
Splitting the exergy destruction in the $k^{th}$ component $\dot{E}_{D,k}$ into endogenous $\dot{E}_{D,k}^{EN}$ and exogenous $\dot{E}_{D,k}^{EX}$ parts shows the interactions among different components in the system. This is very useful to decide whether the improvement should be focused on the $k^{th}$ component being considered or on the remaining system components (Morosuk and Tsatsaronis, 2008).

The exergy destruction in the $k^{th}$ component $\dot{E}_{D,k}$ can be rewritten as:

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{EN} + \dot{E}_{D,k}^{EX}$$  \hspace{1cm} (3-19)

Generally, the endogenous exergy destruction $\dot{E}_{D,k}^{EN}$ can be decreased through improving the $k^{th}$ component itself, which also results in the decrease in the exogenous exergy destruction within the remaining system components. As a consequence, the exogenous exergy destruction in the $k^{th}$ component, i.e. $\dot{E}_{D,k}^{EX}$, is also reduced through the improvement of the remaining system components caused by improvement of the $k^{th}$ component. In other words, an improvement in the $k^{th}$ component not only leads to a reduction of the exergy destruction in the $k^{th}$ component itself, but also promotes a decrease in the exergy destruction within other components.

Splitting the exergy destruction in the $k^{th}$ component $\dot{E}_{D,k}$ into unavoidable $\dot{E}_{D,k}^{UN}$ and avoidable $\dot{E}_{D,k}^{AV}$ parts provides a realistic measure of the potential of improvement for the component being considered, which is given as:

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{UN} + \dot{E}_{D,k}^{AV}$$  \hspace{1cm} (3-20)

The avoidable exergy destruction, $\dot{E}_{D,k}^{AV}$, can be avoided with structural modifications, and efficiency improvements of individual components (Gungor et al., 2013). The unavoidable exergy destruction $\dot{E}_{D,k}^{UN}$ is determined by appropriately selecting the most important thermodynamic parameters of each component that represent only the unavoidable exergy destruction.

These two approaches of splitting the exergy destruction can be combined to produce new terms of interest and provide more detailed information. The combined parts of exergy destruction in the $k^{th}$ component are presented as (Morosuk and Tsatsaronis 2008, 2009):
• **Unavoidable endogenous exergy destruction** $\hat{E}_{\text{UN,EN}}^{\text{D,k}}$: Cannot be reduced due to technical limitations in the $k^{\text{th}}$ component.

• **Unavoidable exogenous exergy destruction** $\hat{E}_{\text{UN,EX}}^{\text{D,k}}$: Cannot be reduced due to technical limitations in the remaining components of the overall system.

• **Avoidable endogenous exergy destruction** $\hat{E}_{\text{AV,EN}}^{\text{D,k}}$: Can be reduced through improving the efficiency of the $k^{\text{th}}$ component.

• **Avoidable exogenous exergy destruction** $\hat{E}_{\text{AV,EX}}^{\text{D,k}}$: Can be reduced by a structural improvement of the overall system or by improving the efficiency of the remaining system components.

Therefore, the exergy destruction in the $k^{\text{th}}$ component $\hat{E}_{\text{D,k}}$ is alternatively written as:

$$\hat{E}_{\text{D,k}} = \hat{E}_{\text{UN,EN}}^{\text{D,k}} + \hat{E}_{\text{UN,EX}}^{\text{D,k}} + \hat{E}_{\text{AV,EN}}^{\text{D,k}} + \hat{E}_{\text{AV,EX}}^{\text{D,k}} \quad (3-21)$$

It is emphasized that the efforts to improve the system component should be focused on the avoidable endogenous $\hat{E}_{\text{AV,EN}}^{\text{D,k}}$ and the avoidable exogenous parts $\hat{E}_{\text{AV,EX}}^{\text{D,k}}$.

The options of splitting the exergy destruction in the $k^{\text{th}}$ component $\hat{E}_{\text{D,k}}$ are shown in Figure 3-4.

![Figure 3-4 Splitting the exergy destruction within the $k^{\text{th}}$ component (Paper IV).](image)

To quantify all the parts of the exergy destruction in the ERS, the sources of the exergy destruction within each system component and their corresponding representative parameters need to be identified. The exergy destruction within the three heat exchangers (the generator, the condenser and the evaporator) are due to heat transfer through a finite temperature
difference, and they are expressed as the pinch temperature differences $\Delta T_{GE}$, $\Delta T_{CO}$ and $\Delta T_{EV}$. The exergy destruction in the ejector is quite complex, involving e.g. friction, mixing loss, shock loss, and is expressed in terms of efficiencies in the ejector. These efficiencies are the nozzle efficiency $\eta_n$, the mixing efficiency $\eta_m$ and the diffuser efficiency $\eta_d$ defined in Eqns. (2-5), (2-12) and (2-15). The pump isentropic efficiency $\eta_{PU}$ denotes the exergy destruction in the pump. The exergy destruction in the throttling valve is caused by the reduction in pressure and temperature from the condenser level to the evaporator level.

With the knowledge of the sources of the exergy destruction in each component of the ERS, several thermodynamic cycles need to be created to calculate all exergy destruction parts in Figure 3-5, following the approaches proposed by Morosuk and Tsatsaronis (2008, 2009):

- **Real cycle**: All the components in the ERS are working at the designed or assumed efficiencies, namely, $\Delta T_{GE}$, $\Delta T_{CO}$, $\Delta T_{EV}$, $\eta_n$, $\eta_m$, $\eta_d$, $\eta_{PU}$ and an isenthalpic process in the throttling valve. This cycle is also used in the conventional exergy analysis to calculate the exergy destruction in each component. It is shown as the black lines (3-4-1 and 3-5-6→2-3) in Figure 3-5(a).

- **Ideal cycle**: All the components in the ERS are working ideally, leading to zero or a minimum exergy destruction ($\dot{E}_{D,k}=0$ or $\dot{E}_{D,k}=\text{min}$). The pinch temperature differences, i.e. $\Delta T_{GE}$, $\Delta T_{CO}$ and $\Delta T_{EV}$, in the three heat exchangers are assumed to be zero. In the ideal cycle, the ejector is reversible, as shown in Figure 2-5. The process of the pump is isentropic, i.e. $\eta_{PU}=1$, and an isentropic process also occurs in the throttling valve. The ideal cycle is shown as the red lines (3i-4i-1i and 3i-5i-6i→2i-3i) in Figure 3-5(b).

- **Hybrid cycles**: These cycles are created to calculate the endogenous exergy destruction in the $k^{th}$ component $\dot{E}_{D,k}^{EN}$. For a hybrid cycle, the process in the $k^{th}$ component has the same efficiency as in the real cycle. The processes in all other remaining system components are ideal corresponding to those in the ideal cycle. For example, to estimate the endogenous exergy destruction in the ejector, $\dot{E}_{D,EJ}^{EN}$, the ejector works with the real efficiency $\eta_n$, $\eta_m$ and $\eta_d$. The remaining components are considered as ideal processes. The three heat exchangers have pinch temperature differences $\Delta T_{GE}$, $\Delta T_{CO}$ and $\Delta T_{EV}$ equal to zero, the pump has an ideal efficiency $\eta_{PU}=1$, and the throttling valve is isentropic. The hybrid cycle for $\dot{E}_{D,EJ}^{EN}$ is shown as the lines 3i-4i-1i and
$3i-5i-6i \rightarrow 2^{****-3i}$ in Figure 3-5(c). The number of hybrid cycles needed is equal to the number of components in the investigated energy conversion system. Therefore, six hybrid cycles need to be created to calculate the endogenous exergy destruction $\dot{E}_{EN,D,k}$ in the ERS. The other five hybrid cycles are found in Paper IV.

- **Unavoidable cycle:** It is used to calculate the unavoidable exergy destruction $\dot{E}_{UN,D,k}$ by considering the technological limitations in all system components. This cycle consists of only unavoidable exergy destruction, which is shown as the blue lines (3UN-4UN-1UN and 3UN-5UN-6UN→2UN-3UN) in Figure 3-5(d). Since it is very difficult to quantify these limitations and determine the precise parameters that represent unavoidable constraints, the selection of values for these parameters could be arbitrary. Taking the ejector as an example, the exergy destruction in the ejector can only be reduced but not totally eliminated (Arbel et al., 2003). Moreover, the ejector efficiencies ($\eta_n, \eta_m, \eta_d$), depend not only on manufacturing techniques (Huang et al., 1999), but also on the operating conditions, as discussed in Paper III. No data concerning practical upper limits for $\eta_n, \eta_m$ and $\eta_d$ have been reported.

![Figure 3-5 Various cycles in the advanced exergy analysis: (a) the real cycle, (b) the ideal cycle (c) the hybrid cycle for the endogenous exergy destruction within the ejector, (d) the unavoidable cycle (reproduced from Paper IV).](image-url)
The combinations of splitting exergy destructions are obtainable by using the endogenous exergy destruction and the unavoidable cycle stated above. With the information of all parts of the exergy destruction in each component in the ERS, detailed knowledge of the system’s working characteristics is produced. This information is helpful in determining the directions for system optimization, as conducted in Paper IV.

3.3 Thermoeconomic optimization of the ejector refrigeration system

The aforementioned analyses are dedicated to energetic and exergetic aspects of the ERS, the thermodynamic irreversibilities, i.e. exergy destruction and exergy loss, have been parametrically evaluated. Rationally, the exergy destruction and exergy loss play very important roles in the investment costs of the energy conversion system. Knowledge of the thermodynamic performance and the cost information are also very important for system design and operation, and is significant to improve the cost effectiveness. This section extends the investigations on the ERS from a thermoeconomic perspective.

3.3.1 Introduction of thermoeconomics

As introduced in Tsatsaronis (1993) and Tsatsaronis and Cziesla (2002), thermoeconomics, also termed as exergoeconomics, is a branch of thermal sciences that combines the exergy analysis with economic principles to provide information of the energy conversion system, which is not available through sole thermodynamic analysis or economic evaluation. The most characterizing element of thermoeconomic analysis is the assignment of all costs to the exergy destruction and exergy loss. In other words, the exergy, instead of energy, is the only consistent measure of economic value. A thermoeconomic analysis can calculate not only the exergy destruction within each system component and the exergy loss in the system, but also the costs associated with all the materials and streams of the exergy carriers in the system. Therefore, it reveals the information about:

- cost associated with the exergy destruction and exergy losses;
- cost of each product produced by the system;
- optimization of the design and/or operation of the system;
- trade-off between thermodynamic performance and economic feature.
The principles and methodologies of thermoeconomic analysis have been well introduced by Bejan et al. (1996). The thermoeconomic analysis and optimization of an energy conversion system can be illustrated as in Figure 3-6.

The thermodynamic analysis includes an energy analysis and an exergy analysis. With the operating conditions of the energy conversion system and defined reference conditions, the energy and exergy analyses can be easily and straightforwardly conducted, referring to Chapter 3.2. The energy and exergy efficiencies are calculated.

Figure 3-6 Illustration for thermoeconomic analysis and optimization.
The economic evaluation is done to estimate the major costs, such as total capital investment, operating and maintenance expenses, fuel costs, based on the economic model that considers various assumptions involving the economics, technology, and environment. Moreover, the costs are expressed as functions of thermodynamic variables, and are obtained by statistical correlations between costs and the main thermodynamic parameters of the system component performed on the real data series (Bejan et al., 1996).

The thermoeconomic analysis is carried out by means of assigning all the costs to the exergy streams. A cost balance is applied to the \( k \)th system component. That is the sum of cost rates associated with all exiting exergy streams is set equal to the sum of cost rates of all entering exergy streams plus the appropriate charges.

\[
\sum (\dot{C}_j)_{k,in} + \dot{Z}_k = \sum (\dot{C}_j)_{k,out}
\]  

(3-22)

with more details:

\[
\sum (\dot{m}_j \cdot \dot{e}_j)_{k,in} + \dot{Z}_k^{CI} + \dot{Z}_k^{OM} = \sum (\dot{m}_j \cdot \dot{e}_j)_{k,out}
\]  

(3-23)

where \( \dot{C}_j \), \( \dot{e}_j \) and \( \dot{e}_j \) are the cost rate, the unit cost for exergy and specific exergy flow for the \( j \)th stream to/from the component, respectively. The term \( \dot{Z}_k \) represents the cost rate associated with the capital investment \( \dot{Z}_k^{CI} \) and the operating and maintenance expense \( \dot{Z}_k^{OM} \) of the \( k \)th component, which is obtained from the economic evaluation. The thermoeconomic performance is evaluated.

To carry out the thermoeconomic optimization, the thermoeconomic objective function (OBF) to achieve the required purposes has to be defined. The OBF can be single or multi objectives. Then the most important parameters are defined as the decision variables that can be changed in the optimization process. Moreover, the energy conversion system is imposed under several boundary conditions associated with available materials, financial resources, and environmental regulations, together with the safety, operability, reliability, availability, and maintainability of the system (Bejan et al., 1996), so the boundary conditions has to be clearly stated. Finally, appropriate techniques, e.g. iteration, genetic algorithm, are employed to search the required OBF and facilitate the process of mathematically
optimizing the system. The optimum results are very useful for cost-effective system design and operation.

With all the information mentioned above, the balances existing in the \( k \)th component at steady state can be schematically summarized as in Figure 3-7, including mass conservation, energy conservation, exergy balance Eq. (3-13) and cost balance Eq. (3-23), which are imposed under certain boundary conditions.

![Figure 3-7 Schematic interpretation of various balances in the \( k \)th component.](image)

### 3.3.2 Application of thermoeconomics in an ejector refrigeration system

Following the procedure in Figure 3-6, a thermoeconomic optimization of an ERS is conducted in Paper V. The inlet and outlet temperature of the heat source fluids through the generator and evaporator, and the heat sink fluid through the condenser, which are generalized as brine side fluids in Paper V, are assumed to be fixed, thus their flow rates are varied to adapt to different heat loads in the generator, the condenser and the evaporator. The pinch temperature differences in the three heat exchangers (\( \Delta T_{GE} \), \( \Delta T_{CO} \) and \( \Delta T_{EV} \)) are selected as the decision variables. Moreover, the economic conditions are imposed for the thermoeconomic analysis of this ERS. The objective function (OBF) is defined as the total cost including the cost rate of all brine side fluids and the electricity cost rate for the pump as well as the cost rate related to the capital investment and operating and maintenance expense. The goal of the optimization is to minimize the OBF by changing the decision variables. Figure 3-8 schematically describes the ERS being considered with the boundary conditions.
Two economic scenarios are considered, the steam, water, electricity are charged at specific prices at scenario I, while scenario II only takes the electricity cost into account. The two economic scenarios are applied to the ERS with a preliminary design condition, referred to as the base case, and the optimization is carried out individually in Paper V.
4 Experimental Method

As the key component in the ejector refrigeration system (ERS), the ejector behavior and performance are significantly related to its geometrical parameters. To investigate the effects of ejector geometrical parameters, a test bench is built. Detailed description of the experimental facilities and uncertainty analysis are presented in this chapter. The results from the experimental investigation are presented in Chapter 6.

4.1 Working fluid selection for experiment

The working fluid is crucial in the ejector design and has to be chosen firstly. Various refrigerants can be used in the ERS. The dry fluids generally have better performance compared to the wet fluids at the same operating conditions, and R245fa and R600 have a relatively high COP in the ERS, according to Paper II. R245fa has been widely used in ERS experiments due to its favorable thermodynamic properties, reasonable working pressures and a high critical temperature. Moreover, being non-corrosive, non-toxic and non-flammable as well as being a dry fluid makes it a very good candidate for the ERS. The concern of using R245fa is the relatively high Global Warming Potential (GWP) of 1050 (Calm and Hourahan, 2011). Although R600 is a natural refrigerant with little environmental concerns, it is flammable. Another natural refrigerant, R600a, has the same issue. Their properties can be found in Table 3-2. R600a and R600 are excluded due to the large required amount of refrigerant in the test bench. For safety reasons, R245fa is selected as the working fluid in the experiment in the lab.

4.2 Ejector design and geometries

After the refrigerant is chosen, the next task is to design the ejectors. The design of a suitable ejector is the most complicated task in this experimental section. The ejector is normally designed based on the theoretical expressions of gas dynamics together with experimental data. The main geometrical parameter, which is the area ratio $Ar$, referring to Eq. (2-27), is established. However, for other parameters, such as the convergent and divergent angles, lengths of the constant area mixing chamber and the diffuser, there are no clear design guidelines. Empirical correlations of steam ejector design are quite well known and can be found in several handbooks,
e.g. ASHRAE (1983). The detailed design of an ejector working with other fluids is rarely reported.

The ejectors working with R245fa in the experiment are designed based on the proposed model in Paper I, and the handbook (ASHRAE, 1983) as well as literature (Eames et al., 2007; Yapıcı et al., 2008). Six nozzles are made to, on one hand, obtain different area ratios, and on the other hand, accommodate to the compressor capacity. The nozzles are interchangeable. Therefore, there are six ejectors (E1 through E6), consistent with the nozzle numbers as shown in Figure 4-1. The nozzles and the ejector main body are schematically shown in Figure 4-2. The important geometries are listed in Table 4-1. The ejector main body and nozzle 4 is made from stainless steel, while the remaining nozzles are made from brass. The nozzle and ejector body are sealed by Teflon washers. The ejectors are connected to copper pipes by flare fittings.

![Figure 4-1 Picture of the nozzles and the ejector body.](image-url)
Table 4-1 Summary of the ejector geometrical details.

<table>
<thead>
<tr>
<th>Ejectors</th>
<th>Diameters (mm)</th>
<th>Angles (°)</th>
<th>Lengths (mm)</th>
<th>Ar</th>
<th>Distance (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>D₁</td>
<td>D₂</td>
<td>D₃</td>
<td>D₄</td>
<td>D₅</td>
</tr>
<tr>
<td>E₁</td>
<td>2.014</td>
<td>2.8</td>
<td>6.3</td>
<td>5.2</td>
<td>10.0</td>
</tr>
<tr>
<td>E₂</td>
<td>2.008</td>
<td>3.2</td>
<td>6.3</td>
<td>5.2</td>
<td>10.0</td>
</tr>
<tr>
<td>E₃</td>
<td>2.021</td>
<td>3.8</td>
<td>6.3</td>
<td>6.2</td>
<td>10.0</td>
</tr>
<tr>
<td>E₄</td>
<td>2.207</td>
<td>5.0</td>
<td>6.3</td>
<td>7.5</td>
<td>12.0</td>
</tr>
<tr>
<td>E₅</td>
<td>2.243</td>
<td>4.0</td>
<td>6.3</td>
<td>6.0</td>
<td>10.0</td>
</tr>
<tr>
<td>E₆</td>
<td>2.504</td>
<td>4.4</td>
<td>6.3</td>
<td>6.5</td>
<td>10.0</td>
</tr>
</tbody>
</table>

4.3 Description of the test bench

The ejector is exposed to three pressure levels in the ERS. It is driven by the high pressure and temperature vapor to entrain the low pressure vapor, and discharges the mixed vapor at an intermediate pressure. A mechanical pump is used to deliver the refrigerant from the condenser to the generator, and lift the condenser pressure to the generator pressure. However, in practical application of the ERS, finding a pump for circulating liquid refrigerant,
such as R245fa, is no easy task with commercially available hardware due to the high pressure difference and small flow rate across the pump. Moreover, leakage, cavitation, and material compatibility problems related to the liquid pump need extra concerns (Srisastra et al., 2008). After carrying out many inquiries to different manufacturers, a proper liquid pump to meet the requirements has not been found.

The experiments are from the beginning aimed to test the ejector performance. It is therefore not necessary to build a complete ERS test bench. Due to the unavailability of the pump, an alternative method to achieve the high pressure has been used. The closed cycle test bench without a pump is schematically shown in Figure 4-3, which is designed to achieve the different pressure levels for the ejector operation. A compressor (Mitsubishi, AH60V, scroll type) is used to fulfill the functions of both the mechanical pump and the generator in the ERS, and produces the required high pressure and high temperature vapor to drive the ejector. An oil separator is installed after the compressor to eliminate the effects of oil on the ejector operation and on the measurements. A bypass loop, including a control valve (CV5, stainless steel needle valve) and the gas cooler 2 (a plate heat exchanger), is designed to handle the excessive flow rate for the ejector if necessary. A pressure relief valve with an opening pressure of 18 bar is used before CV5 as a precaution. The flow out from the oil separator is cooperatively managed by another control valve (CV4) and the bypass loop. The mass flow rate of the ejector primary flow \( \dot{m}_p \) is directly measured by a coriolis flow meter before entering the ejector nozzle. The ejector is installed vertically after several revisions and upgrades of the test bench. The flow after the ejector passes through another coriolis flow meter to measure the mixed flow rate \( \dot{m}_c \). After that, the flow is divided into two paths using two control valves (CV1 and CV2) in order to regulate the pressures. The control valve (CV3) is added between the two paths to increase the flexibility in regulating the pressures. One path of the flow goes through the condenser (a plate heat exchanger) cooled by city water, then passes through three needle valves arranged in parallel acting as a throttling device, and goes through the evaporator (a plate heat exchanger) which is heated by a thermostatic bath. The flow is finally entrained into the ejector. The rest of the flow returns to the compressor. In the suction line of the compressor, the flow goes through cooler 1 (a plate heat exchanger) cooled by city water to prevent an unnecessarily high temperature at the compressor outlet. The flow then passes through a check valve, and a filter and collects in a receiver which is needed to ensure stable compressor operation. Two sight glasses are used before the two coriolis flow meters and one is after the receiver, to
be able to monitor the flow conditions. Several on/off valves (ball valves) are installed to facilitate change of components and maintenance, as well as service. All control valves (CV) are needle type and used to reduce the pressure. All the pipe lines, heat exchangers and the ejector are covered by insulations to reduce heat dissipations.

The pressures at the ejector inlets and outlet are very important. They are measured by three pressure transmitters that are closely located to the ejector’s two inlets and outlet to ensure accurate measurement. They are named $P_s$, $P_e$ and $P_c$, respectively, consistent with Paper I-III. Pressure gauges ($P_1$-$P_4$) are used to indicate the pressure levels at the compressor outlet and after the control valves, and to monitor the system operation. The temperatures are measured by thermocouples, including 17 temperatures in Figure 4-3 and one more to measure the surrounding temperature. The most important temperatures are at the ejector inlets ($T_p$ and $T_s$) and outlet ($T_{out}$).

![Figure 4-3 Schematic description of the test bench.](image)

The layout of the test bench for R245fa circulation is shown in Figure 4-4. It was designed to assist the construction of test bench. Figure 4-5 shows a picture of the test bench built in the lab.
Figure 4-4 Configuration and design for the R245fa loop in the test bench.

Figure 4-5 Photograph of the test facility.
4.4 Experimental procedure

The test bench is used to test the ejector entrainment ratio under different operating conditions. The measurement devices are connected to a data acquisition system to obtain instantaneous monitoring. All readings, including two mass flow rates, three pressures, nineteen temperatures (eighteen thermocouples and one from the coriolis flow meter before the ejector), are recorded automatically every five seconds and stored in an excel file, and monitored by a computer.

Before start-up, all the valves are open except CV5 (see Figure 4-3). The data acquisition system is turned on to check if the pressures indicated by the pressure transmitters are corresponding to the saturation pressure of the room temperature with the assistance of REFPROP (Lemmon et al., 2010). Then the compressor is switched on. After 10-20 minutes the city water is connected with a small initial flow rate in order to get a relatively high pressure for the primary flow. All measured parameters must be stable for at least 25 minutes before data is acquired, and data are acquired during 25 minutes at 5 seconds intervals and stored in an excel file automatically. After that, another adjustment is made to get the next data point. The adjustment includes changing of the openings of CV1-CV3 and throttling valves, the temperatures of the thermostatic bath and the flow rate of city water through the condenser and cooler 1. Before shutting down the test bench, the data acquisition window is printed and saved as a PDF file as a backup.

It should be pointed out that the bypass loop for the compressor is not used during measurements. The compressor turns out to be suitable for the ejector operation, and no excessive flow rate has been encountered. Therefore, the CV5 is always closed during the experiment.

4.5 Uncertainty analysis

The result of a measurement is only an approximation of the true value of the specific quantity. They always need to be accompanied by an uncertainty analysis. The measuring instruments are described in this section. The pressures, temperatures and flow rates are the parameters that need to be measured, and the related uncertainties are discussed.

4.5.1 Uncertainty analysis principles

The errors that occur when performing experimental measurements can be classified as random errors and systematic errors. Random errors are due to
fluctuations in the instruments and are hard to eliminate. Systematic errors can be eliminated or minimized by calibrations of the measurement equipment. More detailed description of uncertainty analysis can be found in Holman (2001), Jonsson (2001) and Callizo (2010).

If the measurement is determined by multiple readings, the random error can be determined by means of standard deviation from the mean values \( s_x \), it can be written as:

\[
\sigma_x = \sqrt{\frac{1}{n(n-1)} \sum_{i=1}^{n} (x_i - \bar{x})^2}
\]

where \( \sigma_x \) is the standard deviation computed from a sample of a large population, i.e. \( n > 10-20 \). The terms \( x_i \) (\( i = 1, 2, \ldots, n \)) are the sample of measurements and \( \bar{x} \) represents the mean values of the sample:

\[
\bar{x} = \frac{1}{n} \sum_{i=1}^{n} x_i
\]

In case the measurement is performed only once, the standard deviation has to be determined from experience, or if the maximum error \( \Delta x_i \) is known, then the standard deviation is calculated by:

\[
\sigma_x = \frac{\Delta x_i}{\sqrt{3}}
\]

The systematic error is not statistical, and its standard deviation \( w_x \) has to be decided based on previous measurement data, information acquired from instructions on measuring instrumentation and sensors, calibration certificates, etc.

By using root-sum-square (RSS), the overall uncertainty \( u_x \), which is due to the random error \( \sigma_x \) and the systematic error \( w_x \), is given as:

\[
u_x = \sqrt{\sigma_x^2 + w_x^2}
\]

Assuming a normal distribution, the above relation gives a 68% confidence level. To increase the confidence level, the overall uncertainty \( U_x \) can be expressed with a factor:
In this thesis, the confidence level of 95% is applied corresponding to a factor $k=2$.

Sometimes the desired quantity is based on several individual quantities with a set of measurements, and it is a given function of these independent quantities:

$$y = f(\bar{x}_1, \ldots, \bar{x}_i, \ldots, \bar{x}_m) \quad (4-6)$$

The combined overall uncertainty of $y$ is obtained from RSS of the standard uncertainties of $s_j$ and $w_j$, and becomes:

$$U_y = k \cdot \sqrt{s_j^2 + w_j^2} \quad (4-7)$$

where the $s_j$ and $w_j$ are given as:

$$s_j^2 = \sum_{i=1}^{m} \left( \frac{\partial f}{\partial \bar{x}_i} \cdot u_{\bar{x}_i} \right)^2 \quad (4-8)$$

$$w_j^2 = \sum_{i=1}^{m} \left( \frac{\partial f}{\partial \bar{x}_i} \cdot w_{\bar{x}_i} \right)^2 \quad (4-9)$$

Alternatively, the uncertainty in the desired quantity $U_y$ can be estimated by sequentially perturbing the mean values $\bar{x}_i$ with its own uncertainty $u_{\bar{x}_i}$ and can be written as:

$$U_y^2 = \sum_{i=1}^{m} \left( \frac{\partial f}{\partial \bar{x}_i} \cdot u_{\bar{x}_i} \right)^2 \approx \sum_{i=1}^{m} [f(\bar{x}_i + u_{\bar{x}_i}) - f(\bar{x}_i)]^2 \quad (4-10)$$

4.5.2 Uncertainty in temperature measurements

The temperatures in the experiment were measured by T-type thermocouples. A method proposed by Palm (1991) was used. All thermocouples were connected to a junction box in which the reference junctions were placed in an isothermal block. As a result, all thermocouples
had the same reference temperature, and the offset errors due to errors in
the reference temperature could be eliminated. Before installation, the
thermocouples were calibrated in a thermostatic bath from 10 °C to 95 °C,
and the standard deviation was found to be within ±0.035 °C.

The DC voltage measurement accuracy at 100 mV range signal is 0.004
mV, which corresponds to an error of ±0.024 °C. The conversion accuracy
(the error in the function converting the voltage to temperature) is less than
±0.05 °C, based on the logger manufacturer (Agilent technologies). So the
systematic uncertainty of temperature measurement is estimated at ±0.2 °C.

4.5.3 Uncertainty in pressure measurements

The absolute pressures at the ejector inlets and outlet are very important and
measured by three pressure transmitters ($P_g$, $P_e$ and $P_c$) from Yokogawa.
These pressure transmitters were also adjusted in the required spans
according to our applications and calibrated by the manufacturer. Four
other pressure gauges ($P_1$-$P_4$) were used to indicate the pressure levels at
different locations (see Figure 4-3), they were not used in the calculations.
The specifications of pressure measurements are listed in Table 4-2.

<table>
<thead>
<tr>
<th>Pressure</th>
<th>Model</th>
<th>Max Range</th>
<th>Span</th>
<th>Output</th>
<th>Accuracy or Scale</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_g$</td>
<td>EJA510A</td>
<td>0-100 abs</td>
<td>0-18 abs</td>
<td>4-20</td>
<td>0.2 % of Span</td>
</tr>
<tr>
<td>$P_e$</td>
<td>EJA510A</td>
<td>0-100 abs</td>
<td>0-5 abs</td>
<td>4-20</td>
<td>0.2 % of Span</td>
</tr>
<tr>
<td>$P_c$</td>
<td>EJA510A</td>
<td>0-100 abs</td>
<td>0-10 abs</td>
<td>4-20</td>
<td>0.2 % of Span</td>
</tr>
<tr>
<td>$P_1$ and $P_2$</td>
<td></td>
<td>-1-30</td>
<td></td>
<td>-</td>
<td>Scale: 0.1 bar</td>
</tr>
<tr>
<td>$P_3$ and $P_4$</td>
<td></td>
<td>-1-10</td>
<td></td>
<td>-</td>
<td>Scale: 0.2 bar</td>
</tr>
</tbody>
</table>

For the current measurement of the three pressure transmitters, the
accuracy in the data logger at a 100 mA range signal is ±0.05% of the
reading and ±0.005% of the range, obtained from the manufacturer (Agilent
technologies), which leads to a maximum error of ±0.015 mA. Thus the
uncertainty of the pressure measurement in the data logger is evaluated at
±0.1% of reading. The conversion error (from current to pressure) is less
than ±0.04%. Therefore, the systematic uncertainties of the pressure
measurements are estimated to be ±0.5%.
4.5.4 Uncertainty in mass flow rate measurements

The flow rates of the refrigerant were directly measured by two highly accurate coriolis mass flow meter from Yokogawa. One was located before the ejector nozzle to measure the mass flow rate of the primary flow \( \dot{m}_g \) for the ejector, and another was to measure the mixed flow rate at the ejector outlet \( \dot{m}_c \). Both flow meters were calibrated by the manufacturer according to the designed applications. Table 4-3 presents the details and specifications of the flow meters.

<table>
<thead>
<tr>
<th>Mass flow</th>
<th>Model</th>
<th>Max (kg/h)</th>
<th>Nom (kg/h)</th>
<th>Span (kg/h)</th>
<th>Output (mA)</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \dot{m}_g )</td>
<td>RCC T34</td>
<td>5000</td>
<td>3000</td>
<td>0-300</td>
<td>4-20</td>
<td>± 0.5% of flow rate ± 0.15/flow rate 100%</td>
</tr>
<tr>
<td>( \dot{m}_c )</td>
<td>RCC T34</td>
<td>5000</td>
<td>3000</td>
<td>0-300</td>
<td>4-20</td>
<td>± 0.5% of flow rate ± 0.15/flow rate 100%</td>
</tr>
</tbody>
</table>

Max - maximum mass flow rate. Nom - nominal mass flow rate.

Since the output signals of the two mass flow meters are the same as the pressure transmitters, which are both 4-20 mA, the uncertainty of the mass flow measurement in the data logger is ±0.1% and the conversion accuracy (from current to flow rate) has the same limit at ±0.04%. Therefore, the systematic uncertainties of the mass flow measurements are estimated to be within ±2%.

4.5.5 Uncertainty of the ejector entrainment ratio

The ejector entrainment ratio \( \mu \) is defined as the mass flow rate of the secondary flow \( \dot{m}_c \) to the mass flow ratio of the primary flow \( \dot{m}_g \). The measured mass flow \( \dot{m}_c \) is the mixed flow, which is the sum of the primary flow and the secondary flow. Thus the ejector entrainment ratio is calculated from the experimental data as:

\[
\mu = \frac{\dot{m}_c}{\dot{m}_g} = \frac{\dot{m}_c - \dot{m}_g}{\dot{m}_g} = \frac{\dot{m}_c}{\dot{m}_g} - 1
\] (4-11)
Therefore, the uncertainty in the entrainment ratio $U_\mu$ is given as the uncertainties in the $\dot{m}_g$ and $\dot{m}_c$.

\[
U_\mu = \pm \frac{1}{\mu} \cdot \sqrt{\left( \frac{\partial \mu}{\partial \dot{m}_g} \cdot w_{\dot{m}_g} \right)^2 + \left( \frac{\partial \mu}{\partial \dot{m}_c} \cdot w_{\dot{m}_c} \right)^2}
\]  

(4-12)

Hence, the relative uncertainty in the entrainment ratio is estimated at $\pm 3\%$.

4.5.6 Uncertainty of the nozzle efficiency

As mentioned, the nozzle efficiency can be quantified from the states at the nozzle inlet and exit based on its definition in Eq. (2-5). However, in the experiments no data is available at the nozzle exit. Hence, Eq. (2-5) cannot be used. An alternative way to determine the nozzle efficiency is by assuming ideal gas behavior. This can be derived from the choked mass flow rate in the nozzle throat for the primary flow $\dot{m}_g$ that can be written as (Huang et al., 1999):

\[
\dot{m}_g = P_g \cdot A_1 \cdot \sqrt{\frac{k}{R T_p} \cdot \left( \frac{2}{k+1} \right)^{k+1}}
\]  

(4-13)

Using the ideal gas assumption will not provide exact results of the nozzle efficiency as the ideal gas law is not accurate at the nozzle inlet. At present, this is the only way nozzle efficiency can be quantified from the experimental data.

Rearranging Eq. (4-13):

\[
\eta_n = \left( \frac{\dot{m}_g}{P_g} \right)^2 \cdot \left( \frac{2}{D_1} \right)^4 \cdot \frac{R \cdot T_p}{\pi^2 k} \cdot \left( \frac{k+1}{k-1} \right)^{k+1}
\]  

(4-14)

Obviously, the nozzle efficiency $\eta_n$ is a function of the mass flow rate of the primary flow $\dot{m}_g$, the measured pressure $P_g$ and temperature $T_p$, the used nozzle throat diameter $D_1$ and the specific heat capacity ratio $k$ from REFPROP 9.0 (Lemmon et al., 2010). However, the uncertainty in $k$ is
ignored in this thesis since there is no information to be found in REFPROP 9.0.

The nozzle throat diameter plays a significant role in the nozzle efficiency. The micro-gauge (2-point bore gauge) from Bowers was used to accurately measure the nozzle throat diameters. Three measuring heads with different ranges of 1.80-2.20 mm, 2.05-2.45 mm and 2.25-2.75 mm were used to measure the nozzle throat diameter in ejectors E1-E3, ejectors E4-E5 and ejector E6, respectively. According to the manufacturer’s specification, the micro-gauges had a maximum uncertainty of ±1% of measuring travel, and a minimum of ±0.001 mm for the diameter range of 1.50-6.35 mm. In the present study, the uncertainty of the nozzle throat diameters is set at ±0.003 mm for all the ejector nozzles.

Then the uncertainty of the nozzle efficiency $\eta_n$ is calculated as:

$$
U_{\eta_n} = \pm \eta_n \left( \sqrt{\left( \frac{\partial \eta_n}{\partial m_g} \cdot w_{m_g} \right)^2 + \left( \frac{\partial \eta_n}{\partial P_g} \cdot w_{P_g} \right)^2 + \left( \frac{\partial \eta_n}{\partial T_g} \cdot w_{T_g} \right)^2 + \left( \frac{\partial \eta_n}{\partial D_1} \cdot w_{D_1} \right)^2} \right)
$$

(4-15)

Hence, the relative uncertainty in the nozzle efficiency $\eta_n$ is estimated to be ±4.2%. The uncertainty in the mass flow rate of the primary flow accounts for approximately 92% of the total uncertainty in the nozzle efficiency.

### 4.5.7 Uncertainty of the mixing efficiency and diffuser efficiency

From the experimental data, the nozzle efficiency $\eta_n$ can be calculated by Eq. (4-14). The mixing efficiency $\eta_m$ and the diffuser efficiency $\eta_d$ are further calculated from the ejector model in Paper I. The calculation flow diagram has been introduced in Paper III and is also presented in Figure 6-6. It is noted that the $\eta_m$, $\eta_m$ and $\eta_d$ are depending on many parameters, and they are closely related and interacting. However, since $\eta_n$ is an input data in the calculations to quantify $\eta_m$ and $\eta_d$, the functions of $\eta_m$ and $\eta_d$ can be expressed as:

$$
\eta_m = f(P_g, P_e, P_c, T_g, T_e, \mu_{exp}, Ar_{exp}, \eta_n)
$$

(4-17)

$$
\eta_d = f(P_g, P_e, P_c, T_g, T_e, \mu_{exp}, Ar_{exp}, \eta_n)
$$

(4-18)
Using Eq. (4-10) and the above uncertainties for $P_g$, $P_e$, $P_c$, $T_e$, $\mu_{\text{exp}}$, $A_{r_{\text{exp}}}$, $\eta_m$, uncertainties of the mixing efficiency $\eta_m$ and the diffuser efficiency $\eta_d$ are evaluated at 3.5% and 1.6%, respectively.

It should be noted that although the mixing efficiency $\eta_m$ and the diffuser efficiency $\eta_d$ are expressed as Eqns. (4-17) and (4-18) based on the model presented in Paper I. In reality, they are also influenced by other parameters that are not included in the model, especially the other ejector physical parameters, e.g. the internal surface roughness, the ejector divergent and convergent angles, the length of constant area mixing chamber.
5 Summary of Papers and an Extended Study

The thesis is based on the four journal articles and one conference. Paper I and Paper III are published in Applied Thermal Engineering. Paper II is published in International Journal of Refrigeration. Paper IV is submitted for publication in Applied Energy. Paper V is a conference paper and has been presented at the 11th IIR Gustav Lorentzen Conference on Natural Working Fluids, Hangzhou, China, 2014. A sensitivity analysis is further presented in this chapter to extend the study in Paper V.

In addition, two more papers have been written during this study. They are omitted in this thesis. Paper VI is submitted for publication in Renewable and Sustainable Energy Reviews. Paper VII is published in the conference proceedings of the 23rd IIR International Congress of Refrigeration, Prague, Czech Republic, 2011.

5.1 Paper I

5.1.1 Outline

The paper presents an ejector model designed to determine the performance of the ERS and to obtain the required ejector area ratio at a specific operating condition. The ejector is modeled based on that the mixing pressure is lower than the secondary flow pressure, and a normal shock occurs before entering the diffuser. The mixing pressure is solved by the ideal gas assumption. The irreversibilities in the ejector are considered through the isentropic efficiencies in the nozzle and the diffuser, and a mixing efficiency (defined as a kinetic energy ratio) in the mixing chamber. The proposed ejector model is validated by two sets of experimental data from literature. The effects of the ejector operating conditions and ejector efficiencies are preliminarily investigated.

5.1.2 Conclusions

The results of the prediction from the proposed model are very accurate when the ejectors are working at their optimum operating conditions. The prediction is less favorable for fixed-geometry ejectors under wide ranges of operating conditions with fixed ejector efficiencies.
The ejector entrainment ratio increases with the increasing of the generator temperature and evaporator temperature, as expected. However, an increase in the condenser temperature leads to a gradual decrease in the entrainment ratio, instead of a sudden drop observed in a fixed-geometry ejector with a critical condenser pressure. This is because the ejector area ratio needs to be adjusted to provide sufficient flow area for the flow to expand, mix and compress for the entrainment ratio to be achieved, and to adapt to different operating conditions in order to maintain the optimum performance. The ejector entrainment ratio, area ratio and mixing pressure are more significantly influenced by the condenser saturation temperatures than by the generator saturation and evaporator saturation temperatures.

The ejector efficiencies are crucial parameters in the present model. The ejector entrainment ratio largely increases with increasing any one of the three ejector efficiencies. The calculated ejector area ratio is more sensitively influenced by the nozzle and mixing efficiencies. The mixing pressure is more closely connected to the diffuser efficiency since the compression process is accomplished through a normal shock and further pressure recovery in the diffuser. Investigation of these efficiencies is of utmost importance for the ejector design and performance evaluation.

5.2 Paper II

5.2.1 Outline

The working fluid significantly influences the ejector behavior and ejector design as well as the ERS performance. Criteria of selecting a working fluid for the ERS have been presented in this paper. Moreover, the working fluids can be classified as wet fluids, dry fluids and isentropic fluids. During the expansion in the ejector, droplets form when using wet fluids. Dry fluids and isentropic fluids also have the risk of droplet formation at operating conditions close to the critical point. Special attention is paid to determining the required superheat of the primary flow before entering the nozzle. A minimum superheat is found to ensure the elimination of droplets and maximize the system performance. With the knowledge of the minimum superheat, four wet fluids (R134a, R152a, R290 and R430A), four dry fluids (R245fa, R600, R600a and R1234ze) and one isentropic fluid (R436B) are selected to comparatively study their performance and feasibility in the ERS.
5.2.2 Conclusions

Taking into consideration environmental effects and safety issues as well as other criteria for selecting a working fluid, there is no “perfect” working fluid suitable for the ERS out of the selected nine candidates. Each fluid has its own advantages and disadvantages.

The study shows that the minimum superheat not only depends on the used working fluid, but also is related to the generator saturation temperature and the ejector nozzle efficiency. For the wet fluids (R134a, R152a, R290 and R430A), the minimum superheat increases with increasing of the generator saturation temperature and the nozzle efficiency. For the dry fluids, R245fa, R600 and R600a, at the conditions considered, no superheat is needed, while for the dry fluid R1234ze and the isentropic fluid R436B, droplets form at a high generator saturation temperature close to the critical temperature. Excessive superheat of the primary flow slightly before entering the ejector nozzle slightly improves the entrainment ratio. However, its effects on the COP are insignificant. The excessive superheat leads to a decrease in system Carnot efficiency and requires a high temperature of the heat source to accomplish the operation. Therefore, it is of importance to introduce a superheat that is as small as possible, but sufficient to effectively avoid the droplet formation.

Parametric investigations of the operating temperatures show that, regardless of the working fluid, a high primary flow temperature (nozzle inlet temperature) and evaporator saturation temperature, or a low condenser saturation temperature is always positive for the ejector entrainment ratio and system COP, as could be expected. There exists a maximum Carnot efficiency for some of the working fluids. In general, the dry fluids have better COPs and larger ejector area ratios than the selected wet fluids and the isentropic fluid.

When the primary flow temperature, the condenser and evaporator saturation temperatures are fixed at 95 °C, 35 °C and 10 °C, respectively, and the ejector efficiencies are equally set at 0.9, R245fa and R600 have the highest COP of 0.38, followed by R600a at 0.35 and R1234ze at 0.33. A lower COP of 0.28 is found for R134a and R430A. For R152a and R290, the COP is around 0.25, the lowest COP is observed at 0.18 for R438B. Therefore, R600 is recommended as a good candidate from the perspectives of system performance and environmental concern. However, its flammability requires extra considerations.
5.3 Paper III

5.3.1 Outline

This paper is an extended investigation on the ejector behavior considering the area ratio, the entrainment ratio and ideal entrainment ratio. The exergy destruction, which is referred to as irreversibility in the paper, within the nozzle, the mixing chamber and the diffuser are evaluated, with a special concern on the shock process. Three common dry refrigerants, R141b, R245fa and R600a, are selected as the working fluids for the parametric study gain a better understanding of the ejector working characteristics in the ERS. The ejector variables have been divided into external parameters, i.e. pressures and temperatures, and internal parameters, namely the ejector efficiencies. The ejector efficiencies are correlated based on data from the literature.

5.3.2 Conclusions

For the external parameters, a higher generator saturation temperature and evaporator saturation temperature, and a lower condenser saturation temperature always boost the ejector performance by reducing the ejector exergy destruction, and increasing the ideal entrainment ratios. Due to different properties of the three refrigerants, the exergy destruction related to the shock process varies. The contribution of the shock to the total exergy destruction for R600a is much greater than that of R141b and R245fa under the same operating conditions. In the diffuser, the shock exergy destruction dominates. The system COP, the ejector area ratio, and the exergy destruction related to the shock process are independent of the superheats of the primary flow and the secondary flow. As a result, the superheats of the primary flow or the secondary flow are not economically justifiable for the dry fluids.

Regarding the internal parameters, the ejector efficiencies have dramatic effects on the system COP and the exergy destructions within the nozzle, mixing chamber and diffuser. Changing any of these ejector efficiencies would not only bring large changes of its own exergy destruction, but also result in series of variations of exergy destruction in the other two ejector parts. The variation tendencies of exergy destruction influenced by the three efficiencies are quite similar. At the considered operating condition (the saturation temperatures in the generator, condenser and evaporator are fixed at 95 °C, 35 °C and 10 °C, respectively, with no superheats, and the ejector efficiencies are the same with a value of 0.9), the largest exergy destruction
occurs in the mixing process for R141b and R245fa, but for R600a the largest exergy destruction occurs in nozzle.

The quantification based on experimental data from literature demonstrates the ejector parameters are closely related. The operating conditions and ejector area ratios have great impact on the ejector efficiencies, and these ejector efficiencies in turn significantly influence the system COP. A special care is always needed when dealing with these efficiencies.

5.4 Paper IV

5.4.1 Outline

The conventional exergy analysis pinpoints the location, magnitude and sources of exergy destruction and exergy losses in the system. The advanced exergy analysis is able to further quantify the interactions among the system components and evaluate the realistic improvement potential. Splitting the exergy destruction in each system component into endogenous/exogenous and avoidable/unavoidable parts provides more detailed insights and additional useful information. This paper presents the investigation of the ERS using both conventional and advanced exergy analysis approaches.

Definitions of all the split exergy destruction parts, namely endogenous, exogenous, avoidable, unavoidable, and their combined exergy destruction parts, are described. Detailed calculations of these exergy destruction parts are schematically interpreted and performed with assistance of the real cycle, the ideal cycle, several hybrid cycles and the unavoidable cycle. The sensitivity study is carried out to analyze effects of the pinch point temperature differences in the generator, condenser and evaporator, as well as the ejector efficiencies on the exergy destruction parts. A thermodynamic improvement study is further conducted.

5.4.2 Conclusions

The conventional exergy analysis shows that at the considered operating condition, more than a half of the total exergy destruction comes from the ejector, and the generator has the second largest exergy destruction, followed by the condenser, the evaporator, the throttling valve and the pump. This indicates that the component with the highest priority to improve is the ejector.
The advanced exergy analysis gives more informative results. It is found that the exergy destruction within the ejector mostly comes from itself, indicated by its endogenous part of 71.2%, and the rest is from the remaining components, which once again emphasize the necessity to improve the ejector. The endogenous exergy destruction within the generator is much smaller than the exogenous part, revealing that its exergy destruction is mainly caused by the remaining system components, and its exergy destruction can be reduced through the improvements of the remaining components, especially the ejector. The unavoidable exergy destructions within the components (except for the pump) are higher than the avoidable part, and 35% of the overall exergy destruction can be avoided. The combinations of splitting approaches identify the order for system improvements: the ejector should be firstly considered, then the condenser, followed by the generator and evaporator. The pump and throttling valve are the last ones to be considered.

The results from the sensitivity analysis reveal that among the three heat exchangers, the pinch temperature difference in the condenser has the most significant influence on the total system exergy destruction and its own avoidable endogenous part of exergy destruction. The pinch temperature difference in the evaporator has the second largest influence, followed by the generator temperature difference. The ejector efficiencies also play a very important role in the parts of exergy destruction. An increase in the ejector efficiency leads to a large decrease in its own endogenous, exogenous, and the avoidable exergy destruction.

Case studies show that compared to the base case the improved ERS can obtain 30.2% reduction in system exergy destruction, 61.5% reduction in system avoidable exergy destruction and an increase in system exergy efficiency from 6.9% to 9.5%.

The application of advanced exergy analysis to the ERS provides detailed and useful information, and identifies the potential for improving such a system. The advanced exergy analysis can be viewed as a valuable supplement to the conventional exergy analysis.

5.5 Paper V

5.5.1 Outline

The thermoeconomic analysis combines the exergy analysis with economic principles to provide the cost information that is important to the design
and operation of an energy conversion system. This paper presents a thermoeconomic optimization of an ERS. The thermodynamic analysis, economic analysis, thermoeconomic analysis and optimization are performed step by step. The objective function (OBF) is defined as the sum of all the costs. The pinch temperature differences in the three heat exchangers are considered as the decision variables, with two different economic scenarios imposed to the system: scenario I includes all the costs related to the brine side fluids (steam and water), electricity, and costs related to capital investment and operation and maintenance expense. Scenario II excludes the costs of the steam and water by using free resources. An iteration technique is employed to optimize an ERS with the aim of minimizing the OBF.

5.5.2 An extended sensitivity analysis

Due to the large variations of the economic parameters in different locations, it is interesting to carry out a sensitivity analysis to study the impacts of these parameters on the ERS performance. This analysis is helpful to predict the results while some modifications are necessary in the modeling. It is considered as an additional complement to Paper V.

![Figure 5-1: Sensitivity of OBF to (a) the purchase equipment cost of the ejector, (b) the electricity price, (c) the interest rate, and (d) the annual operating hours.](image)
A sensitivity study concerning the effects of the ejector price, the electricity price, the interest rate and the annual operating hours on the OBF is further performed here. Figure 5-1 shows how the OBF increases with the increase of the ejector price, electricity price and interest rate, and how the OBF decreases with increasing of annual operating hours. Clearly, the base case and optimized case are subjected to similar variations. It is seen that the OBF values in scenario II are much smaller than those in scenario I, which once again means that the prices of steam and water (the brine side fluids) play very important roles in the OBF. Moreover, the differences of OBF between the base case and the optimized case in scenario I are larger than that in scenario II. Therefore, when the brine side fluids are charged at certain prices, i.e. scenario I, the system has more potential to be optimized compared to scenario II where the brine side fluids are free.

5.5.3 Conclusions

When the economic conditions are imposed on an ERS, the costs are allocated to each exergy stream as the exergy costing. Different economic conditions lead to large differences in the exergy costing. For scenario I, the OBF of the optimized case is reduced by 8.1% relative to the base case, and the system COP and exergy efficiency of the optimized case are improved by 23.8% and 23.9%, respectively. For scenario II, the optimized OBF is reduced by 7.5% compared to that in the base case. However, the system COP and exergy efficiency are decreased by 15.3% and 14%, respectively. That is to say, the thermoeconomic optimization of scenario I is accomplished by improving the system thermodynamic performance, while scenario II has to compromise the thermodynamic performance to gain the thermoeconomic optimization. Thus the thermoeconomic optimization is not always the same as the thermodynamic improvement of the system.

The prices of the brine side fluids play very important roles in the thermoeconomic optimization, and costs of the brine side fluids contributes 43.9% and 40.2% to the OBF in the base case and the optimized case in scenario I. When the brine side fluids are considered as free in scenario II, the cost is dominated by the capital investment and operation and maintenance expenses.

The sensitivity study shows how the OBFS increase with the increasing of the ejector price, electricity price and interest rate, but decrease with the increasing of operating hours. When all the brine side fluids are charged, the ERS has more potential to be optimized. It shows that the total cost is about 44% lower if the brine side fluids are free.
6 Experimental Results and Discussion

The experimental facilities have been described in Chapter 4. Six ejectors (E1-E6) have been tested, their throat diameters and area ratios are listed as: E1 ($D_1=2.014$ mm and $Ar=9.79$), E2 ($D_1=2.008$ mm and $Ar=9.85$), E3 ($D_1=2.021$ mm and $Ar=9.71$), E4 ($D_1=2.207$ mm and $Ar=8.15$), E5 ($D_1=2.243$ mm and $Ar=7.89$), E6 ($D_1=2.504$ mm and $Ar=6.33$), as shown in Figure 4-2 and Table 4-1. This chapter presents the test results of these six ejectors. The superheat of the secondary fluid is studied. The quantified ejector efficiencies and the calculated result from the proposed ejector model (Paper I) are discussed. The assumption of neglecting the velocities at the ejector inlets and outlet is validated.

For each data point, two mass flow rates $m_g$ and $m_c$, three pressures $P_g$, $P_e$ and $P_c$ and nineteen temperatures (including temperatures at the ejector inlets $T_p$, $T_s$ and outlet $T_{out}$), is obtained by extracting and averaging from a condition that all twenty-four measured parameters have been kept steady for at least 25 minutes. The primary flow pressure $P_g$ is the major parameters that could be regulated in the test bench, thus it is chosen as the independent variable in the discussion. It is noted that each primary flow is accompanied by a certain superheat. The superheat has been demonstrated to have a slight increase in the ejector entrainment ratio theoretically (Paper III) and experimentally (García del Valle et al., 2014). However, the effect of the superheat of the primary flow is not discussed here since the superheat of the primary flow can barely be controlled.

It should be noted that since the test bench lacks effective control functions and the pressure levels are not well regulated, the achieved operating conditions are very limited. Especially the secondary flow temperature has never been pulled down to the typical temperature for air-conditioning operation. However, it is still of interest to discuss some features that the ejectors exhibit.

6.1 Ejector performance

Since pressures of the secondary flow $P_e$ and the ejector outlet $P_c$ are not effectively regulated as constant, they could be considered as a consequence of the primary flow. Figure 6-1 shows the effects of the primary flow pressure $P_g$ on the other two pressures $P_e$ and $P_c$. The data for different
ejectors are marked with different colors. Clearly, an increase in the primary flow pressure $P_g$ results in the general tendencies of increasing the secondary flow pressure $P_e$ and the ejector outlet pressure $P_c$. Unfortunately, the established pressure differences $P_c-P_e$ and the pressure lift ratio $P_c/P_e$ are rather low for all ejectors, ranging from 0.22 to 0.94 bar and from 1.15 to 1.57 respectively. This is due to that the ejectors are not well designed on the one hand, and the throttling valves are needle valves and could not significantly reduce the pressure as desired on the other hand.

With respect to the ejector outlet pressure $P_c$, it can be seen that when it comes to the same primary flow pressure $P_g$, ejectors E1–E3, which have the similar $Ar$, could have smallest values of $P_c$. The ejector E4 ($Ar=8.15$) has a modest values of $P_c$ and E5 ($Ar=7.89$) has a little higher $P_c$. The ejector E6 with a smallest $Ar=6.33$ has the largest $P_c$. However, the critical back pressure $P_b^*$, as shown in Figure 2-4, is not determinable for each individual ejector since the pressures of the primary flow $P_g$ and the secondary flow $P_e$ always change rather than stay constant.

Moreover, the obtained entrainment ratio $\mu_{\text{exp}}$ calculated as $(m_{c}-m_{g})/m_{g}$ from the data varies from 0.14 to 0.31. Since the pressures of the primary flow $P_g$ and the secondary flow $P_e$ as well as the outlet pressure $P_c$ vary simultaneously, as shown in Figure 6-1, no general conclusions can be drawn about the influence of these parameters on the entrainment ratio.

![Figure 6-1 Variations of the secondary flow pressure, the ejector outlet pressure and the entrainment ratio with the primary flow pressure.](image-url)
6.2 Superheat of the secondary flow

In Paper III, the superheat of the secondary flow is parametrically studied and it has insignificant influence on the entrainment ratio \( \mu \). To validate this conclusion, the effect of superheating the secondary flow is performed experimentally under the conditions that the primary flow pressure \( P_g \) and temperature \( T_p \), the pressures of the secondary flow \( P_e \) and the outlet flow \( P_c \) are kept constant. This can be achieved by only adjusting the temperature of the thermostatic bath (see Figure 4-3). The superheat is studied on ejector E3 when \( P_g=8.61 \) bar, \( T_p=99.1 \) °C, \( P_e=1.22 \) bar and \( P_c=1.69 \) bar, ejector E5 (\( P_g=7.35 \) bar, \( T_p=105.3 \) °C, \( P_e=1.34 \) bar and \( P_c=1.95 \) bar), and ejector E6 with two cases: Case 1 (\( P_g=7.44 \) bar, \( T_p=83.4 \) °C, \( P_e=1.71 \) bar and \( P_c=2.26 \) bar) and Case 2 (\( P_g=6.77 \) bar, \( T_p=79.3 \) °C, \( P_e=1.61 \) bar and \( P_c=2.06 \) bar), respectively. The results are shown in Figure 6-2. It is obviously seen that the effect of the superheat of the secondary flow (\( \Delta T_s \)) on \( \mu_{\text{exp}} \) is negligible, which confirms the theoretical results from Paper III.

![Figure 6-2 Variations of the entrainment ratio with the superheat of the secondary flow.](image)

6.3 Mass flow rate of the primary flow

For a vapor ejector in the ERS, the primary flow is choked at the nozzle throat. The mass flow rate of the primary flow \( m_g \) is directly related to the primary flow pressure and temperature \( P_g \) and \( T_p \) and the nozzle throat diameter \( D_1 \). Figure 6-3 shows the real time data of the primary flow, i.e. mass flow rate \( m_g \), pressure \( P_g \) and temperature \( T_p \), in a test when using ejector E2. The corresponding saturation temperature \( T_{g,sat} \) from REFPROP

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(Lemmon et al., 2010) is also plotted to indicate the magnitudes of the superheat of the primary flow \((T_p - T_{g,\text{sat}})\). At the first 50 minutes, the primary flow is in two-phase condition, and after that it is superheated. More importantly, the primary flow pressure \(P_g\) and the mass flow rate \(\dot{m}_g\) have experienced similar variations. Therefore, it can be concluded that when the nozzle throat diameter is fixed, the mass flow rate of the primary flow is dominantly determined by the incoming pressure \(P_g\) since the flow is choked at the nozzle throat.

![Figure 6-3 Real time data of the primary flow](image)

The mass flow rate for an ideal gas through the nozzle with an isentropic process \(\dot{m}_{g,\text{ideal}}\) can be written as:

\[
\dot{m}_{g,\text{ideal}} = P_g \cdot A_1 \cdot \sqrt{\frac{k}{R \cdot T_p} \left( \frac{2}{k+1} \right)^{\frac{k+1}{k-1}}} \tag{6-1}
\]

When using the measured primary flow conditions \(P_g\) and \(T_p\) in Eq. (6-1), the ideal mass flow rate can be calculated. The ideal mass flow rate \(\dot{m}_{g,\text{ideal}}\) and the measured mass flow rate \(\dot{m}_g\) as a function of the primary flow pressure \(P_g\) are shown in Figure 6-4. Obviously, the measured mass flow rate \(\dot{m}_g\) is quite close to the calculated ideal mass flow rate \(\dot{m}_{g,\text{ideal}}\). They have similar changes, meaning that the mass flow rate of the primary flow of R245fa can be reasonably well predicted by the ideal gas assumption. Moreover, both ideal mass flow rate \(\dot{m}_{g,\text{ideal}}\) and measured mass flow rate \(\dot{m}_g\)
increase approximately linearly with the increase of incoming pressure $P_g$ for all six ejectors. This also confirms the characteristic of pressure $P_g$ and the mass flow rate $\dot{m}_g$ in Figure 6-3. The primary flow temperature $T_p$ also influences the mass flow rates $\dot{m}_{g,\text{ideal}}$ and $\dot{m}_g$. Taking ejector E6 (in blue) in Figure 6-4 as an example, it is found that there exist different values of mass flow rate $\dot{m}_g$ at the same pressure $P_g$, which is due to different generator temperatures $T_p$. A 20 K increase in the temperature $T_p$ decreases mass flow rate $\dot{m}_g$ by about 5%. Therefore, the effect of temperature on the $\dot{m}_{g,\text{ideal}}$ is not very significant.

Another decisive parameter for the mass flow rate of the primary flow is the nozzle throat diameter $D_1$. Ejectors E1, E2 and E3 have similar mass flow rates since they have similar nozzle throat diameter $D_1$, which is also observed for ejectors E4 and E5. Large difference in nozzle throat diameter leads to largely different mass flow rate. Moreover, when the ejectors work at the same primary flow condition, the ideal mass flow rate $\dot{m}_{g,\text{ideal}}$ depends only on $D_1$, this can be easily obtained from Eq. (6-1). However, there is one more parameter that impacts the real mass flow rate $\dot{m}_g$, which is the nozzle efficiency $\eta_n$. The nozzle efficiency $\eta_n$ explains the difference between mass flow rate $\dot{m}_{g,\text{ideal}}$ and $\dot{m}_g$ at the same condition in Figure 6-4.

![Figure 6-4 Variations of mass flow rates with the primary flow pressure.](image-url)
6.4 Ejector efficiencies

6.4.1 Nozzle efficiency

The actual flow in the ejector nozzle is a non-isentropic expansion and introduces irreversibilities, which result from frictional effects, flow separation and internal heat transfer. The nozzle efficiency $\eta_n$ is defined in Eq. (2-5), but is calculated from Eq. (4-14) which assumes ideal gas behavior. This assumption introduces errors that have yet to be quantified. Figure 6-5 shows the nozzle efficiency $\eta_n$ calculated according to Eq. (4-14) versus the primary flow pressure $P_g$. The nozzle efficiency $\eta_n$ varies from 0.853 to 0.962. The distribution of nozzle efficiency $\eta_n$ is scattered and the magnitude of the nozzle efficiency $\eta_n$ depends not only on the ejector design and manufacturing methods, but also on the operating conditions. Obviously, ejector E4 has the smallest nozzle efficiency $\eta_n$, which might be because nozzle 4 is roughly manufactured and its internal surface is not as smooth as the other nozzles, introducing larger irreversibilities in the nozzle.

![Figure 6-5 Variations of nozzle efficiency with the primary flow pressure.](image)

6.4.2 Mixing efficiency and diffuser efficiency

The flow in the mixing chamber and the diffuser is even more complicated, involving supersonic flow, strong flow interactions, shear forces, turbulent mixing, shocks and vortices. The detailed flow information is still not clear. In the proposed ejector model (Paper I), the mixing efficiency $\eta_m$ is defined as a kinetic energy ratio, and the diffuser efficiency $\eta_d$ is defined as an isentropic efficiency because of the shock, friction and flow separation in
Since the nozzle efficiency $\eta_n$ has been estimated assuming ideal gas behavior, the mixing efficiency $\eta_m$ and the diffuser efficiency $\eta_d$ are quantified to get the best prediction of both measured entrainment ratio $\mu_{\text{exp}}$ and the used ejector area ratio $A_{\text{r,exp}}$. The process is similar with that in Paper III, as shown in Figure 6-6.

The quantified efficiencies $\eta_m$ and $\eta_d$ are plotted in Figure 6-7 as a function of the primary flow pressure $P_g$. The mixing efficiency $\eta_m$ has a range from 0.800 to 0.948, while the diffuser efficiency $\eta_d$ varies from 0.695 to 0.763. Because the geometries of the mixing chamber are fixed and different ejector nozzles have different shapes, e.g. nozzle divergent angles, nozzle exit position (NXPs), nozzle external diameters, they lead to a large variation in the mixing efficiency $\eta_m$. The rather small variations of the diffuser efficiency $\eta_d$ could be explained by the the fact that the same diffuser for the six ejectors is used. For the individual ejector at different conditions, the variations of the ejector efficiencies come from the different operating conditions.
Figure 6-7 Variations of the mixing efficiency and diffuser efficiency with the primary flow pressure.

To some extent, the relatively low values of efficiencies $\eta_m$ and $\eta_d$ indicate that the ejectors are not well functioning. This might mainly be due to that the ejectors are working at off-design operating conditions, and the ejector geometries are not well designed. Meanwhile, the manufacturing techniques could also take responsibility for the low efficiencies $\eta_m$ and $\eta_d$. For example, the rough internal surfaces of the used ejectors, the sharp corners, the welding joints, etc., could decrease the ejector efficiencies. Finally, although the ejectors are covered by the insulation, the heat losses to ambient always exist. This also has influence on the ejector efficiencies since the ejector walls have been assumed to be adiabatic in the ejector model.

6.4.3 Other results from quantifying the ejector efficiencies

In the process of quantifying the ejector efficiencies, the ejector entrainment ratio $\mu_{\text{cal}}$, the area ratio $A_{\text{cal}}$, and the ejector outlet temperature $T_{\text{out,cal}}$ are determined by the model and the results compared with the experimental data. They are subjected to the changes of ejector operating conditions and very sensitive to the ejector efficiencies ($\eta_n$, $\eta_m$, and $\eta_d$). The errors are given as:

$$\text{error} = \frac{\text{experiment} - \text{calculation}}{\text{experiment}} \times 100\%$$

(6-2)

Figure 6-8, Figure 6-9 and Figure 6-10 show the comparisons between the calculated results ($\mu_{\text{cal}}$, $A_{\text{cal}}$ and $T_{\text{out,cal}}$) obtained at the process of
quantifying the efficiencies $\eta_m$ and $\eta_d$, and the experimental data ($\mu_{\text{exp}}, A_{\text{exp}}$ and $T_{\text{out,exp}}$).

**Figure 6-8** Comparison of the calculated entrainment ratio to the measured entrainment ratio.

**Figure 6-9** Comparison of the calculated ejector area ratio to the used ejector area ratio.
Since the calculated entrainment ratio $\mu_{\text{cal}}$ and area ratio $A_{\text{r,cal}}$ has been targeted to get close to measured $\mu_{\text{exp}}$ and $A_{\text{r,exp}}$ by changing the efficiencies $\eta_m$ and $\eta_d$, the errors are within ±6.0% and ±9.0%, respectively. There are six ejector area ratios $A_{\text{r,exp}}$ used, thus the calculated area ratio $A_{\text{r,cal}}$ is plotted as six vertical groups in Figure 6-9. It is noted that no strategy is applied to obtain close predictions of the calculated ejector outlet temperature $T_{\text{out,cal}}$. The calculated values of the ejector outlet temperatures $T_{\text{out,cal}}$ have rather large deviations from the experimental values $T_{\text{out,exp}}$, as shown in Figure 6-10. The largest difference between $T_{\text{out,cal}}$ and $T_{\text{out,exp}}$ is found to be 12.6 °C. Most calculated ejector outlet temperatures $T_{\text{out,cal}}$ (91.3% of the date) are within the difference of ±7.3 °C from the experimental values $T_{\text{out,exp}}$.

It should be noted that Figure 6-8, Figure 6-9 and Figure 6-10 are just to indicate how close the calculated values $\mu_{\text{cal}}$, $A_{\text{r,cal}}$ and $T_{\text{out,cal}}$ are to the measured data when quantifying the efficiencies $\eta_m$ and $\eta_d$. Ideally, errors should be zero. However, it is very hard to further significantly decrease the errors since the calculated $\mu_{\text{cal}}$, $A_{\text{r,cal}}$ are subjected to different sensitivities of efficiencies $\eta_m$ and $\eta_d$, and the calculation aims to make both calculated $\mu_{\text{cal}}$ and $A_{\text{r,cal}}$ as close to the measured $\mu_{\text{exp}}$ and $A_{\text{r,exp}}$ as possible. The calculated $\mu_{\text{cal}}$, $A_{\text{r,cal}}$ and $T_{\text{out,cal}}$ are only validated under the specific operating conditions and the corresponding obtained ejector efficiencies $\eta_m$, $\eta_m$ and $\eta_d$ given in the Figure 6-5 and Figure 6-7. The ejector efficiencies play crucial roles on the accuracy of the predictions for the proposed ejector model.
6.5 Validation of the assumption of neglecting the velocities

In the ejector modeling, the total energy at the ejector inlets and outlet are normally expressed as equivalent to the fluid enthalpy. This is achieved by ignoring the velocities at the ejector inlets $u_g$ and $u_e$ and outlet $u_c$ since the kinetic energy is much smaller than the enthalpy of the fluid at these locations. This assumption is also applied in the present study. It is of interest to identify how insignificant the kinetic energy could be when compared to the enthalpy. Using the test data and the areas of the ejector inlets and outlet, the velocities, i.e. $u_g$, $u_e$ and $u_c$, are obtained in the range of 2.5-5.1 m/s, 3.6-10.4 m/s and 11.8-15.5 m/s, respectively.

The enthalpy differences in the ejector nozzle ($\Delta h_g$) and in the secondary fluid from the inlet into the mixing chamber ($\Delta h_e$) as well as in the diffuser ($\Delta h_c$) are introduced to compare the magnitudes of the kinetic energy. By defining a ratio of the kinetic energy to the enthalpy difference at the ejector inlets and outlet, these equations are obtained as:

$$
\psi_g = \frac{1}{2} \frac{u_g^2}{\Delta h_g}
$$

$$
\psi_e = \frac{1}{2} \frac{u_e^2}{\Delta h_e}
$$

$$
\psi_c = \frac{1}{2} \frac{u_c^2}{\Delta h_c}
$$

The maximum values of the defined $\psi_g$, $\psi_e$ and $\psi_c$ are estimated at 0.03%, 1.29% and 0.90% for the primary flow inlet and the secondary flow inlet as well as the outlet, respectively. This indicates that the kinetic energy is really insignificant compared to the enthalpy difference, and confirms the validity of neglecting the velocities at the ejector inlets and outlet.
7 Conclusions and Suggestions for Future Work

7.1 Conclusions

Theoretical investigations on the ejector refrigeration system and experimental measurements of vapor ejectors have been performed to discover more detailed working characteristics. Based on the studies carried out within this thesis, the following conclusions can be drawn.

7.1.1 Theoretical analysis

An ejector model has been developed to evaluate the ejector performance and the corresponding required ejector area ratio. The model can also be used to quantify the ejector efficiencies by coupling it to established data. The application of the ideal gas law in the ejector model has been verified to be effective and valid.

The droplet formation during expansion in the nozzle occurs for wet fluids, and it may also happen for dry and isentropic fluids at certain operating conditions. A minimum superheat of the primary flow before entering the ejector nozzle is proposed to eliminate droplet formation and to ensure the ejector performance. It depends on the working fluid and the generator saturation temperature as well as the nozzle efficiency. The excessive superheat of the primary flow and the superheat of the secondary flow are not practically justified.

Regardless of working fluids, a high primary flow temperature or evaporator temperature, or a low condenser temperature is found to benefit the ejector entrainment ratio and system COP, as could be expected. There is an optimum generator temperature giving the highest Carnot efficiency for some working fluids. Moreover, the ejector area ratio is required to increase if the entrainment ratio increases. The ejector entrainment ratio and area ratio are more significantly influenced by the condenser temperature and less influenced by the temperatures of the primary flow and in the evaporator. In general, dry fluids have higher COPs and larger ejector area ratios than the selected wet fluids and isentropic fluid.

The three defined ejector efficiencies have dramatic effects on the system COP and ejector exergy destruction inside the ejector. The exergy destruction related to the shock process dominates in the diffuser. When
working at the same condition, different refrigerants could have largely different exergy destruction inside the ejector. Particular concern should be paid to the correct selection of the values for these efficiencies.

The advanced exergy analysis of an ejector refrigeration system demonstrates that at the selected condition the exergy destructions within the ejector and the evaporator as well as the throttling valve mostly come from themselves. Moreover, 35% of the overall exergy destruction can be avoided, indicating the realistic potential for improvement. To improve system performance, the ejector should be firstly considered, then the condenser, followed by the generator and evaporator. The pump and throttling valve are the last ones to be considered.

The thermoeconomic analysis pinpoints that the costs of the brine side fluids are very important for the thermoeconomic features of the ejector refrigeration system, and is about 40% of the total cost at the considered condition. It should be noted that the thermoeconomic optimization does not necessarily lead to thermodynamic improvements. The analysis shows that the total cost is about 44% lower if the brine side fluids are free.

7.1.2 Experimental study

The experiment only focuses on the vapor ejectors. Six ejectors have been tested. However, the desired test conditions could not be reached in spite of the large effort spent. The reason for this is thought to be the design of the ejectors. As noted above, minor differences in the design, like surface roughness, shape of edges, lengths of the sections etc., are important for the performance. Additionally, the pressure levels were difficult to control independently. The obtained experimental results are therefore partly difficult to evaluate.

The measured ejector data is dominated by the primary flow pressure. An increase in the primary flow pressure results in the increase of the secondary flow pressures and the ejector outlet pressure. The obtained pressure lift ratio and the entrainment ratio vary from 1.15 to 1.57, and 0.14 to 0.31, respectively.

The effect of superheat of the secondary flow on the entrainment ratio is insignificant, confirming the theoretical results. The validities of neglecting the velocities at the ejector inlets and outlets in the ejector modeling are examined and confirmed.

The ejector efficiencies are studied with assistance of the measured data and the proposed ejector model. The nozzle efficiency varies from 0.853 to
The mixing efficiency and diffuse efficiency are in the ranges of 0.800 to 0.948 and 0.695 to 0.763, respectively. The ejector efficiencies depend on the operating conditions and the ejector dimensions.

7.2 Suggestions for future work

The model presented in Paper I involves ideal gas assumptions. Using the real gas behavior to formulate all the ejector processes is a good way to eliminate the errors from ideal gas assumption.

In the modeling of the ejector refrigeration system, the proposed ejector model only considers the ejector area ratio. This is clearly not enough. A more sophisticated ejector model needs to be developed to investigate the effect of operating conditions on the other ejector geometries, like the divergent angle and length of nozzle, nozzle exit position (NXP), the convergent angle and length of the constant-area in the mixing chamber, as well as the angle and length of the diffuser, to guide the ejector design. CFD can be very helpful to explore flow details in a fixed-geometry ejector.

The ejector efficiencies are known as very important parameters in the ejector models, and largely determine the accuracy of the model predictions. They are closely related to the operating condition and the ejector geometrical features, and could be quite different from case to case, refrigerant to refrigerant, making the determination of these ejector efficiencies very difficult. The ejector efficiencies need more wide and deep study theoretically and experimentally.

The suggested work above can be eventually assigned to the detailed guideline for the ejector geometry design, which is still a remarkable gap that needs to be filled, especially for refrigerants and variable-geometry ejectors. The ejector design is never an easy task.

The advanced exergy analysis provides very informative insights of the ejector refrigeration system. It should be further used to study the system features under different operating conditions, and to compare the results of different refrigerants at the same conditions. Moreover, the recently developed methods, for instance advanced thermoeconomic analysis and exergoenvironmental analysis (Tsatsaronis, G., 2011), are of interest to apply to the ejector refrigeration system in order to improve the system performance from the economic and the environmental (ecological) point of view.
Regarding the experiments, it definitely needs to be further improved. On the one hand, all the ejectors require careful redesign and remake using fine manufacturing techniques. Due to the low mixing and diffuser efficiencies, the ejector main body (see Figure 4-2) should be firstly considered to be improved, for instance, (1) the convergent part of the mixing chamber should be enlarged to give more space for the secondary flow passing through and to reduce the losses, (2) a smooth transition from the suction chamber to the mixing chamber, e.g. a proper radius at the entrance of the mixing chamber, (3) the inlet of the secondary flow could be moved a little further from the entrance of the mixing chamber and the diameter could be larger (Eames 2013). For the nozzle N3-N6, there is a 2 mm constant-area at the throat, which is needed to facilitate the manufacturing. This could be eliminated, and it would be better to have rounded corners instead of the angle edges inside the nozzle. Furthermore, an ejector with a moving spindle in the nozzle or a moving nozzle can be investigated to obtain the features of a variable-geometry ejector.

On the other hand, the test bench can be improved as well from the perspectives that (1) the condenser should be moved before the flow is divided into two paths, so the ejector outlet pressure (back pressure) could be more effectively controlled, (2) real throttling valves have to be appropriately chosen to replace the needle valves that is used as an expansion device in Figure 4-3 in order to effectively reduce the pressure after the relocated condenser and to achieve the desired low pressure for the secondary flow. As a result, cooler 1 has to be an evaporator, (3) a heat exchanger should be added before entering the nozzle to regulate the superheat of the primary flow, (4) to accommodate the installed compressor, diameters of the nozzle throat can be a little smaller in order to achieve a higher pressure as well as a wider range of operating conditions.
# Nomenclatures

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>$A$</td>
<td>area (m$^2$)</td>
</tr>
<tr>
<td>$Ar$</td>
<td>ejector area ratio</td>
</tr>
<tr>
<td>$\dot{C}$</td>
<td>cost rate of exergy ($\text{$} \cdot \text{h}^{-1}$)</td>
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<tr>
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<td>coefficient of performance</td>
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<td>$D_{th}$</td>
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<td>velocity (m·s$^{-1}$)</td>
</tr>
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<td>$U$</td>
<td>uncertainty</td>
</tr>
</tbody>
</table>
\( V \)  
volume (m³)

\( W \)  
work (kJ)

\( \omega_s \)  
systematic error

\( Z \)  
cost rate related capital investment and operating and maintenance expense (\$/h⁻¹)

\( Z_r \)  
compressibility factor

\( y \)  
exergy destruction ratio

**Greek**

\( \varepsilon \)  
exergetic efficiency

\( \eta \)  
efficiency

\( \theta \)  
angle (°)

\( \mu \)  
entrainment ratio

**Superscripts**

AV  
avoidable

CH  
chemical

CI  
capital investment

EN  
endogenous

EX  
exogenous

KN  
kinetic

M  
mechanical

OM  
operating and maintenance expense

PH  
physical

PT  
potential

T  
thermal

UN  
unavoidable

*  
critical

**Subscripts**

\( c \)  
condenser

\( c,i \)  
condenser inlet

\( c,o \)  
condenser outlet

cal  
calculation result

Cannot  
Carnot cycle related
CO condenser
d diffuser
e evaporator
e,i evaporator inlet
e,o evaporator outlet
EJ ejector
EV evaporator
exp experimental data
g generator
g,i generator inlet
g,o generator outlet
GE generator
i ideal process
in inlet
j exergy carrier positions
k the kth component
m mixing
min minimum
n nozzle
out outlet
p the primary flow
PU pump
ref reference
s the secondary flow
sh shock
sat saturation
th throat
tot total
TV throttling valve
0-5 ejector locations in Figure 2-1
2’ and 4’ and ci’ ideal ejector processes in Figure 2-2
References


Eames, I.W., Personal communication, Oct 27, 2013.


Paper I
Paper II
Paper III
Paper IV