

THE BINARY FLUID EJECTOR REFRIGERATING SYSTEM FOR AIR CONDITIONING APPLICATION

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THE BINARY FLUID EJECTOR REFRIGERATING SYSTEM FOR AIR CONDITIONING APPLICATION

A DISSERTATION SUBMITTED IN PARTIAL FULFILLMENT OF THE
REQUIREMENTS FOR THE AWARD OF THE DEGREE OF

DOCTOR OF ENGINEERING

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Summary

Air conditioning is one of the most dynamic areas of refrigerating technologies while remains high energy-intensive.

Today, 90% of climate control equipment belongs to vapor compression systems that consume electricity. At the same time, the value of cold at this level of temperatures is low. Specific exergy at 7°C equals to 0.082. Slightly higher (0.09) is exergy of heat required for space heating. Thus, cooling and heating shall not rely on electricity or high-grade heat but shall use an affordable low-grade heat. Such an approach will define a widespread transition to heat-utilizing thermotransformers. That fact substantiates the relevance and practical value of this work.

The choice of thermotransformers today is limited by sorption and jet systems, where the cycles of heat conversion to cold or anergy into heat are realized. High-grade heat-driven power supply systems for space heating or cooling application do not represent reliable approaches unless exergy of this heat is wholly utilized for a combination of consecutive abovementioned services production. Market attention is currently paid to sorption chillers or heat pumps, while ejector heat pumps were, until recently, unclaimed. Many studies have resolved the critical issues of ejector systems that sharply increased market interest to them. Promising, in particular, are binary fluid ejector refrigeration systems (BERS), this work is devoted to.

This thesis provides a comprehensive justification of the criteria for selecting the fluid components to form the zeotropic mixture applied in BERS.

- The effect of several thermodynamic properties of fluid components on entrainment ratio and COP was studied.
- CFD research of binary fluid ejector led to practical algorithm development for optimal ejector geometry calculation was conducted.
- Preliminary calculation and design based on empirical velocity coefficients, CFD modeling and sequential variation of dimensions to establish a steady flow without turbulent eddy and axial deviations of the jet were among the algorithm's steps to obtain a maximum entrainment ratio, final design and manufacturing recommendations.
- Analysis of ejector operating at off-design conditions and identification of compensation methods to maintain the efficiency of the system by varying mass fractions and operating parameters was provided.
- Theoretical and experimental research of energy and exergy characteristics of BERS defined optimal operating parameters for air-conditioning and refrigerating systems and its combined schematic solutions, operating with R1233zd(E)/Butane binary fluid.
- Test results of industrial thermovacuum drying systems were achieved and correlated by the method described in this work for steam/air binary fluid.
- Schematic solution with the application of binary and multi-component fluid heat pumps were developed in the presented work. Those solutions can be applied for system's components production, exhaust heat utilization at gas or coal power generating plants, multiple services generation systems, transport systems, commercial and industrial drying technologies, gas liquefaction, fire extinguishing systems, etc.

This thesis also initiates an analysis and formulates the preliminary conclusions on the following statements:

1. Selected criteria of binary fluid components analysis for BERS were proposed;
2. An approach of ejector performance compensation operating at off-design conditions was developed and validated;
3. Exergy analysis was conducted to obtain the designed parameters for heating and cooling systems;
4. Practical verification of CFD model on multiple embodied ejectors proved the correctness of the selected calculation and modeling approaches with an error not exceeding 5%.

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I want to express sincere gratitude to my supervisor, Prof. Takahiko Miyazaki, for the patient guidance, advice, and great support during the research for the past three years.

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Acronyms and Glossary

BERS	Binary Fluid Ejector Refrigeration System
CFD	Computational Fluid Dynamics
COP	Coefficient of Performance
EER	Energy Efficiency Ratio
ERS	Ejector Refrigeration System
GTI	Gas Technology Institute
JTT	Jet Thermo Transformers
TDVS	Thermo-vacuum Drying System
VCRS	Vapor Compression Refrigeration System

Nomenclature

D	Exergy Destruction, W
E	Exergy, W
f	Cross section area, m ²
G	Mass flow rate, kg/s
h	Specific enthalpy, kJ/kg
k	Adiabatic index
K_1, K_2, K_3, K_4	Integrated velocity coefficients
l	Specific work, kJ/kg
L	Work, kW
P	Pressure, Pa
θ	Trouton's constant, J/(mol K)
r	Specific evaporation heat, kJ/kg
R	Specific gas constant, J/(kg K)
s	Specific entropy, kJ/(kg K)
T	Temperature, K
t	Temperature, °C
U	Entrainment Ratio
V	Volume, m ³ /kg
w	Velocity, m/s
X	Mass fraction of working fluid in mixture
Z	Compressibility factor
γ	Relative mass velocity
ε	Carnot efficiency of reverse cycle
ζ	Thermal efficiency of the system
η	Carnot efficiency of direct cycle
λ	Relative velocity
Π	Relative pressure
ρ	Density, kg/m ³
$\varphi_1, \varphi_2, \varphi_3, \varphi_4$	Experimental velocity coefficients

Subscripts

<i>'</i>	Working flow expansion parameter
<i>*</i>	Critical parameter
<i>A</i>	Nozzle outlet cross section area
<i>B</i>	Cylindrical mixing chamber inlet cross section area
<i>C</i>	Cylindrical mixing chamber outlet cross section area
<i>comp</i>	Compression
<i>cond</i>	Condensation parameters
<i>ej</i>	Ejector
<i>e.v.</i>	Expansion valve
<i>eva</i>	Evaporation parameters
<i>exp</i>	Expansion
<i>gen</i>	Generation parameters
<i>in</i>	Input energy
<i>mix</i>	Mixed flow
<i>out</i>	Output energy
<i>rf</i>	Refrigerant fluid flow
<i>theor</i>	Theoretical value
<i>wf</i>	Working fluid flow

CHAPTER 1

Chapter 1. Introduction: Current state-of-the-art review on ejector technologies and analysis of efficiency enhancement criteria for Ejector Refrigerating Systems (ERS)

Air conditioning (A/C) has become an ultimate feature and an increasingly important life support necessity, demonstrating 9.3% of year-to-year growth and reaching the global A/C market size of 130 mln. units in 2018. Most of the commercially available systems based on the electrical vapor-compression technology saw up to 20% of the overall electricity consumption in residential and commercial buildings, accounting for over 500 mln. tons of indirect CO₂ emissions from power generation for A/C needs. Peak power consumption is observed in the summer period, which causes a grid load increase by 25-40%, while the efficiency of power generators falls by about 5-10% of its nominal value. In addition, power consumption by vapor-compression air-conditioning systems leads to unjustified losses from the internal and external irreversibility and along with its seemingly high efficiency (EER of 10.2-13.3) become the most serious factors of severe environmental load and climate change. The level of technical excellence of vapor-compression air conditioners has already reached its limit; therefore the only replacement of outdated systems for the game-changing technologies shall serve for energy mix sophistication at buildings and dwellings worldwide.

Alternative A/C technologies are often considered as less efficient and less reliable or much expensive, capacious, and maintenance-intensive solutions. On

the other hand, the all-growing utilization of affordable waste heat or renewable energy, speaks in favor of its **strong integration potential** with the space cooling technologies targeting the higher energy efficiency, cost-effectiveness, flexible to part-load performance, customer-friendly and competitive with the conventional vapor compression technologies.

The **ejector-based technologies** represent one of the promising variants of integration of the low-grade heat potential as a driving force for air-conditioning, refrigeration, heating, and power generation services. The interest of the researchers to ejector technologies is increasing from year to year, which is represented in Figure 1.1.

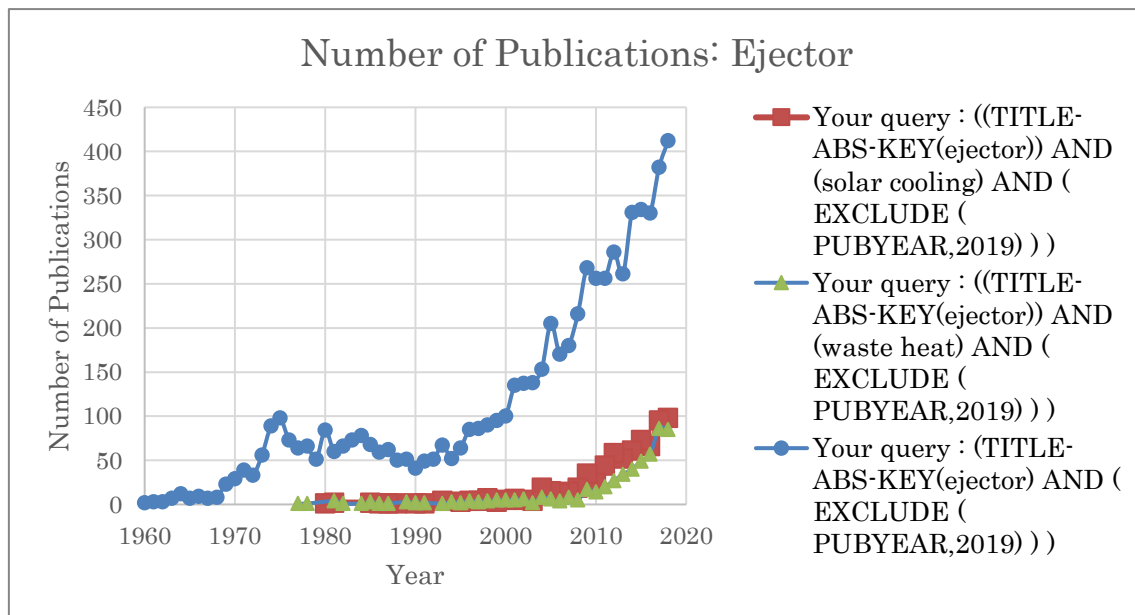


Figure 1.1 Represent a number of publications related to ejector technologies (Scopus).

1.1 ERS as a new generation of thermo-transformation systems - a survey of modern literature.

The first application of the jet refrigeration system was described in 1884. In 1902, Charles Parsons worked on Steam jet cooling system [1], in 1905 Maurice Leblanc built the first machine in 1907 and received a US patent in 1911 [2].

First machine operating on refrigerants was tested in 1928 by Prof. Wilson at University of Florida, USA. The second half of 20th century was characterized by finding solutions that could significantly improve the performance of ejector-based cooling technologies by applying new refrigerants and schematic solutions, improving the flow part of the ejector, utilization of the renewable heat (solar and geothermal), and offering various cost-effective and reliable ways to pump the nearly saturated refrigerant [3-15]. Most of the studies on ejector technologies were carried out for steam and available refrigerants as a working media, but the efficiency gain was still insignificant to consider technology as an emerging one that time.

Though several industries like chemical, aerospace, metallurgy, etc. are intensively applied ejectors in their commercial portfolio, such applications as cooling, heating and power engineering is yet to be commercial available and limited only to sporadic steam-water ejector air-conditioners, the production of which is mostly focused on navy and other military purposes (nuclear submarines space cooling for example) that can sacrifice the low efficiency in favor of highest safety.

An extensive application of low-boiling refrigerants in the ejector-based cooling systems was originated in 1954 by Sergey Zhadan, a Ph.D. student of Prof.

Martynovsky. Dr. Zhadan conducted many experimental tests while studying **Ejector Refrigerating Systems (ERS)** operating on R-12. In 1969, Dr. Akhmadiyar Davletov, under the guidance of Prof. Martynovsky and Dr. Zhadan, introduced in a first time a **Solar Thermal Driven ERS** at the Academy of Sciences of Turkmenistan (Scientific Production Association "Solntse"). In 1971, Dr. Larysa Krasnyuk defended a Ph.D. thesis on residential ejector-based refrigerators with thermopumps of bellows-sealed and lever types [10]. In 1978, Volodymyr Petrenko, a Ph.D. student of Dr. Zhadan, defended a Ph.D. thesis on ERS theoretical and experimental study on R-142 in air conditioning mode, attempting to find the scope of ERS application using a waste heat source from the foundry process [12]. As a result of these consequent studies, the ERS application potential and validated operational conditions were outlined. Unfortunately, the low-performance characteristics (COP below 0.4) and significant energy losses in ejector could not promise a prosperous future to ERS and slow down its development for many years in favor of the main competitor - sorption refrigeration machines (COP above 0.6).

The restoration has come only in the last decade of the 20th century when the new schematic approaches have come out, and the optimized ejectors' geometry greatly improved its performance. The ERS test series at the Rogbane Research Center (Conakry, Guinea), Nottingham University and the University of Taipei renewed interest in ERS operating with various refrigerants [16-20], but the challenge of refrigerant supply to the vapor generator and operation at the off-design conditions remained the major drawbacks to keep these systems away from the mass market. In 1991, the Rogbane-Conakry Research and

Development Center came out with a hypothesis to regulate the operating parameters at off-designed conditions by compensating it with another one, which was later described by A.S. Volovyk [21]. At the same time, the attempts to "feed" the steam generator of ERS were studied in Australia, India, Singapore, Thailand, Sweden, Poland, and Belgium [22-35]. Many researchers still believe that the principal reserves of increasing the efficiency of the ERS are linked with a geometry of the flow profile of the ejector parts [36-40]. However, all the attempts to complicate the design of the ejector's flow part did not add any significant value to the ejector's performance [41-46].

In the review paper of Prof. Saffa Riffat [31], referring the latest achievements of Dr. Kaspersky [48] and Prof. Shen et al. [49], the ERS schemes are considered operating without a pump (Fig. 1.2a) and the ERS with a gas-liquid ejector or injector (Fig. 1.2 b).

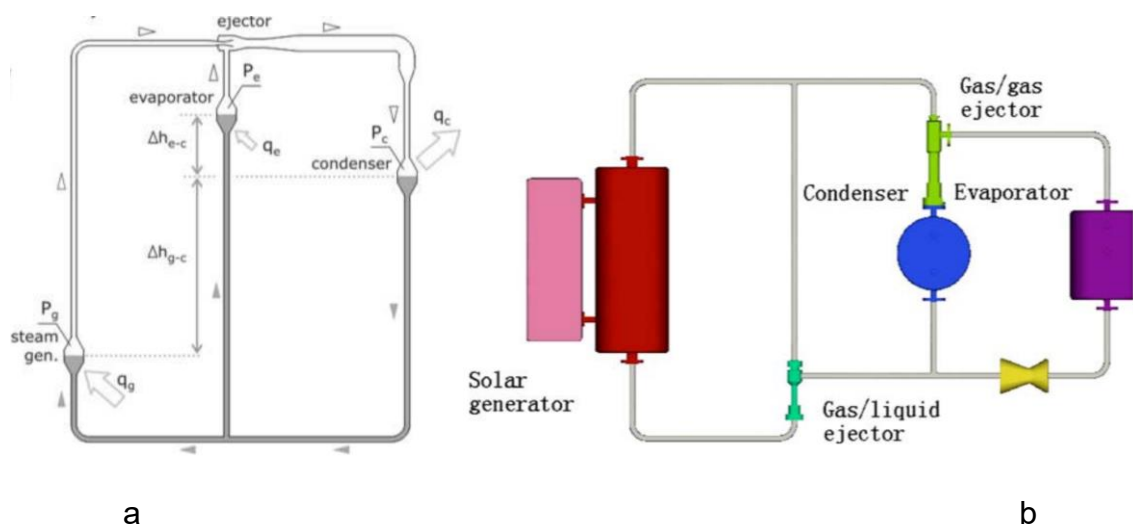


Figure 1.2 Schematic diagrams of a) pumpless ERS using a condensate-generator [48] height difference. Δh_{e-c} is the difference between the levels of the liquid in the evaporator and the condenser; Δh_{g-c} is the difference between the levels of the liquid from the generator and the condenser; b) ERS with an injector as a pump [49].

The first schematic requires a height difference of 50 to 700m, which allows its application only for high-rise constructions. With such differences in heights, pressure losses in the steam pipe and hydraulic resistances in the liquid pipeline affect the system performance. Except for the pump, this schematic does not include a throttle valve as the liquid refrigerant is supplied from a bottom level to the evaporator on an upper level, so the pressure is lost by overcoming the hydraulic resistances.

The schematic solution employing an injector as discussed in the dissertation of Nadia Shchetinina [11] back in the 1980s was appeared nonoperational since subcooling of the liquid was required on a suction line of the injector in order to condense the vapor coming from the nozzle. This subcooling is equivalent to an elevation height of 20-30 meters. An integrated solution was proposed later combining the gravitational and injector components [48].

Another pumpless ERS was proposed by Prof. Lehmus [9] and Prof. B.J. Huang (Fig. 1.3) [16, 51]. By including an additional vapor generator and alternating charge/discharge functions between them on practice appeared too complicated due to cycle time delay, quick heat transfer between the nearly saturated refrigerant liquid and the hot wall surface of the generator, increases the overall refrigerant charging capacity, increased mass-dimensional characteristics, installed and maintenance costs, etc.

The gravitational thermopump on the water and R-141b were tested by Prof. Chen [52], as well as Prof. Satha Aphornratana [53, 54] from Thailand (Fig. 1.4)., Temperature fluctuations, absence of pressure-equalizing line and inability for stable operation made those attempts unreliable and ineffective, therefore

authors concluded that such thermopump required more studies. Fundamentally, the loops of such thermopumps were similar to those developed by Dr. Olexiy Buyadgie (Ph.D. thesis), but a number of simple features, like organization of liquid supply and discharge, high-quality insulation, pressure-equalizing lines resulted in a stable functioning of thermopump, while experimental validations test for ERS on R-142B were conducted.

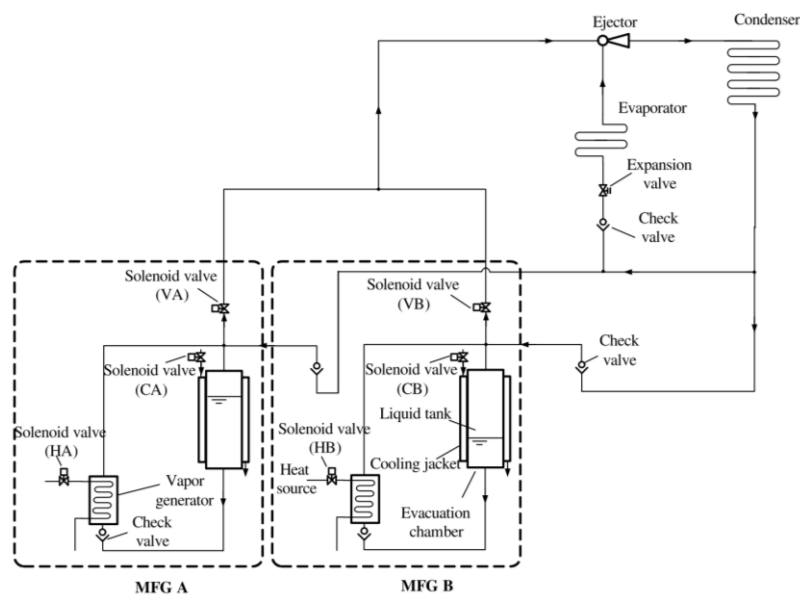


Figure 1.3 Diagram of a non-suction-type electrochemical generator in a multifunctional generator [51]. MFG - multifunctional generator.

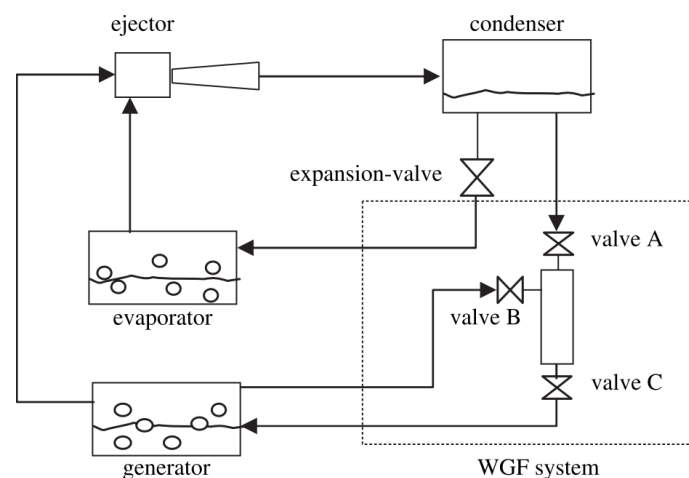


Figure 1.4 ERS scheme with gravity-type pump [54].

Many researchers continue to search for ways to improve the efficiency of the ejector, which mainly consist of the following:

1. The theory of an ejector with mixing at constant pressure was developed and used by Keenan et al. [55, 17, 56-62]. It was assumed that the mixing of the working and ejected flows occurs at constant pressure. Keenan conducted mathematical analysis and experimental research. One of the problems was the optimal shape of the mixing chamber. Sokolov and Zinger [63] determined that the conical receiving chamber has higher speed coefficients. Based on the works of Sokolov, Wei [64] added the method of calculating and analyzing the ejector taking into account the impact losses.

Khan [65] suggested that the velocity coefficients of the nozzle, diffuser and mixing chamber are not fixed values, as in Sokolov's works. They should vary depending on the design of the ejector and the cycle parameter. At the testing of a steam ejector, Shen et al. [66] determined that it performs a small compression work if the diameter of the mixing chamber exceeds the theoretical or if the distance from the nozzle to the mixing chamber is less and far from the cylindrical chamber the working parameters deteriorate sharply. El-Dessouki [59] developed a semi-empirical model that determines the entrainment ratio as a function of the expansion rate and pressures of the ejected, working, and compressed flows. Besides, he introduced the refinement of the pressure at the nozzle outlet as a function of evaporation and condensation pressure, the ratio of cross-sections as a function of ejection coefficient and vapor pressure. Huang [17] introduced two empirical corrections based on the calculated characteristics of the ejector on R141b, obtained as a result of testing 15 ejectors. Performance took into account

the ratio of the estimated cross-section of the ejected flow to the critical cross-section of the nozzle f_{out}/f_{cr} , the ratio of the cross-section of the output cross-section of the mixing chamber to the cross-section of the nozzle f_{mix}/f_{cr} and the ratio of generation pressure to evaporation pressure P_{gen}/P_{eva} and critical pressure at the outlet from the diffuser to evaporation pressure P_{cond}/P_{eva} . The error in calculating the ejection ratio is +/-10%. Ouzzane [67] developed a mathematical model for calculating injectors, based on the properties of real gases and conservation laws, using NIST routines to determine the properties of substances. The model accuracy of the experimental research of Huang et al. [60] was 6% for the entrainment ratio and 8% for the saturated steam temperature corresponding to the critical backpressure. Valle [68] proposed a method for calculating the ejection coefficient using the properties of real gases, computational fluid dynamics, and numerical solution methods.

2. In comparison with the experimental data of Huang, the absolute average error of calculations was less than 7%. It means that the results of calculations on a real substance are close to the experimental ones and can be used to calculate other injectors.

After conducting their research, Keenan et al. [57,58] found that an ejector with a constant mixing pressure has better performance than an ejector with a constant mixing cross-section. In this regard, the study of such ejectors temporarily stopped. Fabri et al. [55] found that in the process of mixing two flows in the injector $M_{rf} < 1 < M_{wf}$, the back pressure does not always affect the flow. Subsequently, he expanded this idea to a supersonic ejected flow. The results show that the operating parameters depend on the pressures of the working

stream at the nozzle outlet and the pressure of the ejected stream. Yan [69] found that mixing at a constant pressure does not always give better results than with a constant mixing section.

Partial differential equations for the flow can be obtained by establishing a two-dimensional model using differential methods. It will help to more accurately describe the flow in the mixing chamber than the linear model. Coff et al. [70] analyzed the mixing process using the free jet theory. Guo et al. [71,72] considered the effect of viscosity. Their studies included a large number of tests using statistical methods for determining the function of the velocity curves. They determined the approximate model of velocity distribution, the length of the free jet, and the pressure in the mixing chamber. Zhang et al. [73] established a two-dimensional axisymmetric compression model and analyzed the characteristics of the ejector at various operating pressures. Studies have shown that the ejection coefficient begins to fall at high pressures of the working stream due to jumps in the mixing chamber. Low suction pressure can cause a backflow in the receiving chamber; this can affect the safety of the system. Zhu et al. [74] adopted a two-dimensional function to approximate the velocity distribution in the ejector. It is based on the velocity distribution in the pipe and introduces a critical section at the entrance to the mixing chamber. Compared with one-dimensional models, the two-dimensional velocity distribution function gives more accurate results.

For a more accurate and comprehensive analysis of the interaction of gases, some researchers [75,76] tried to explain the processes of outflow and mixing, using software packages for solving problems of computational hydrodynamics.

3. The transition from one-dimensional and two-dimensional model to three-

dimensional modeling; Riffat et al. [75] analyzed the three-dimensional model of an ammonia ejector in 1996. But the incompressible flow cannot be compared with the real flow, because the equation describing the compressible flow was very complicated. Rasley et al. [76] modeled the three-dimensional flow inside the ejector on R245. In this study, a compressible real gas model was implemented on a large number of grid elements. The result provided a good imitation of the processes inside the ejector, including an expansion of the working flow and thermodynamic shock waves. Bartosiewicz et al. [77] analyzed six turbulence models for the study of ejectors. The analysis focused on the location of shock waves, their strength, and prediction of pressure recovery. Hemidi et al. [78,79] compared the CFD model with the experimental results. It showed that the determination of the basic performance parameter is not sufficient for a proper assessment.

4. Han's assumption that the velocity coefficients of the nozzle, diffuser and mixing chamber are not fixed values, but vary depending on the design of the ejector and the operating parameters of the cycle, have been hypothesized about the influence of the thermophysical properties of the working substances on these quantities.

These theoretical hypotheses lacked many assumptions that are not so obvious but necessitate verification.

5. Regarding the impact loss, it was necessary to find out how long it was necessary to reduce it in order for the integral result to be the best. According to the speed coefficients, it was necessary to determine their value for different substances, which is the main difference that most influences the ejection

coefficient.

6. It was also required to create a universal mathematical model for calculating the ejector on any substances and their mixtures, as well as to confirm it and check it on a three-dimensional computer simulation.

ERS schematic design and Operating Principles

Schematic and process diagrams of the Ejector Refrigeration System represented in Fig. 1.5, and Fig. 1.6.

The ejector is a compression device that does not contain moving parts. Fig. 1.7 represents a schematic of an ejector.

The system contains a vapor generator, evaporator, condenser, ejector, pump. Supply heat is utilized in vapor generator, where working fluid evaporates at high temperature and high pressure. Vapor from vapor generator flows through convergence/divergent ejector nozzle where pressure converts to velocity. Working flow expands to the lowest pressure level in the system, i.e., evaporation pressure. In the suction chamber, accelerated working flow entrains low-temperature secondary flow from the evaporator. From suction chamber fluid flows to a cylindrical mixing chamber, where two flows are mixing and equalize parameters. Mixed flow from cylindrical chamber flows through a diffuser, where pressure recovers.

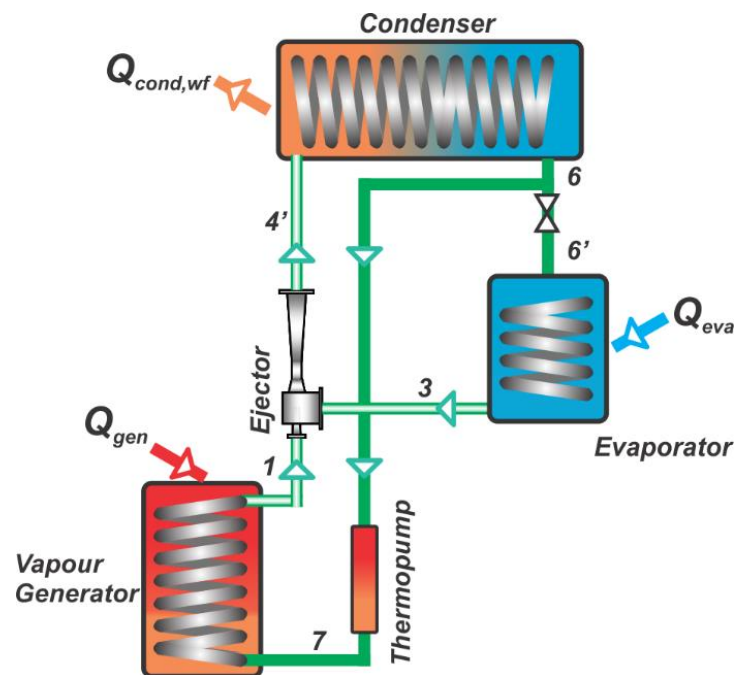


Figure 1.5 Schema of the ERS

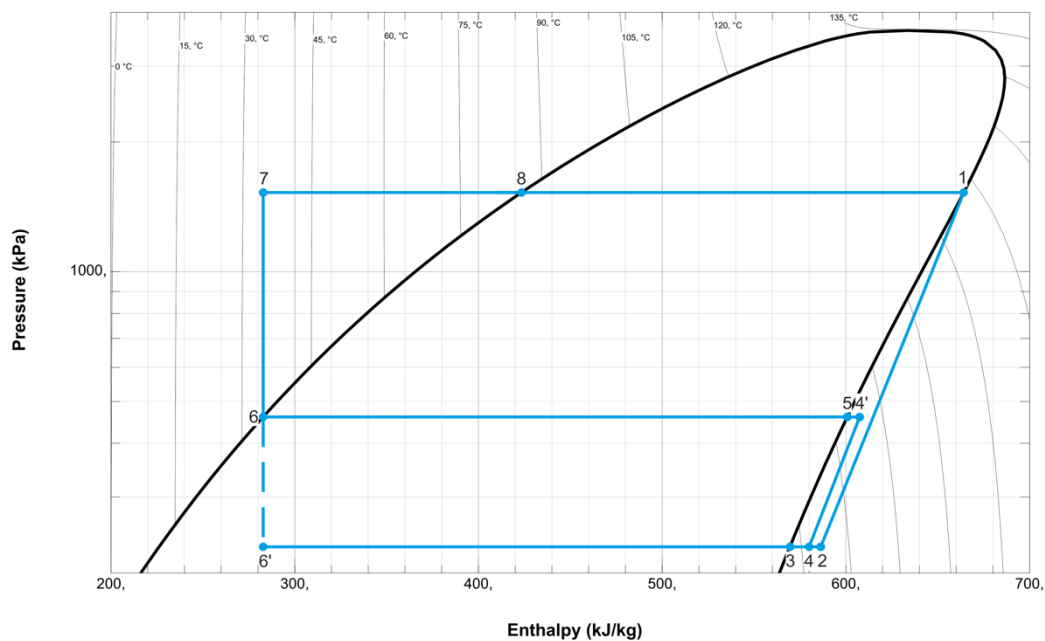


Figure 1.6 Operating diagram of ERS. 7-8-1 – heating and boiling in vapour generator; 1-2 – working fluid expansion in the ejector nozzle; 2-4 and 3-4 – working and refrigerant vapour mixing ; 4 – 4' – vapor mixture compression in ejector; 4'-5-6 vapour condensation; 6-6' - liquid throttling to evaporator; 6-7 liquid fluid feeding to the vapour generator; 6'-3 – refrigerant fluid evaporation in the evaporator.

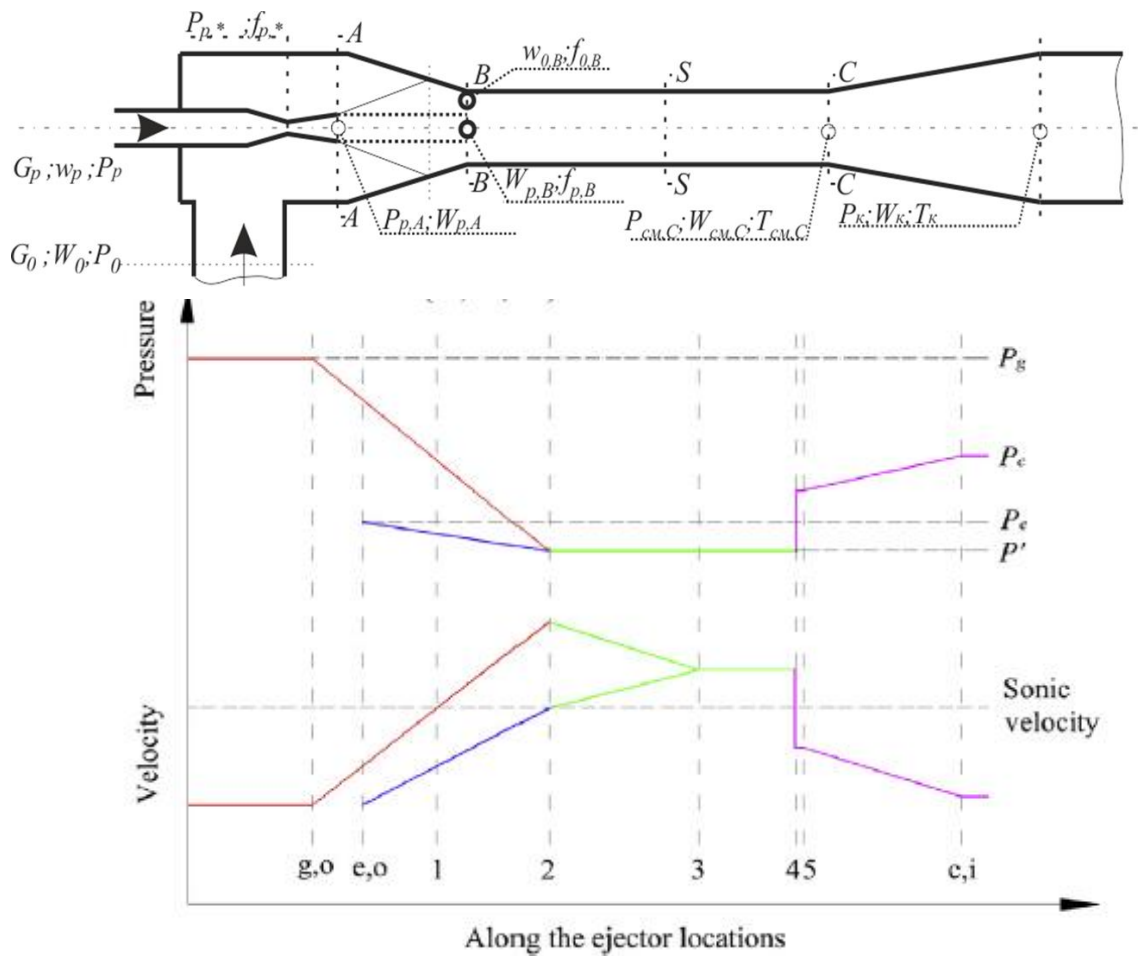


Figure 1.7 Schematic drawing of the Ejector and Pressure velocity change along ejector profile . A – Nozzle outlet, B – Mixing chamber inlet, C – Mixing chamber outlet.

1.2 Objectives of study

The main objective of the study is to provide an analysis of binary fluid properties for the ejector refrigeration system application by implementing the following tasks:

1. Define a set of properties that effects ejectors efficiency.
2. Develop an approach and methodology for accurate ejector efficiency and geometry evaluation that can be used for industrial manufacturing.
3. Provide a CFD modeling and analysis of fluid flow phenomena is ejector

flow part.

4. Provide an industrial verification ejector flow part design and efficiency.
5. Select a binary fluid mixture for air-conditioning application based on modern safety requirements.
6. Provide an exergy analysis of single and binary fluid ejector system.

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CHAPTER 2

Chapter 2. Theoretical analysis of binary fluid application in the ERS and particularities of the binary fluid ejector design.

2.1 Thermodynamic analysis of losses reduction in BERS (optimization of shock losses and heat exchange losses at variable temperatures).

Among the heat utilizing refrigeration systems operating with real fluids, expansion-compressor systems are the best from the point of thermodynamic perfection. A power cycle is the Organic Rankine cycles, and refrigeration is the reverse Rankine cycle. However, due to operating limitations have not found a wide application and serves as a reference heat utilizing refrigeration cycle. It should be noted that for ideal Carnot cycles, the efficiency of Carnot Power Cycle η using low-grade heat is low 0.1-0.2. At the same time, in air conditioning mode Carnot efficiency of cooling cycle ε is high, reaches 7-10. The efficiency of heat utilizing cooling systems is defined by Eq. 2.1,

$$\zeta = \eta \varepsilon \quad (2.1)$$

i.e. may reach values 0.7-2. The actual efficiency of the expansion-compressor cycle is about 0.5-1.2.

Thus, providing analysis of cold production methods in heat utilizing systems, a reference cycle is identified, i.e., cycle where efficiency depends on thermodynamic and thermal properties of fluids. Considering that the expansion and compression processes are adiabatic, then the energy characteristics of the cycle depend on the pressure and density ratios and active and passive flows.

An ideal case of energy exchange between active and passive flows is expander compressor system, where expansion and compression are provided without the direct interaction of flows. Schema is represented on Fig. 2.1

In jet devices, where flows interact, especially in the two-phase area, energy characteristics decreases significantly. It is connected to the need to expand active flow to lowest pressure in cycle and then compress working and secondary flows to condensation pressure.

Conditions of the expansion and compression are unequal. As a result, compression from evaporation pressure to condensation requires more energy that is produced by working flow expansion from generation to evaporation pressure range. During flows mixing in ejector suction chamber, the mixture is at intermediate parameters. Thus, compression is performed at different adiabatic curve than is located to the right than

expansion. The process is represented in Fig. 2.2

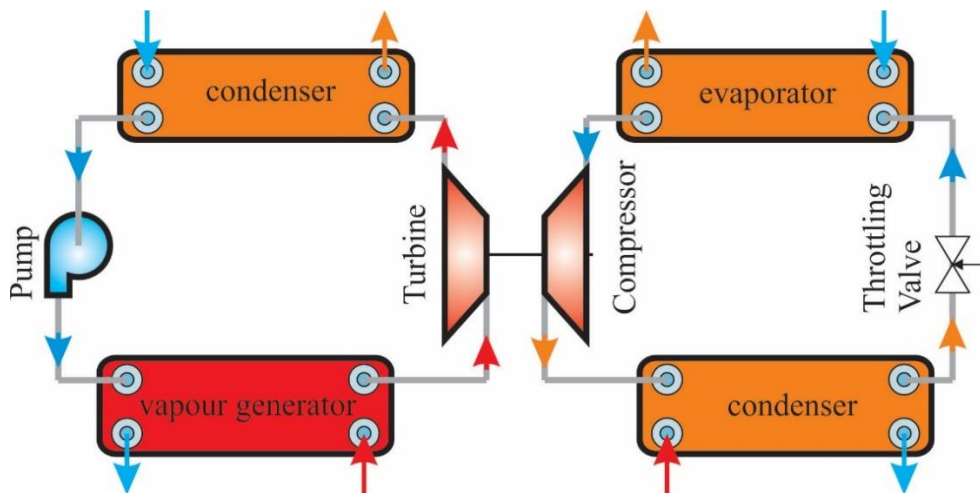


Figure 2.1 Schema of contactless Expansion-Compression System.

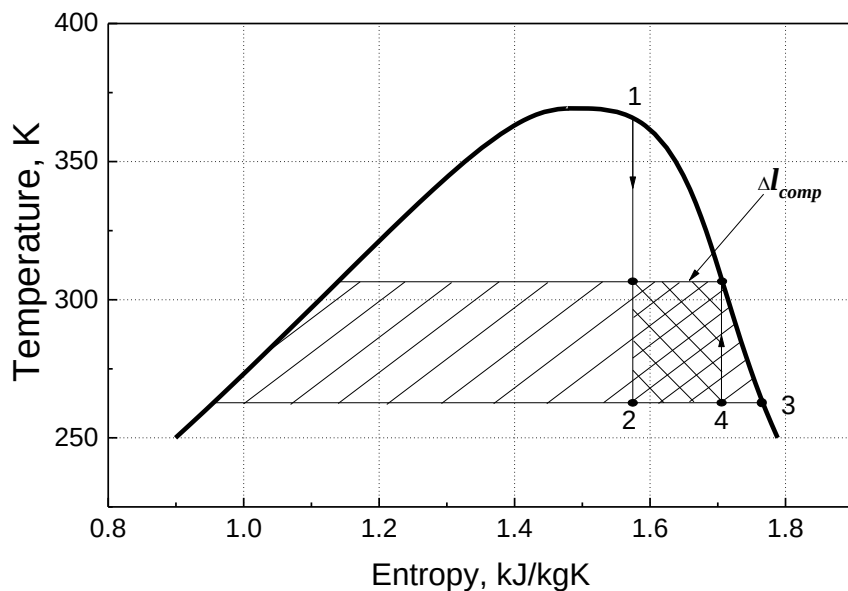


Figure 2.2 T-S diagram of theoretical expansion and compression processes in ejector. 1 – working vapour at nozzle inlet, 2 – working vapour outlet from the nozzle, 3 – refrigerant vapour from evaporator, 4 – theoretical mixed from is mixing process conducted at constant pressure.

Increasing entrainment ratio leads to the shift of compression adiabatic curve to the right. It increases compression work consumption.

As it can be seen from the above, the first and significant loss in ejector system comparing to expansion-compressor systems is a need of working flow expansion to evaporation parameters. It is well represented by entrainment ratios for two schematics:

a) expansion-compression cycle (Fig. 2.3 and 2.4).

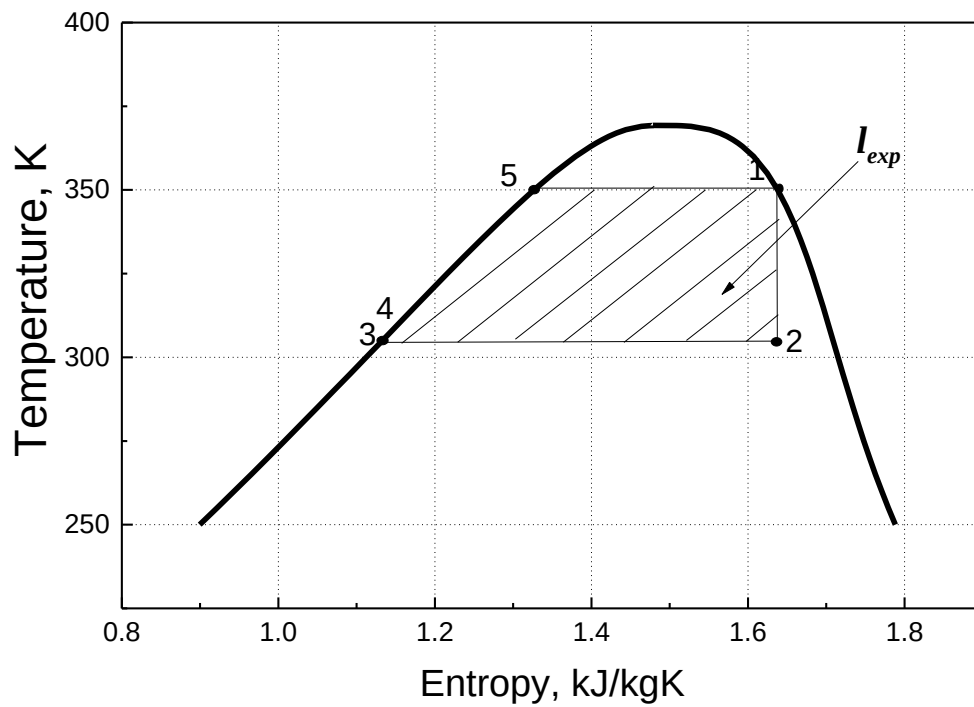


Figure 2.3 Expansion-Compressor cycle. a) T-S diagram of power cycle. 1-2 expansion in turbine, 2-3 condensation, 3-4 pumping into vapor generator, 4-5-1 heating and vapor generation.

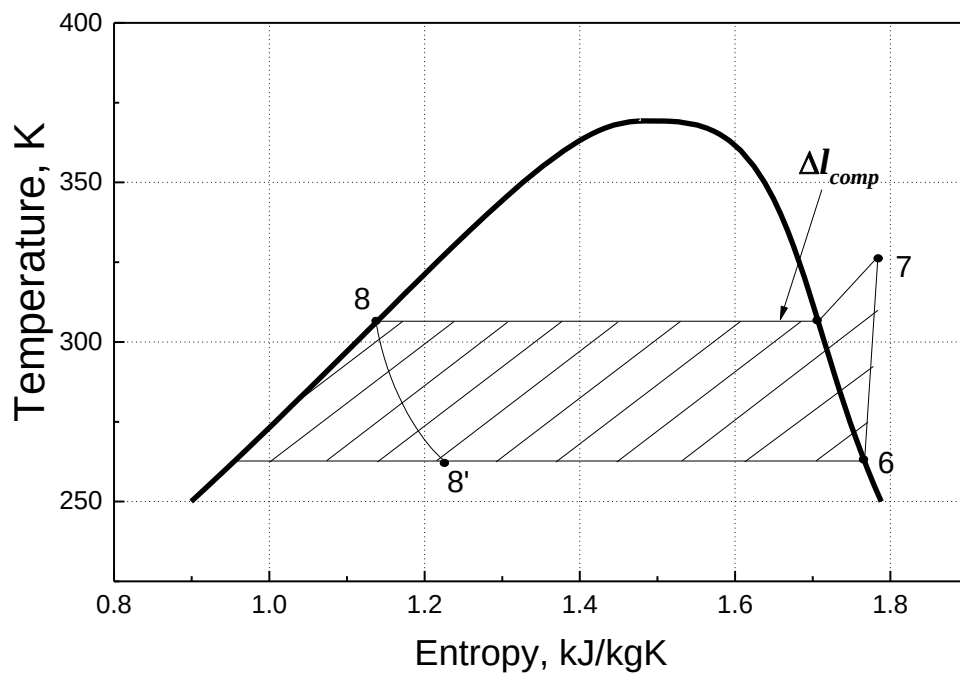


Figure 2.4 T-S diagram of refrigeration cycle. 6-7 compression, 7-8 condensation, 8-8' throttling.

Work balance for expansion-compression system defined by Eq.2.2:

$$L_{\text{exp}} = L_{\text{comp}} \quad (2.2)$$

Or

$$G_{\text{gen}} \cdot l_{\text{exp}} = G_{\text{eva}} \cdot l_{\text{comp}} \quad (2.3)$$

Eq. 2.4 represents an entrainment ratio evaluation:

$$U_{\text{theor}} = l_{\text{exp}} / l_{\text{comp}} \quad (2.4)$$

where

$$l_{\text{exp}} = \frac{P_{\text{gen}} \cdot V_{\text{gen}}}{k_{\text{gen}} - 1} \left[1 - \left(\frac{P_{\text{cond}}}{P_{\text{gen}}} \right)^{\frac{k_{\text{gen}} - 1}{k_{\text{gen}}}} \right] \quad (2.5)$$

$$l_{\text{comp}} = \frac{k_{\text{eva}} \cdot P_{\text{eva}} \cdot V_{\text{eva}}}{k_{\text{eva}} - 1} \left[\left(\frac{P_{\text{cond}}}{P_{\text{eva}}} \right)^{\frac{k_{\text{eva}} - 1}{k_{\text{eva}}}} - 1 \right] \quad (2.6)$$

b) ejector cycle (Fig. 2.5 and 2.6):

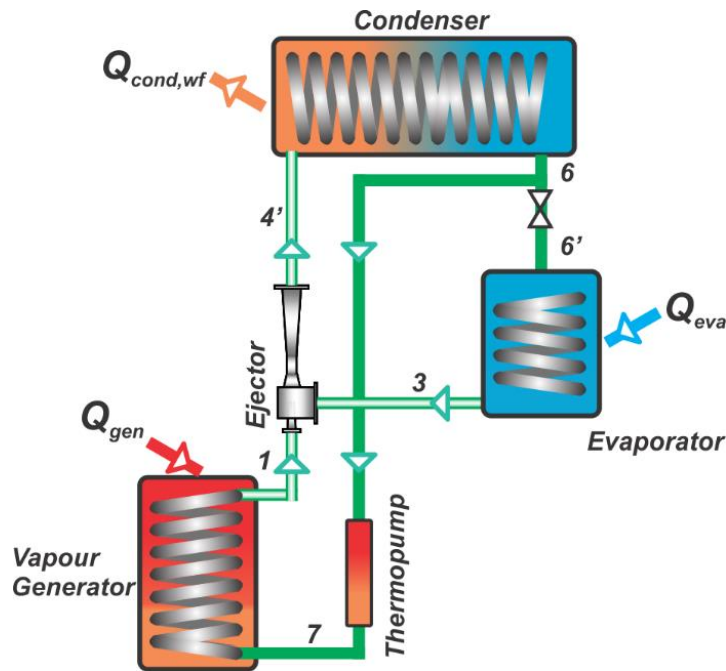


Figure 2. 5 Schema of ERS

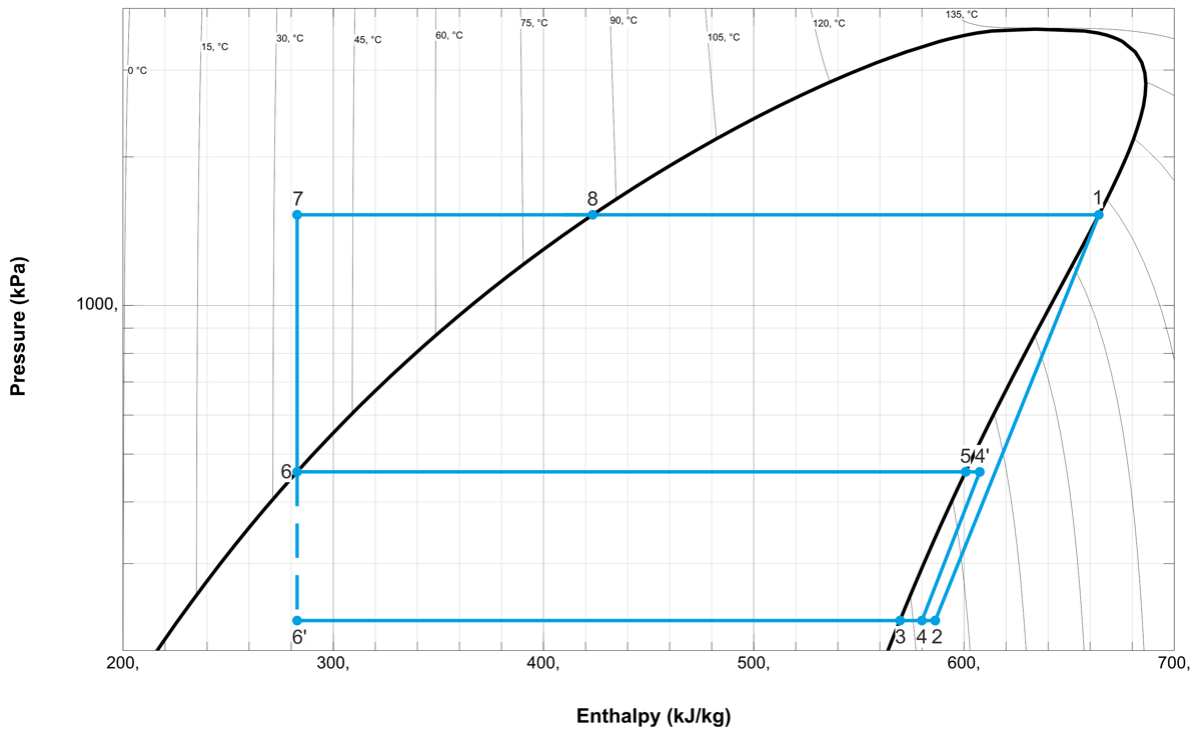


Figure 2.6 P-H diagram of processes in ERS: 7-8-1 – heating and vapour generation, 1-2 working vapour expansion in ejector nozzle, 2-4 and 3-4 mixing in suction chamber, 4-4' compression in ejector, 4'-5-6 mixed flow cooling and condensation, 6-6' – throttling to evaporator, 6-7 – liquid pumping to vapour generator.

Work balance for ejector is defined by eq 2.7

$$L_{\text{exp}} = L_{\text{comp}} \quad (2.7)$$

or

$$G_{wf} \cdot l'_{\text{exp}} = (G_{wf} + G_{rf}) \cdot l_{\text{comp}} \quad (2.8)$$

Entrainment ratio is defined by Eq. 2.9

$$U_{\text{theor}} = l'_{\text{exp}} / l_{\text{comp}} - 1 \quad (2.9)$$

where

$$l_{\text{exp}} = \frac{P_{\text{gen}} \cdot V_{\text{gen}}}{k_{\text{gen}} - 1} \left[1 - \left(\frac{P_{\text{eva}}}{P_{\text{gen}}} \right)^{\frac{k_{\text{gen}} - 1}{k_{\text{gen}}}} \right] \quad (2.10)$$

$$l_{\text{comp}} = \frac{k_{\text{eva}} \cdot P_{\text{eva}} \cdot V_{\text{eva}}}{k_{\text{eva}} - 1} \left[\left(\frac{P_{\text{cond}}}{P_{\text{eva}}} \right)^{\frac{k_{\text{eva}} - 1}{k_{\text{eva}}}} - 1 \right] \quad (2.11)$$

Another significant source of energy losses in jet devices is a loss of inelastic impact

of flows that is a principle of ejector operation [1,2]. This loss is proportional to a square of velocity difference of the flows.

$$\Delta E = \frac{U}{2(1+U)} (w_{wf} - w_{rf})^2 \quad (2.12)$$

$$w_{wf} = \lambda_{wf} a_{wf*} \quad (2.13)$$

$$w_{rf} = \lambda_{rf} a_{rf*} \quad (2.14)$$

$$\lambda_{wf} = \sqrt{\frac{k_{wf} + 1}{k_{wf} - 1}} \sqrt{1 - \Pi_{wf} \frac{k_{wf} - 1}{k_{wf}}} \quad (2.15)$$

$$\lambda_{rf} = \sqrt{\frac{k_{rf} + 1}{k_{rf} - 1}} \sqrt{1 - \Pi_{rf} \frac{k_{rf} - 1}{k_{rf}}} \quad (2.16)$$

$$\Pi_{wf} = \frac{p'_{eva}}{p_{gen}} \quad (2.17)$$

$$\Pi_{rf} = \frac{p'_{eva}}{p_{eva}} \quad (2.18)$$

Providing analysis of entrainment ratio reduction, taking into account shock losses, following assumption should be made:

Mixing process in suction pressure is performed at constant pressure.

Velocity equalization is taking place along the length of the suction chamber where pressure is constant.

Frictional and flow turbulization losses are neglected.

For flow mixing at constant pressure in a cylindrical mixing chamber, the following equations are valid.

Conservation of momentum (Eq. 2.19):

$$G_{gen} w_{wf} + G_{eva} w_{rf} = (G_{rf} + G_{wf}) w_{mix} \quad (2.19)$$

Mechanical energy conservation (Eq. 2.20) :

$$G_{wf} l_{exp}' = (G_{rf} + G_{wf}) l_{mix} + \Delta E \quad (2.20)$$

Kinetic energy conservation:

$$E_{in} = E_{out} + \Delta E \quad (2.21)$$

Kinetic energy of working and secondary flow at mixing chamber inlet cross section.

$$E_{in} = \frac{w_{wf}^2 G_{wf}}{2} + \frac{w_{rf}^2 G_{rf}}{2} \quad (2.22)$$

Kinetic energy of mixed flow at mixing chamber outlet cross section (Eq. 2.23 – 2.25):

$$E_{out} = \frac{(G_{wf} + G_{rf}) w_{mix}^2}{2} \quad (2.23)$$

$$U = \frac{l_{exp} - 2l_{comp} - \frac{(w_{wf} - w_{rf})^2}{2} + \sqrt{\left(2l_{comp} - l_{exp} + \frac{(w_{wf} - w_{rf})^2}{2}\right)^2 - 4(l_{comp}^2 - l_{comp}l_{exp})}}{2l_{comp}} \quad (2.24)$$

$$\frac{\Delta E}{E_{in}} = \frac{U \left(\sqrt{\frac{2kP_{gen}V_{gen}}{k-1} \left(1 - \Pi_{wf}^{\frac{k-1}{k}}\right)} - \sqrt{\frac{2kP_{eva}V_{eva}}{k-1} \left(1 - \Pi_{rf}^{\frac{k-1}{k}}\right)} \right)^2}{\frac{2U^2kP_{eva}V_{eva}}{k-1} \left(1 - \Pi_{rf}^{\frac{k-1}{k}}\right) + \frac{2kU}{k-1} \left(P_{gen}V_{gen} \left(1 - \Pi_{wf}^{\frac{k-1}{k}}\right) + P_{eva}V_{eva} \left(1 - \Pi_{rf}^{\frac{k-1}{k}}\right) \right) + \frac{2kP_{gen}V_{gen}}{k-1} \left(1 - \Pi_{wf}^{\frac{k-1}{k}}\right)} \quad (2.25)$$

Calculation performed at 85°C generation temperature, 35°C condensation and 12°C evaporation. Adiabatic index equals to : $k_{H_2O} = 1,3$; $k_{NH_3} = 1,3$; $k_{R134a} = 1,13$; $k_{RC318} = 1,07$; $k_{R152a} = 1,18$; $k_{R22} = 1,22$.

Energy losses reduce the entrainment ratio by 30-40%. Operating at parameters near critical point significantly reduce ejector efficiency. Thus, based on experimental values of entrainment ratio, it can be considered that other losses reduce the entrainment ratio for an additional 30% (Fig. 2.7).

Naturally, significant impact losses increased attention to an analysis of energy losses[3]. Taking into account only shock losses, fluids with high molecular mass, low critical velocity, the low adiabatic index shows better results, but the final conclusion can be made only after analysis of all factors that affect entrainment ratio.

It is reasonable to use difference fluid for working and secondary flows. If in the expansion-compressor system it does not provide any difficulties, then in ERS, since the fluids interacted directly while mixing. It requires fluid separation at variable concentrations, change of operating parameters, etc.

As a result, binary fluid operation in ERS causes a chain of causes and effects that were described in [3-5] and requires additional study.

Eq 2.25 represents effects on Π function at various entrainment ratios that are defined by backpressure and can reach any values in a reasonable range defined by operating parameters.

Received curves for various fluids at designed operating conditions shows the

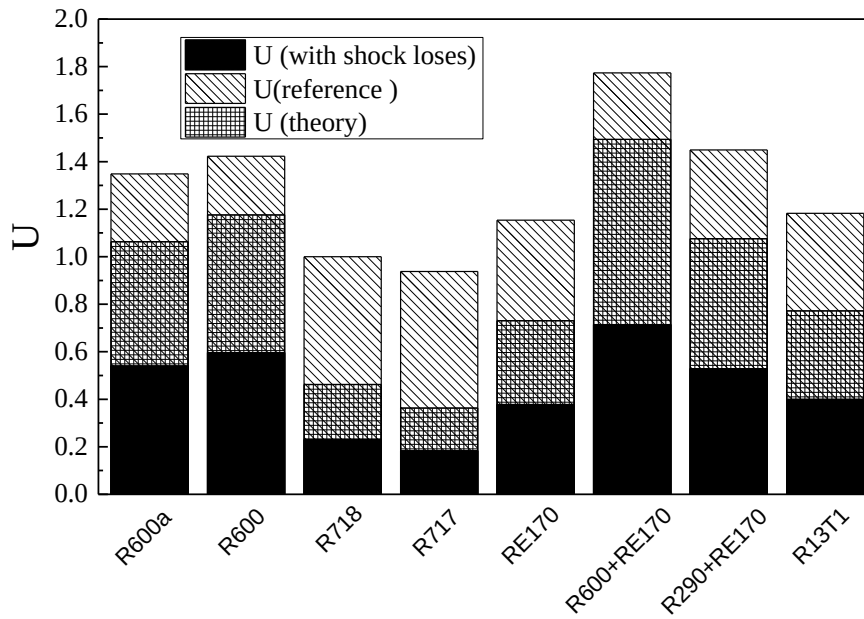


Figure 2.7 Comparison of theoretical entrainment ratio of expansion-compression system and ejector.

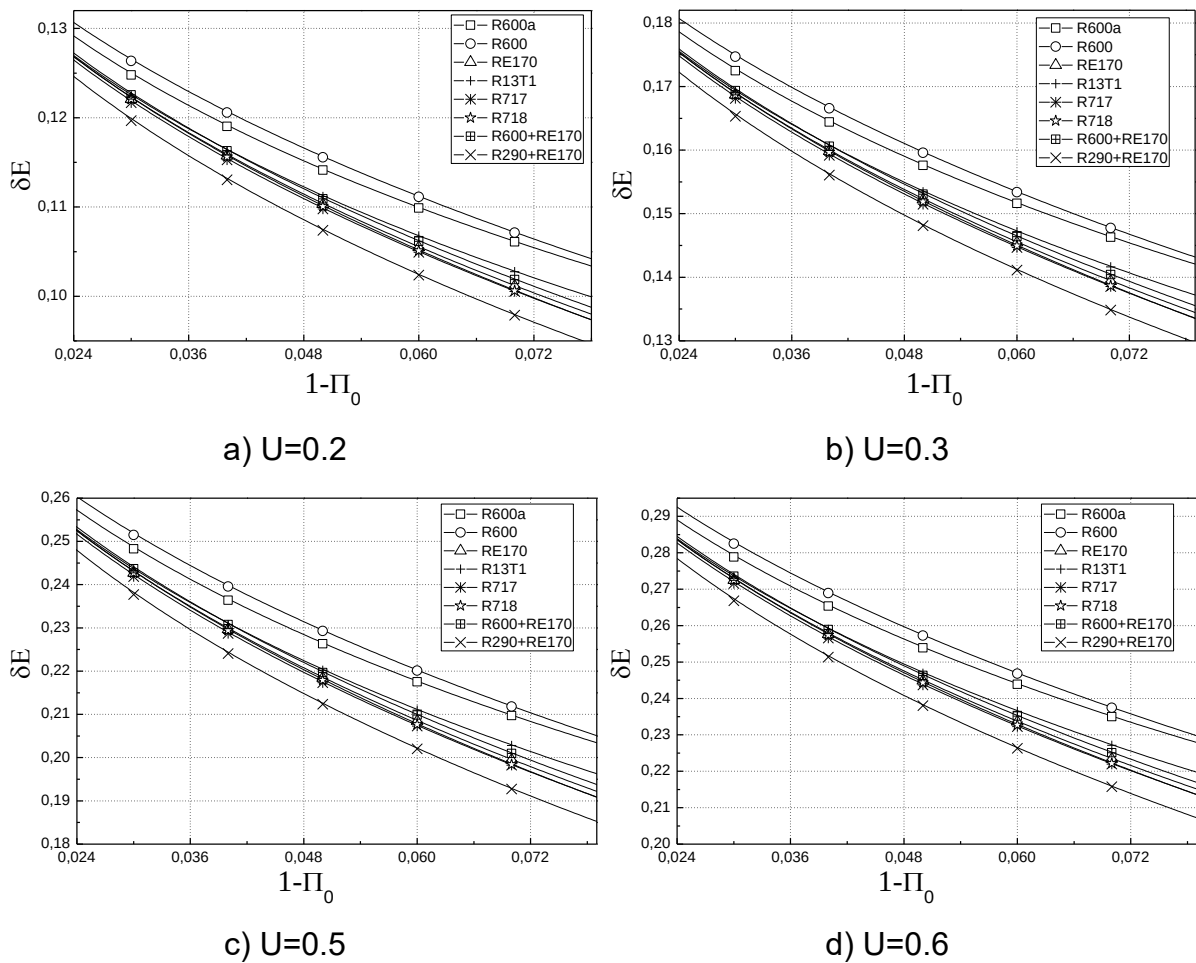


Figure 2.8 Dependence of shock losses in suction chamber at various Entrainment Ratio.

dependence of shock losses from evaporation pressure and entrainment ratio.

Fig. 2.8 represents the dependence of energy losses at various entrainment ratios. For example, steam at pressure values lower than evaporation pressure up to 0.8, specific energy loss decreases more than 50%. At the same time, higher shock losses reduction is observed at lower entrainment ratio that can be considered as increased condensation pressure.

At high backpressures, an expansion of working flow prevents from high velocities, and as a result, kinetic energy is required to pass the backpressure. In this case, the proper design of the suction chamber is required. In read cases, overexpansion of working flow leads to adverse effect. Since the optimal pressure value in suction chamber depends on other parameters, then absolute values received by calculation should not be considered as data for ejector flow part design but can assume a point of methods of ejector efficiency improvement.

Thus, reasons for the low efficiency of ERS comparing to an ideal system can be assumed:

1. Theoretical entrainment ratio of real fluids is two times lower than theoretical entrainment ratio of expansion-compression system and depends on the approach to a critical point, as well as an adiabatic index.
2. Shock losses reduce the entrainment ratio by 30-40% and have an opposite effect on ejectors performance. That means that a decision to reduce losses without taking into account other factors may also lead to the reduction of the entrainment ratio.
3. Flow overexpansion requires a proper design of flow part since velocity may reach close to critical, without overexpansion leads to a significant reduction of ejectors efficiency.

Since ERS realizes direct and reverse cycles at the same time and efficiency of direct cycle utilizing low-grade heat is lower by 10-15 times that reverse cycle efficiency, it is should not be expected to achieve high compression ratio in the ejector. The acceptable energy efficiency of single fluid ERS in air conditioning mode may be achieved at compression ratio 1.8-2.3. At compression ratios, 3-3.5 entrainment ratio became commercially unreasonable. The compression ratio depends on operating parameters, that should be carefully selected for ERS in order to achieve high efficiency. Modern air conditioning systems designed to provide air temperature up to 16°. That requires evaporation temperature of 5-7°C. Usually, comfortable temperature in the room lies in a range of 22-24°C, that allows increasing evaporation

temperature up to 12-15°C. As a result, the efficiency of ERS increases significantly by 15-55%. Condensation temperature affects the COP of ERS significantly. This temperature depends on ambient parameters and available cooling media.

2.2 Binary Fluid Ejector Refrigeration System.

The traditional ERSs are characterized by the simplicity of its design and use, but are behind in efficiency compared to other types of thermally-driven refrigeration systems because of additional losses occurred during the interaction of active and passive flows of the working fluids with essentially diverse velocities. Significant losses appear in the steam-water ERSs, operating at vacuum in the condenser and the evaporator. Nevertheless, steam-water ERSs found its individual application in the facilities where energy effectiveness and operational disadvantages are negligible comparing to safety requirements [1,6]. The researches of the ERSs utilizing low-boiling point refrigerants were started in 1950. These refrigerants can increase the COP of the ERS in 1.5-2 times versus steam-water ERS and simplify the system's operation. However, the commercialization of ERS became possible only after the individual drawbacks were overcome: increase of the ERS energy efficiency, cheap and reliable way to feed the vapor generator with the liquid working fluid, elimination of lubricants due to the absence of moving parts.

The presented research thread suggests that any further gain in the ERS COP is feasible as a result of a proper media selection, including binary and multicomponent fluids [3,7,8].

Figure 2.9 and Figure 2.10 shows a schematic diagram and process diagram of BERS. Comparing to Single fluid ERS, Binary Fluid ERS operates using two fluids. Also, the system contains a fractionating condenser. The working fluid that flows through Vapor generator, Ejector nozzle where it mixes with secondary fluid. From mixed ejector fluid flows through the fractionating condenser, where the working fluid is condensed. Secondary Fluid remains in a vapor state while the Primary Fluid Condensate from the fractionating condenser is pumped to vapor generator. Then, Secondary fluid from fractionating condenser goes to the condenser. Liquid from the condenser is throttled to the evaporator, and vapor from evaporator flows to the ejector suction chamber.

The condensation processes in the BERS run at constant pressure and variable temperatures. The final condensation temperatures of the working fluid and refrigerant

fluid can vary significantly, as shown in Figure 2.11.

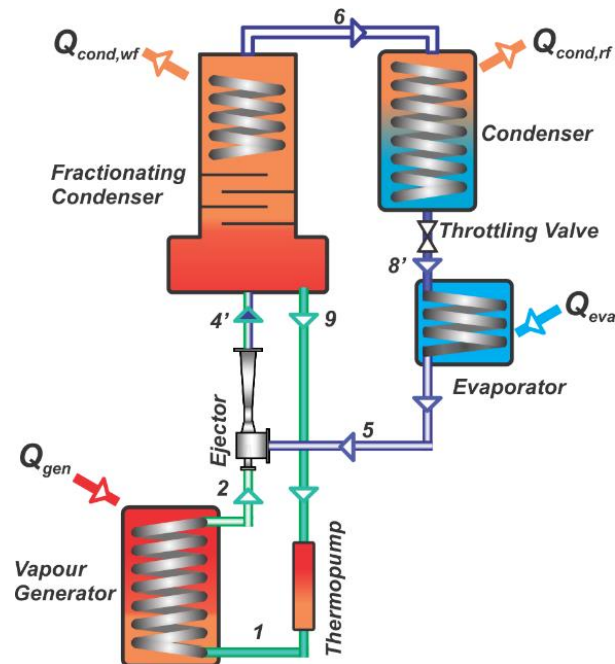


Figure 2.9 Schematic Diagram of BERS

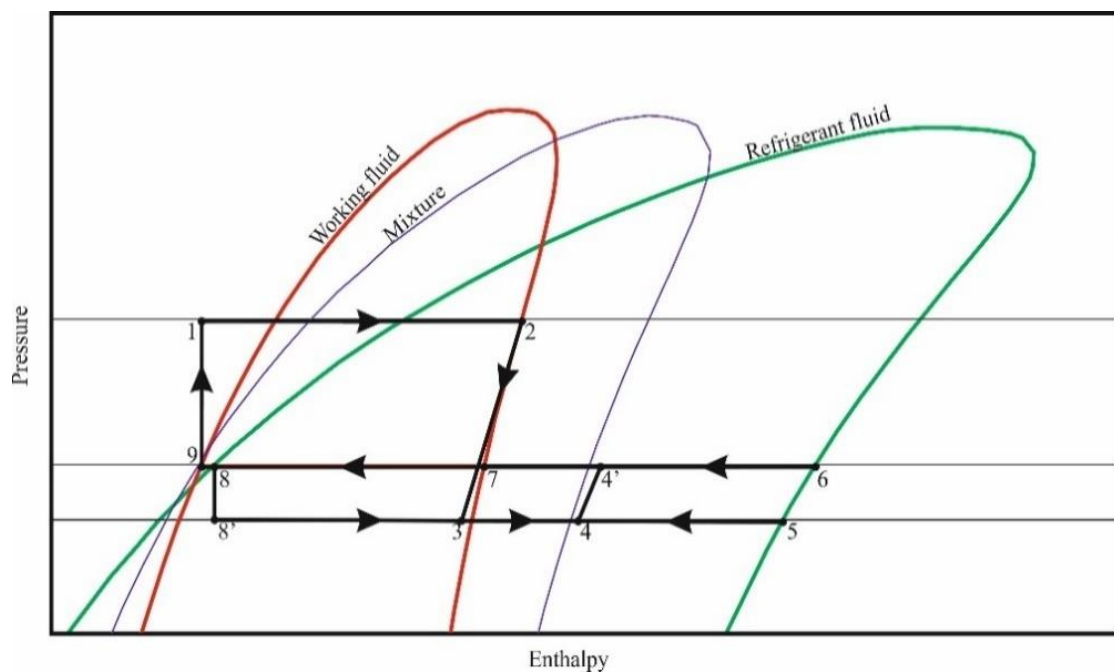


Figure 2.10 Operating diagram of BERS. 1-2 – working fluid heating and evaporation in the vapour generator, 2-3 – working vapour expansion in the nozzle, 3-4 and 5-4 – working flow and refrigerant flow mixing in the confuser, 4-4' – mixture compression in the cylindrical mixing chamber, 7-8 working fluid condensation in the fractionating condenser, 6-9 refrigerant fluid condensation, 8-8' – refrigerant fluid throttling, 8-5 – refrigerant fluid evaporation, 9-1 – working fluid pumping into the vapour generator.

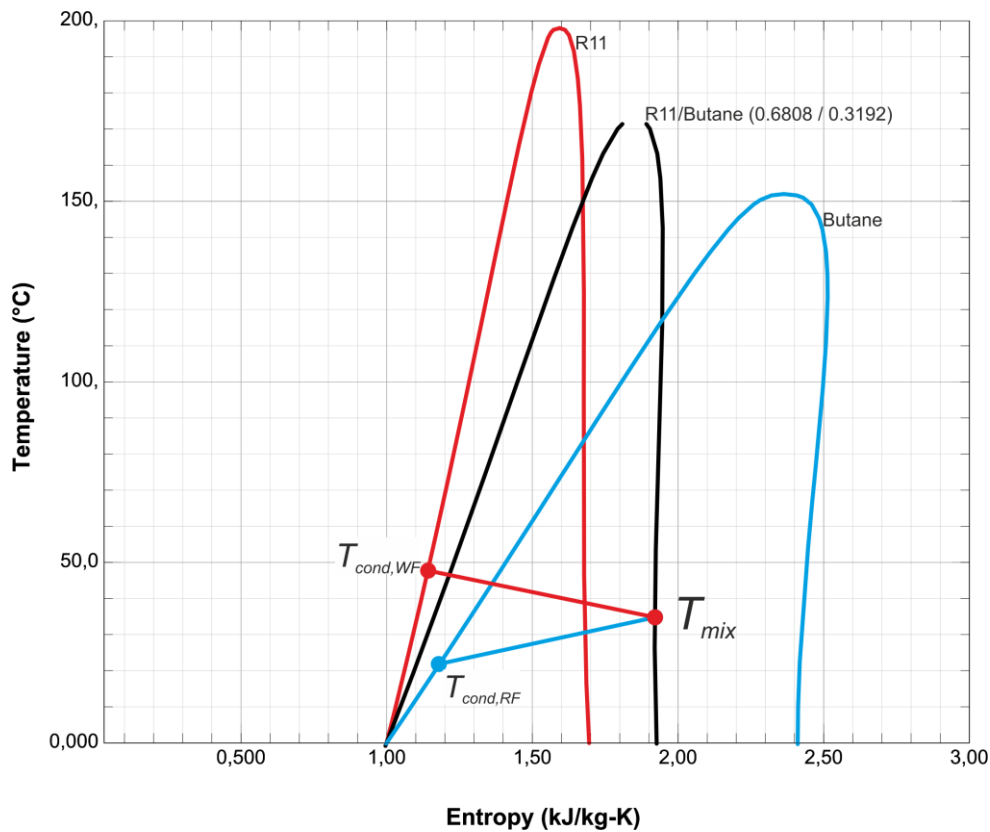


Figure 2.11 Diagram of working and refrigeration fluids condensation processes

2.2.1 Criteria of Fluid Selection

The current thermally-driven cooling systems run the power and the refrigeration cycles simultaneously. The effectiveness of such cycles is determined, mainly, by the fluids' properties and different criteria are applied for selection of each of it. For example, sorption type cold-generators use binary fluids, which have a wide range of normal boiling temperatures or at various aggregative states. Expander-compression heat-utilizing refrigeration systems also work more effectively with several working fluids instead of a single one. The same applies to the ERS but with specific requirements to the properties of fluids due to the direct interaction of the high-speed flows.

In order to decrease the shock losses in the ejector and save the kinetic energy of the mixed flow, the speed of sound of the working fluid should be lower than of the refrigerant fluid so that the flows will collide at a lower velocity difference. Although this requirement defines the desired molecular weight ratio, it identifies the latent heat of evaporation of the employed media. Thus, the working fluid should exhibit a high

molecular weight, low speed of sound, and low latent heat of evaporation. The refrigerant fluid should have low molecular weight and high latent heat of evaporation. It should also be considered the other material properties of the binary fluid, such as intermiscibility of the fluid components, azeotropic, and the thermodynamic properties at the selected cycle parameters. The COP of ERS is represented as the relation of the generated cold to the supplied heat (Eq. 2.26):

$$COP = \frac{Q_{eva}^{rf}}{Q_{gen}^{wf}} = U \frac{q_{eva}^{rf}}{q_{gen}^{wf}} \quad (2.26)$$

The entrainment ratio (U) also can be described as the relation of the work of adiabatic expansion to the work of adiabatic compression (Eq.2.27-2.30).

$$G_{wf} l_{exp} = l_{comp} (G_{wf} + G_{rf}) \quad (2.27)$$

$$U_{theor} = \frac{l_{exp}}{l_{comp}} - 1 \quad (2.28)$$

$$l_{exp} = \frac{Z_{wf} RT_{gen,wf}}{k_{gen,wf} - 1} \left[1 - \left(\frac{P_{gen,wf}}{P_{eva,rf}} \right)^{(k_{gen,wf} - 1)/k_{gen,wf}} \right] \quad (2.29)$$

$$l_{comp} = \frac{Z_{rf} RT_{eva,rf}}{k_{eva,rf} - 1} \left[\left(\frac{P_{cond,mix}}{P_{eva,rf}} \right)^{(k_{eva,rf} - 1)/k_{eva,rf}} - 1 \right] \quad (2.30)$$

The COP is in direct proportion to entrainment ratio U and the latent heat of evaporation ratio.

$$\frac{r_{rf}}{r_{wf}} \quad (2.31)$$

Generally, high latent heat of evaporation ratio corresponds to a low work of expansion and compression ratio, i.e., U will be lower.

Fluid components selection is grounded on the requirements of the Kyoto and Montreal Protocols, Copenhagen and Doha Conferences in respect of valid GWP and ODP values. Selection of working and refrigerant fluids was performed from the variety of media suitable for ERS based on its thermodynamic properties, safety, and toxicity criteria (APPENDIX A).

Refrigerant R11, banned by the Montreal protocol, was considered only for comparison purposes. However, it was proved that binary fluid ERS (BERS) using R11

as a working fluid brings leverage for the system performance at specific operation modes.

It is believed that ODP and GWP parameters for binary fluid using R11 as a working fluid and R600 as refrigerant fluid (which is almost environmentally neutral) can take additive properties, which can fit into the acceptable limits [9]. At the same time, R11 can serve as an inhibitor of fire and explosion hazards, attributable to R600. Mixture R1233zd(E)/R600 is preferable from the terms of safety and ODP, GWP requirements. Though, it was proved that about the only type of the chemically active gas compounds used for fire extinguishing and explosion prevention are halons - halogenated hydrocarbons [10,11], further studies are required to identify the ideal binary fluid for BERS or engineering a novel fluid to fit the optimal specifications.

The most attractive refrigerants to be used in the ERS are likely to be organic. However, the calculations showed its low efficiency when applied in the ERS. For example, COP of the ERS using water and ammonia has low entrainment ratio and COP (in a range of 0.09-0.27 at the operation mode $t_{\text{gen}}=85^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}\text{C}$, $t_{\text{eva}}=3^{\circ}\text{C}$). Meanwhile, CO₂ based systems can perform only at a low-temperature stage of the cascade ERS. The binary fluids using water as a refrigerant have a high ratio of the specific cooling capacity to the specific heat supply. However, the entrainment ratio is still low, considering the properties of heteroazeotrope binary fluids that water forms with refrigerants or ethers[9]. Propane and butane are the promising refrigerants for zeotrope binary fluids in with high molecular weight components. Extensive analytical and CFD modeling showed that zeotrope mixtures have more perspective to be used in the BERS.

Fig. 2.12 represents an optimal type of binary fluid application based on operating parameters.

Due to the lack of binary mixture data, T-X diagram of the mixture was obtained from REFPROP (Fig. 2.13-2.15).

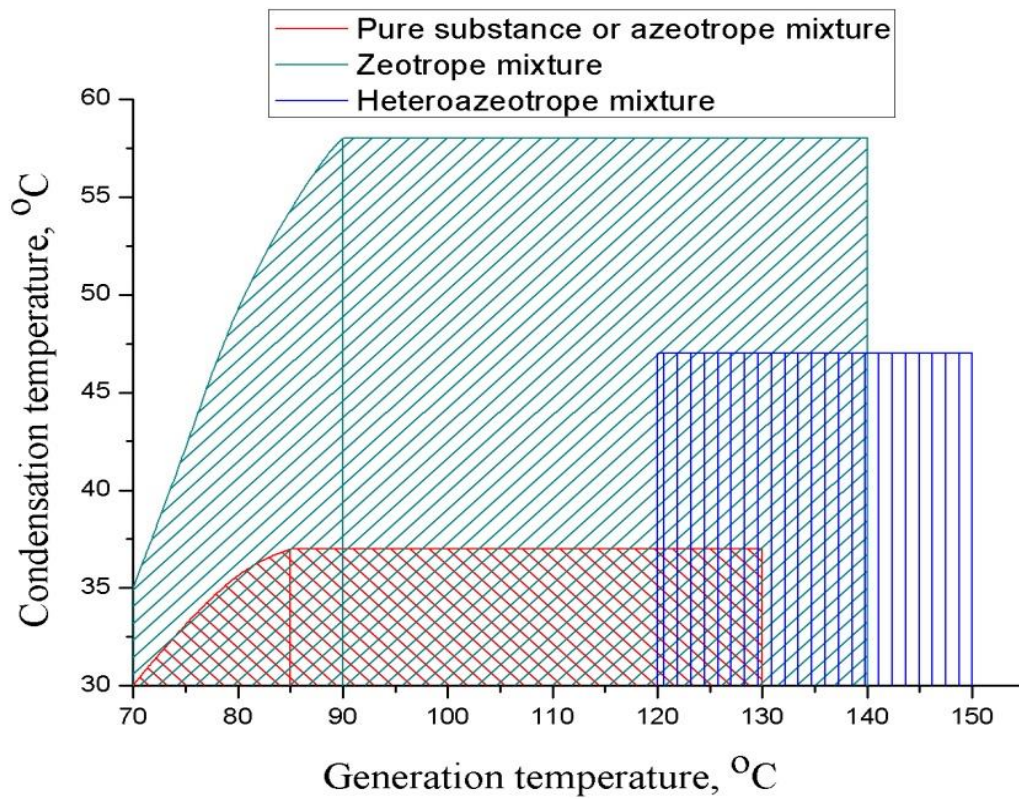


Figure 2.12 The diagram for a choice of type of a refrigerant at different operating parameters of Ejector Cooling cycle

14: Temperature vs. Composition plot: R1234ze(E)/R161 (50/50)

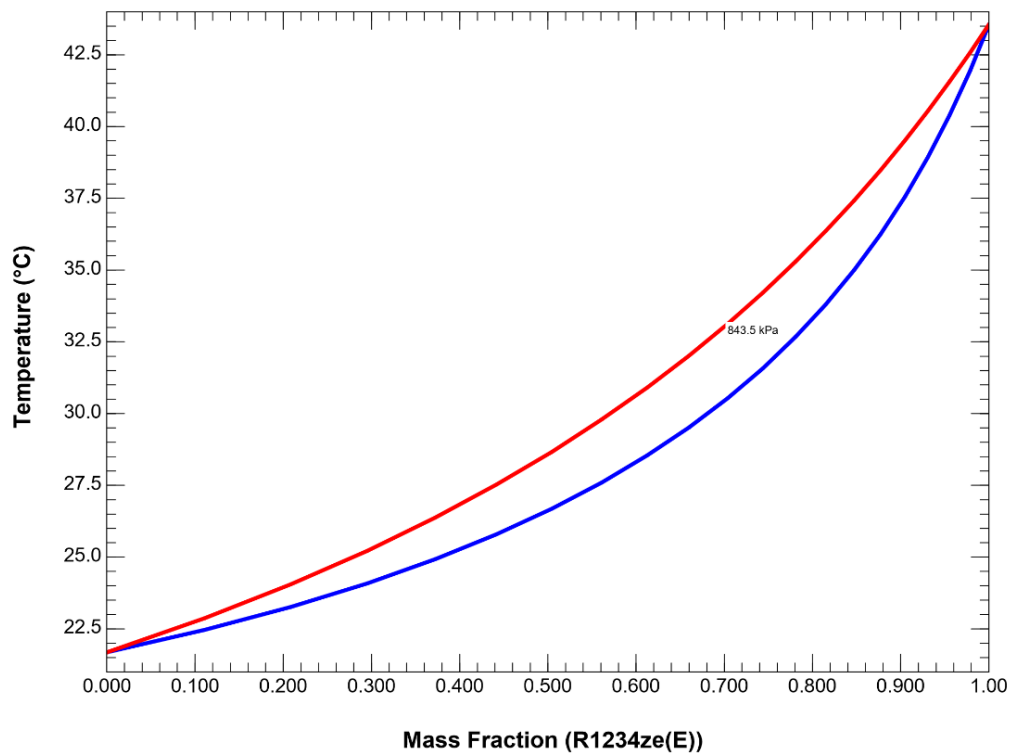


Figure 2. 13 T-X diagram of R1234ze(e)/R161

6: Temperature vs. Composition plot: R1234ze(E)/dimethylether (50/50)

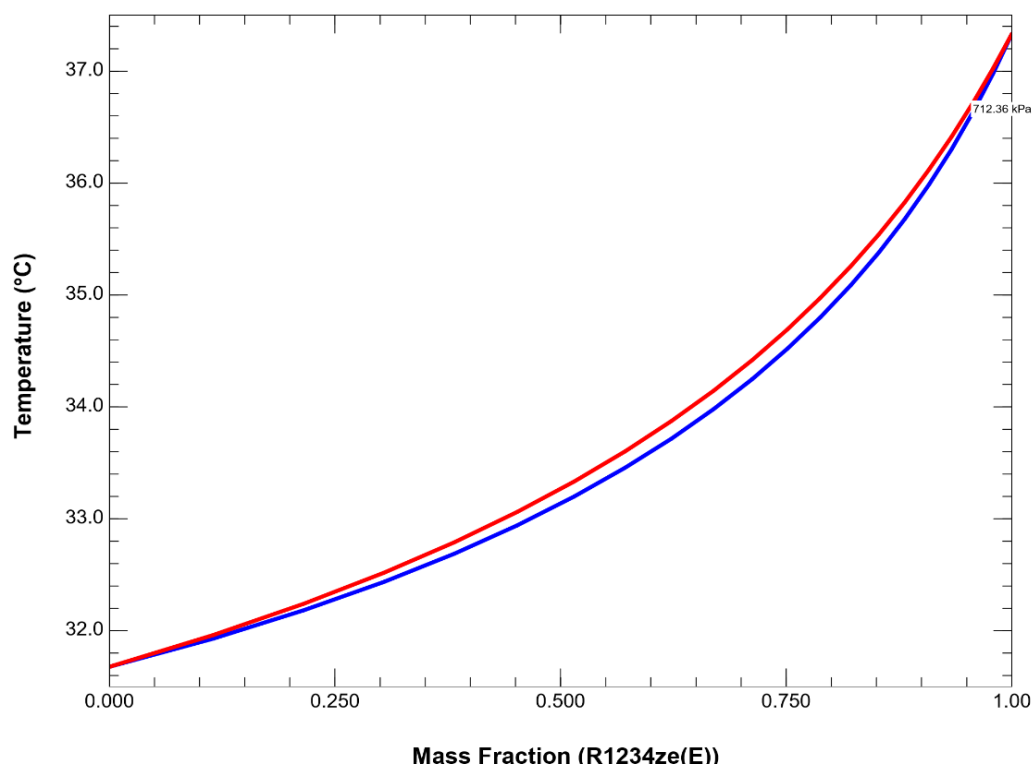


Figure 2. 14 T-X Diagram of R1234zde/DME

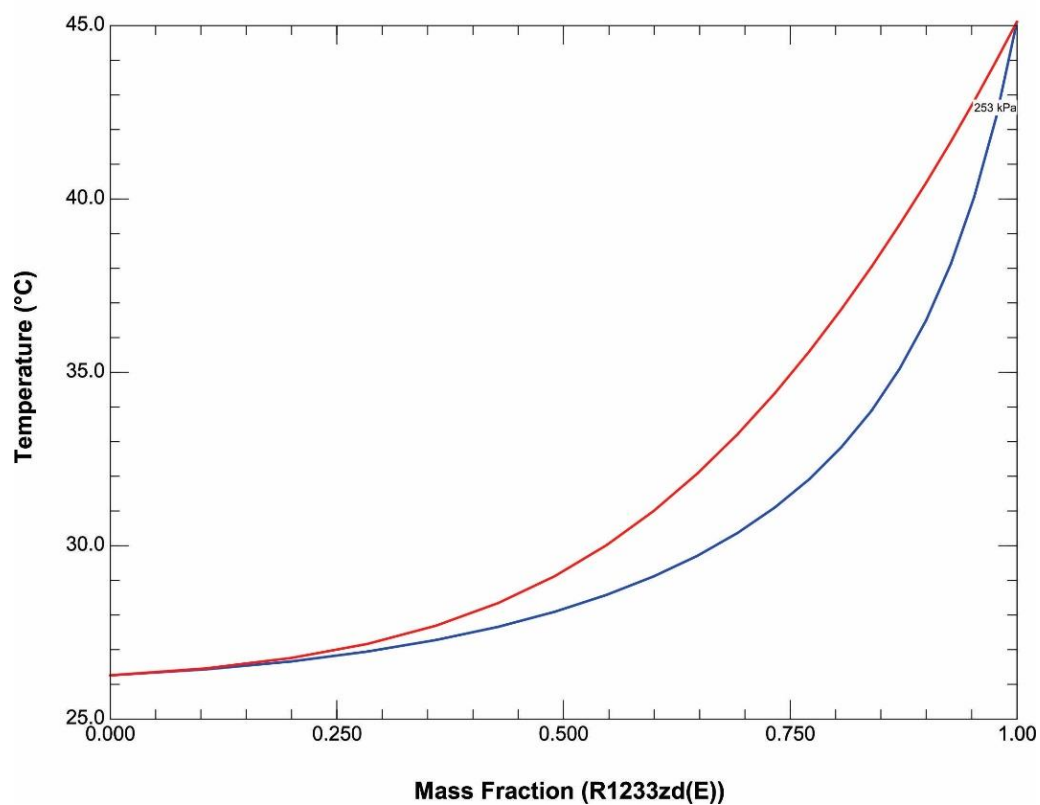


Figure 2.15 T-X diagram of R1233zd(E)/Butane

2.2.2 Influence of fluids thermodynamic properties on ejector efficiency .

For the analysis of the criteria of fluids selection for BERS, the various compositions of the available refrigerants were considered. Appendix B shows the entrainment ratio and values of the corresponding factors affecting the variation of the entrainment ratio in the operational mode: $t_{gen}=85^{\circ}\text{C}$, $t_{cond}=35^{\circ}\text{C}$, $t_{eva}=3^{\circ}\text{C}$. The values of the entrainment ratio were calculated using the in-house code described in [12].

Molecular weight, speed of sound, and latent heat of evaporation are unlikely to be the only criteria for the selection of ideal fluid components for BERS with the highest performance. Sometimes, low entrainment ratio is not compensated by the ratio of the specific cooling capacity to the specific heat supply.

The entrainment ratio, however, tends to be a subject of the molecular weights of the binary mixture components (Fig. 2.16a). Normally, the entrainment gain at molecular weights ratio loss reveals that expansion and compression specific work ratio is correctly balanced unlike the increased shock losses, which are not supportive for the entrainment. It proves that an optimum value of the molecular weights ratio has to be identified in order to reach the maximal entrainment ratio in the BERS. In our case, this ratio is in the range of 1.8 – 2.2. Since the curve is plotted for the single set of the operating parameters, and possibly can be different if calculated for the other operation modes.

The relationship $(P_{gen,wf}/\rho_{gen,wf})/(P_{eva,rf}/\rho_{eva,rf})$ can also influence the entrainment ratio. When it decreases, the entrainment ratio usually grows (Fig. 2.16b).

Another studied criterion is the compressibility factors ratio. The entrainment ratio increases with the increase of this criterion provided all other factors remain unchanged (Fig. 2.17a).

The COP of BERS is also determined by the latent heat of evaporation ratio of the fluid components. The entrainment tends to decrease when the latent heat of evaporation ratio growth that makes the COP negligibly affected by this ratio. It proves the canceling effect of two components on the COP (Fig. 2.17b).

Additionally, the entrainment ratio relation to the fluid components critical temperatures balance $(T_{crit,wf}/T_{crit,rf})$ and normal boiling temperatures balance $(T_{nb,wf}/T_{nb,rf})$ was analysed.

The first graph (Fig. 2.18a) shows the increase of the entrainment ratio along with an increase of the critical temperatures balance.

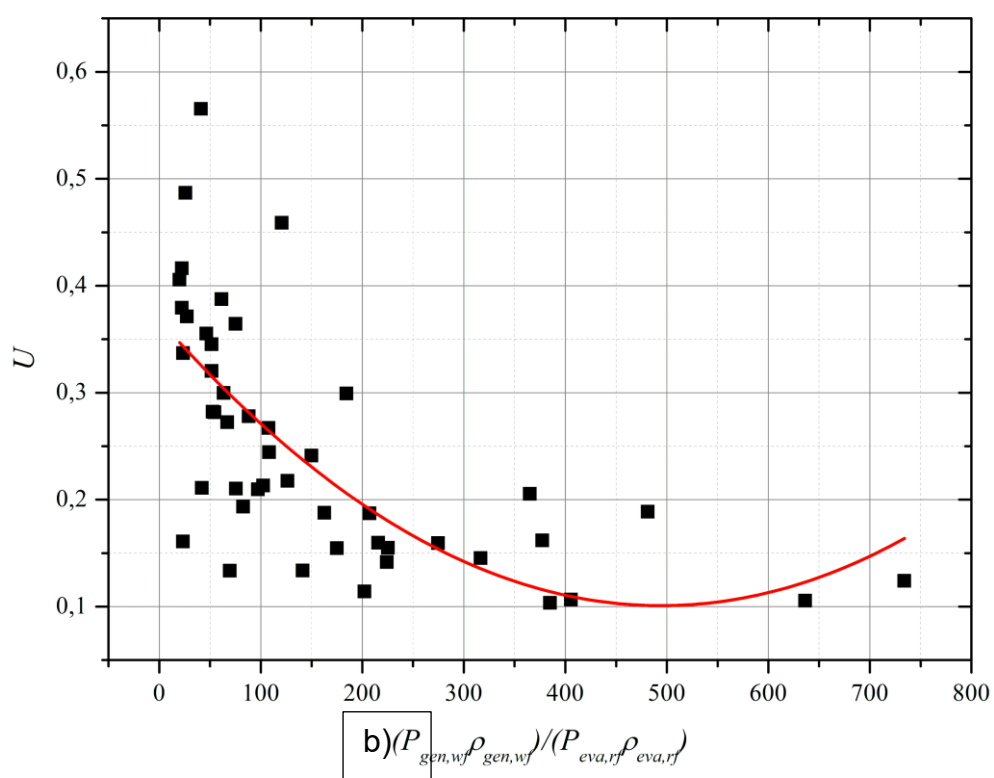
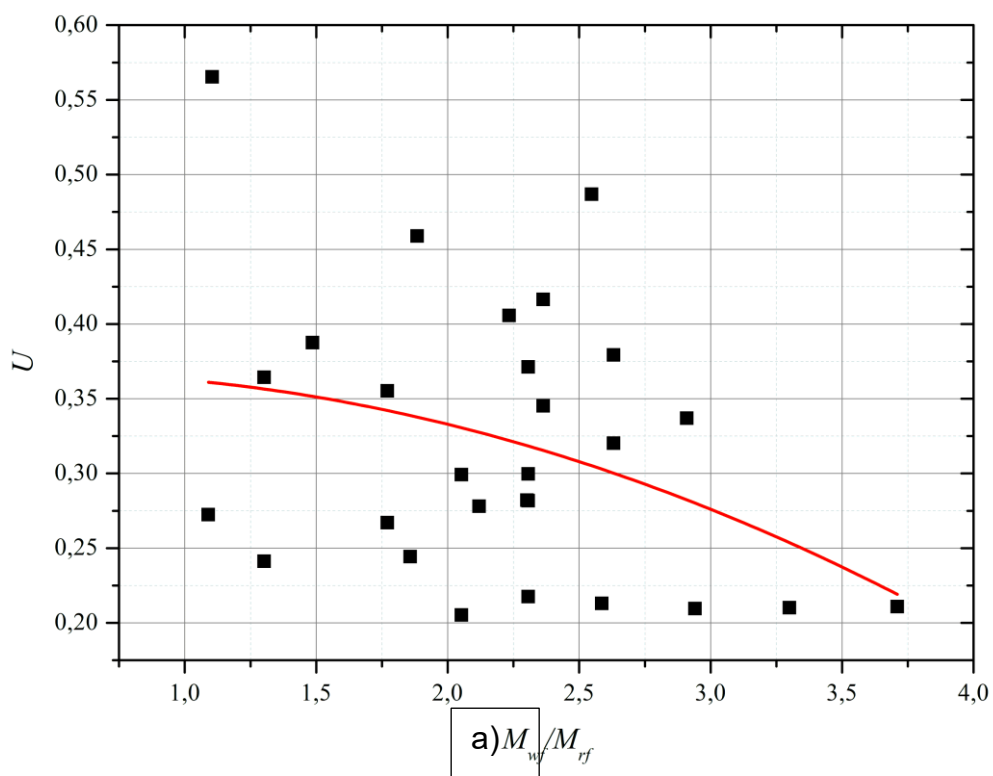


Figure 2.16 Dependence of a – the entrainment ratio vs. molecular weights ratio graph; b – the entrainment ratio vs. $P_{gen,wf} \rho_{gen,wf} / P_{eva,rf} \rho_{eva,rf}$ graph.
(APPENDIX B)

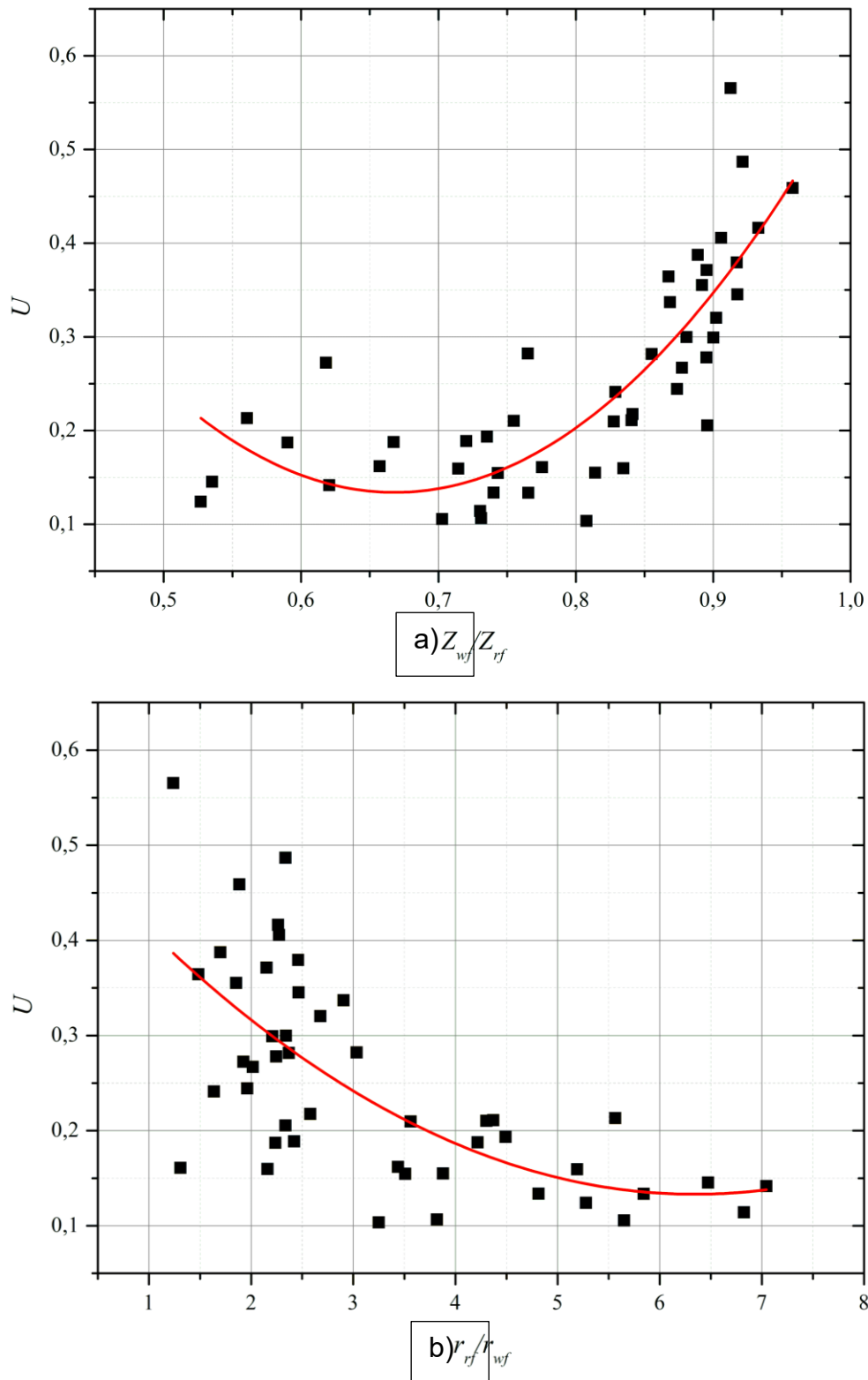


**Figure 2.17 a – the entrainment ratio vs. compressibility factors ratio graph;
b – the entrainment ratio vs. the latent heat of evaporation ratio graph.
(APPENDIX B)**

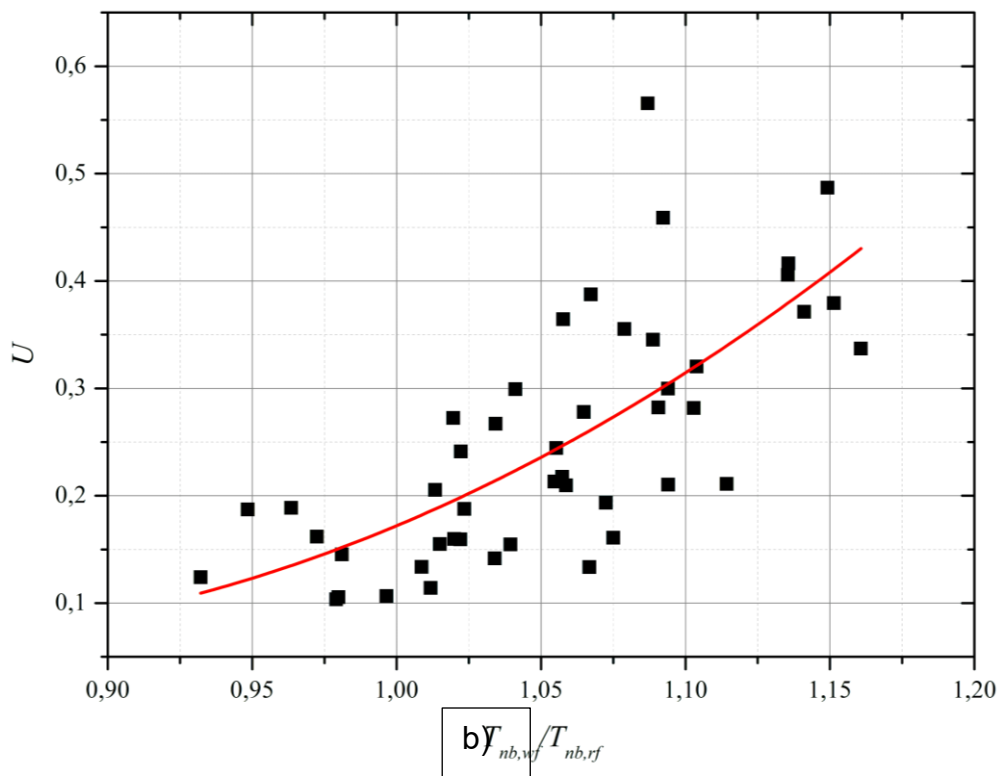
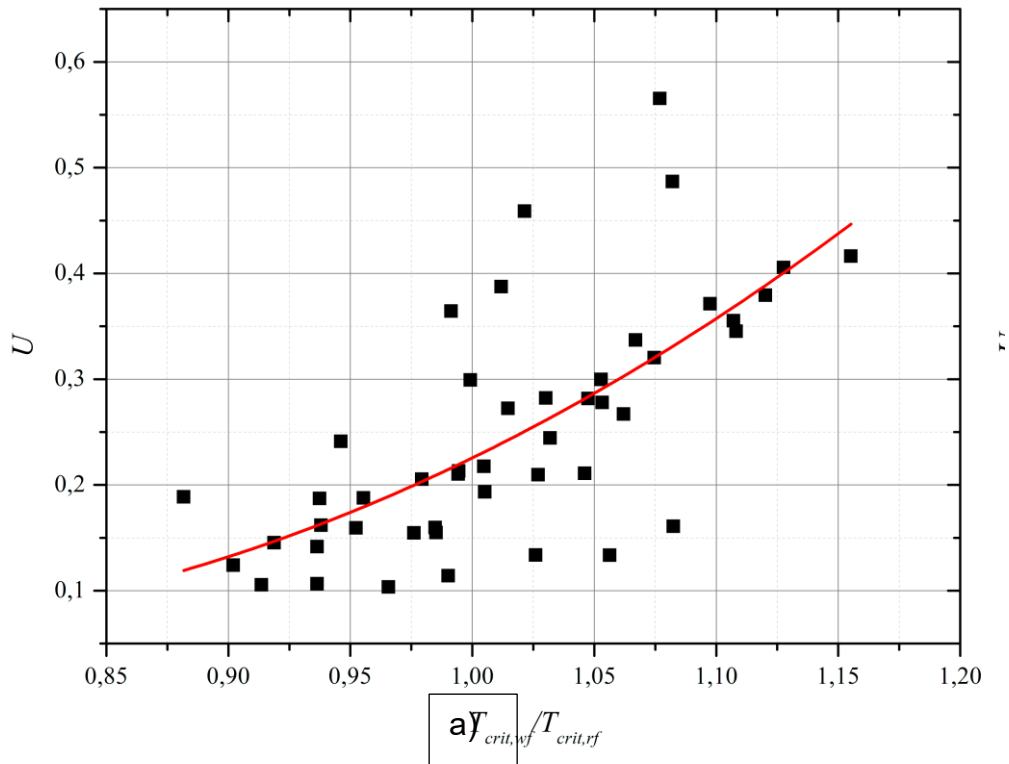


**Figure 2. 18 a – the entrainment ratio vs. critical temperatures balance graph;
b – the entrainment ratio vs. normal boiling temperatures balance graph.
(APPENDIX B)**

The second graph shows the same relation, but comparing to normal boiling temperatures ratio (Fig. 2.18b).

The second graph shows the same relation, but comparing to normal boiling temperatures ratio (Fig. 2.18b).

The nature of the first curve can be explained by the fact that the generation process in the power cycle occurs at a certain distance from the critical point, which increases its efficiency. On Fig. 2.16b, the effect of shock losses reduction on the entrainment ratio prevails the effect of $P_{gen,wf} \rho_{eva,rf} / P_{eva,rf} \rho_{gen,wf}$.

All of the curves indicate only traceable patterns that allow assessing or predicting the behavior of the binary fluid components in the BERS.

2.3 BERS Efficiency Evaluation

The mixtures selection was based on criteria of the significant difference of molecular weights of working and refrigerant fluids. These criteria also provided the sufficient ratio of the critical speeds of sound and the latent heats of evaporation.

The analysis of Eq. 2.32 and 2.33 shows that fluids with high molecular weight have a low critical speed of sound and low latent heat of evaporation and vice-versa:

$$a_{crit} = \sqrt{\frac{kRT}{\mu}} \quad (2.32)$$

$$\theta = \frac{\mu_s}{T_s} \quad (2.33)$$

where a^* - critical speed of sound (m/s); θ - Trouton's constant (J/mol K).

Binary fluid chosen by designed criteria leads to significant shock losses reduction (Eq. 2.34, 2.35):

$$W_i = a_* \lambda_i \quad (2.34)$$

$$\Delta E = \frac{1}{2} \frac{G_{wf} G_{rf}}{G_{wf} + G_{rf}} (W_{gen} - W_{eva})^2 \quad (2.35)$$

where W - flow rate (m/s); λ_i - relative velocity, ΔE - shock losses, W.

ERS model and its applications for air conditioning using both single fluids and zeotropic mixtures were studied in [13]. Comparison of the ERS performance using zeotropic and azeotropic mixtures as substitutes of pure refrigerants was made in [14]. The framework of binary fluid ERS applications for solar cooling was provided in [15], where benefits of such system were analyzed, and a significant increase of COP was reported. Comparison of binary and single fluid ERS operating at off-design conditions was made, and it was predicted that binary fluid ERS has better performance

compared to single fluid one.

New ejector refrigeration system using zeotropic mixtures was theoretically studied in [16]. This system's performance was compared with single fluid ERS under the same operating conditions. Binary fluid ERS proved to reach higher COP.

Generally, mixtures selected for BERS had 10-20°C higher normal boiling temperature of the working fluids than of the refrigerant fluids. The ratio of generation pressure to condensation and evaporation pressures was lower while working vapor density exceeded the ejecting vapor density significantly [3]. These factors made BERS entrainment ratio lower than of the single-fluid unit and disfavor mixture application in the ejector refrigeration systems. Optimization of BERS efficiency became possible when certain assumptions and approaches to the fluid selection principles were applied [8].

- compressibility factors ratio $Z = pv/RT$ (at the set operating parameters or normal conditions) should be considered while a selection of fluid components:
- in order to increase the entrainment ratio and COP, control for concentrations of fluid components in the evaporator and in the vapour generator is required.

The candidate working fluid should have high molecular weight, high normal boiling temperature, and low specific heat of evaporation. The candidate refrigerant fluid should have reverse characteristics: low normal boiling temperature, low molecular weight, and high specific heat of evaporation. Compressibility factor of working fluid shall be high and for refrigerant fluid - low. Generally, compressibility factor Z of the working fluids is lower that of the refrigerant fluids that limits BERS efficiency.

The in-house code based on the method described in [12] was used for calculation of the entrainment ratio and COP at the given working cycle parameters.

The set of equations of energy, momentum, and mass conservation is the basis of entrainment ratio calculation (Eq. 2.33):

$$\begin{cases} (G_{wf} + G_{rf})W_3 = G_{wf}W_{g1} + G_{rf}W_{ev1} \\ \varphi_2 (G_{wf}W_{g2} + G_{rf}W_{ev2}) - (G_{wf} + G_{rf})W_3 = (P_3 - P_{g2})f_{g2} + (P_3 - P_{ev2})f_{ev2} \\ G_c = G_g + G_{ev} = G_g(1+U) \end{cases} \quad (2.33)$$

where $\varphi_2=0.975$; f_{g2} , f_{ev2} – areas of working and ejected flows in inlet cross-section of cylindrical mixing chamber; P_{g2} , P_{ev2} , P_3 – static pressure of working and ejected flows in inlet cross-section and mixed flow in the outlet section of the cylindrical mixing chamber.

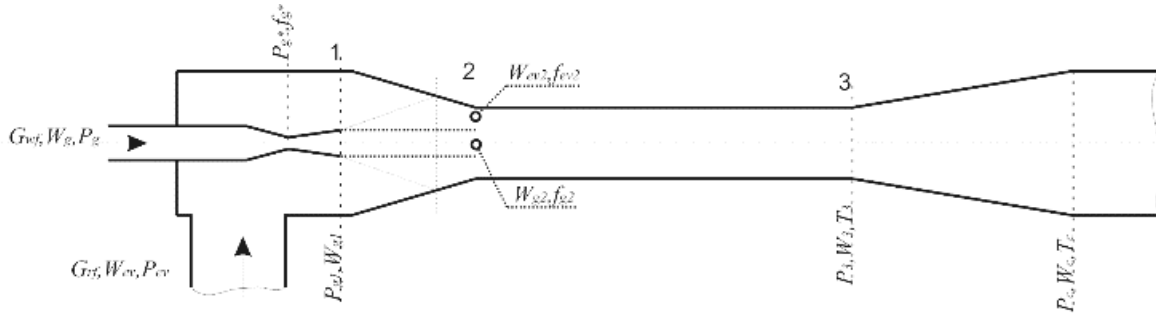


Figure 2.19 Scheme of an ejector. 1 – nozzle outlet cross-section area; 2 – cylindrical mixing chamber inlet cross-section area; 3 - cylindrical mixing chamber outlet-cross-section area.

For candidate binary fluid, the entrainment ratio and COP were calculated. The calculation was performed using an algorithm, as shown in Fig. 2.20.

1. Specify initial concentrations of fluid components in the evaporator, vapor generator, and in the outlet of an ejector. Define operating temperatures;
1. Calculations of the mixture properties at different concentrations and operating parameters (saturation pressure, a specific volume of vapor and adiabatic index) are performed using REFPROP;
2. Entrainment ratio is described as a function of temperatures, pressures, volumes, concentrations, the adiabatic index, the critical speed of sound, and the central dynamic functions Π , γ , λ [1]. Entrainment ratio is determined by equation 2.34 upon reaching the limit regimes:

$$U = \frac{K_1 \left(a_{gen,*}^{wf} / a_{cond,*}^{mix} \right) \lambda_{ph} - K_3 \lambda_{cond,C}}{K_4 \lambda_{cond,C} - K_2 \left(a_{eva,*}^{rf} / a_{cond,*} \right) \lambda_{eva,B}} \quad (2.34)$$

where

$$K_1 = \varphi_1 \varphi_2 \varphi_3 \quad (2.35)$$

$$K_2 = \varphi_2 \varphi_3 \varphi_4 \quad (2.36)$$

$$K_3 = 1 + \frac{\Pi_{cond,C} - \frac{P_{eva}^{rf}}{P_{cond}^{mix}} \left\{ \beta - \frac{(\beta-1)}{2} \Pi_{eva,B} \left[1 + \left(\frac{P_{cond}^{mix}}{P_{eva}^{rf}} \right)^{1-\alpha} \left(\frac{\Pi_{cond,C}}{\Pi_{eva,B}} \right)^{1-\alpha} \right] \right\}}{\left(\frac{1}{\varphi_3} \frac{a_{cond,*}^{mix}}{a_{gen,*}^{wf}} \frac{P_{gen}^{wf}}{P_{cond}^{mix}} \right) k_{gen} \Pi_{gen,*} \lambda_{cond,C} \gamma_{gen,A} \beta} \quad (2.37)$$

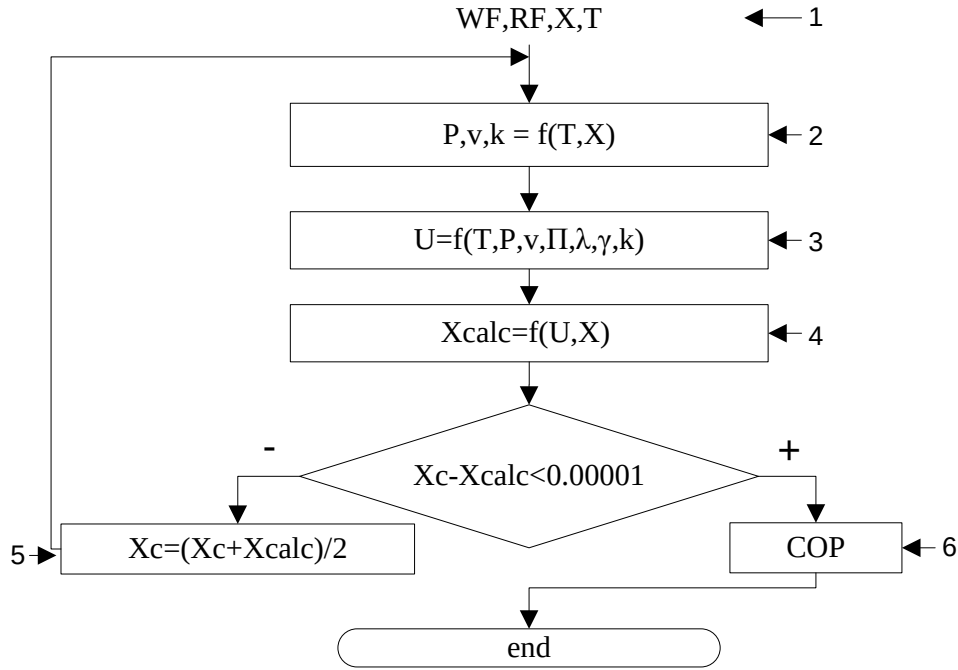


Figure 2.20 Algorithm of BERS calculation

WF - working fluid; RF - refrigerant fluid; X - concentrations of refrigerant fluid in evaporator, generator, condenser; T - temperatures in generator, evaporator, condenser; P - pressures, v - volumes, k - adiabatic index.

$$K_4 = 1 + \frac{\Pi_{cond,C} - \Pi_{cond,B} \left\{ \beta - \frac{(\beta-1)}{2} \left[1 + \left(\frac{P_{cond}^{mix}}{P_{eva}^{rf}} \right)^{1-\alpha} \left(\frac{\Pi_{cond,C}}{\Pi_{eva,B}} \right)^{1-\alpha} \right] \right\}}{\left(\frac{1}{\varphi_3} \frac{a_{cond,*}^{mix}}{a_{eva,*}^{rf}} \frac{P_{eva}^{rf}}{P_{cond}^{mix}} \right) k_{eva} \Pi_{eva,*} \lambda_{cond,C} \gamma_{eva,B} \beta} \quad (2.38)$$

$$\lambda = \sqrt{(k+1)/(k-1)} \sqrt{1 - \Pi^{(k-1)/k}} \quad (2.39)$$

$$\Pi = P / P_{eva} \quad (2.40)$$

$$\alpha = 0.5, \beta = 2 \quad (2.45)$$

$$\gamma = \sqrt{(k+1)/(k-1)} \left(\frac{\Pi}{\Pi_*} \right)^{1/k} \sqrt{1 - \Pi^{(k-1)/k}} \quad (2.46)$$

where Π is the relative pressure; K_1 the integrated velocity coefficient of the working flow; K_2 the integrated velocity coefficient of the ejected flow; φ_1 , φ_2 , φ_3 and φ_4 are the experimental velocity coefficients; K_3 and K_4 are the integrated velocity coefficients; a and b are the empirical coefficients and g the relative mass velocity.

4. The optimal concentration at the outlet of the ejector is calculated using the resulting entrainment ratio and concentrations. If the estimated concentration does not coincide with the specified value, then go to Step 5. By adopting the concentration of the mixture at the outlet from the ejector as a weighted average between the defined and calculated concentrations, return to step 2, and the calculation is repeated. If the calculated concentration coincides with the set one, then go to step 6 and calculate the system's COP.

2.4 Description of the 3D CFD model – binary fluid ejector efficiency calculation and optimal geometry evaluation based on the mathematical model

2.4.1 CFD model description.

Modern algorithms of ejector efficiency evaluation are based on 1-D models proposed by Keenan [17], Eames [18], Huang [19]. Those models were well described, verified, and improved by Chen [20], Zhu [21]. However, even small dimensions divergence of ejector flows part decreases significantly entrainment ratio comparing to maximum possible, that leads to minimizing the efficiency of Ejector System.

The tasks of this article are:

1. Maximize the use of computer technologies to obtain reliable geometry of the flow part;
2. Create a universal model of ejector efficiency evaluation, that allows providing calculations for various fluids and their mixtures;
3. Determine velocity coefficients of the ejector flow part for different refrigerants, taking into account the modern level of manufacturing;
4. Provide experimental verification of achieved results and their theoretical justification.

Considering the described tasks, the one-dimensional model does not fit the requirements to obtain a maximum entrainment ratio.

It can be observed that the entrainment ratio increased by 15-25%, comparing values of entrainment ratios obtain in 1970th and new results obtained for the same fluids and regimes. It caused by more correct calculations and more precise manufacturing of flow parts.

The possibility to increase the efficiency of the ejector is evidenced by the fact that calculated values of entrainment ratio are sometimes below the experimental values obtained by CFD simulation. That indicates that the calculation models contain deliberately understate parameters of the ejector flow part efficiencies. In order to resolve these contradictions, it was attempted to clarify the values of velocity coefficients of the nozzle, suction chamber, mixing chamber and diffuser for several standard refrigerants. The results of CFD modeling are taken as a basis since it takes into account shock waves and changes of parameters of working fluid in any point of ejector flow part.

Order of CFD modeling procedures:

1. A preliminary calculation of the geometrical parameters of the flow part of the ejector is performed using a universal program.
2. The obtained geometric characteristics serve as the basis for building an ejector in the editor.
3. On the basis of the equations of the mathematical model of the ejector, the number of dimensions of the flow part of the ejector is refined, which are not uniquely determined.
4. The construction of the computational grid based on the finite element method. In order to maximize the concentration of grid cells on the model, only a quarter of the ejector consisting of 1684499 elements was used in the calculation process, which does not affect the quality of the results obtained.
5. The flow-through part of the ejector with its shape and dimensions obtained by modeling is transferred to an electronic drawing that is used directly in the manufacture of the ejector. As a result, the resulting real ejector has the highest energy performance possible.

2.4.2 Governing equations.

1. The continuity equation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho U) = 0 \quad (2.47)$$

where ρ – density (ML^{-3}), t – time (T), U – velocity (LT^{-1}).

2. The equation of conservation of momentum

$$\frac{\partial(\rho U)}{\partial t} + \nabla \cdot (\rho U \otimes U) = -\nabla p + \nabla \cdot \tau + S_M \quad (2.48)$$

$$\tau = \mu \left(\nabla U + (\nabla U)^T - \frac{2}{3} \delta \nabla \cdot U \right) \quad (2.49)$$

where τ stress tensor ($\text{ML}^{-1}\text{T}^{-2}$), S_M – pulse source ($\text{ML}^{-2}\text{T}^{-2}$), δ Kronecker delta unit matrix, T – static temperature (Θ)

3. The equation of total energy

$$\frac{\partial(\rho h_{\text{полн}})}{\partial t} - \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho U h_{\text{полн}}) = \nabla \cdot (\lambda \nabla T) + \nabla \cdot (U \cdot \tau) + U \cdot S_M + S_E \quad (2.50)$$

$$h_{\text{полн}} = h + \frac{1}{2} U^2 \quad (2.51)$$

where h_{tot} total enthalpy depending on static enthalpy $h(T, p)$, S_E – energy source ($\text{ML}^{-1}\text{T}^{-3}$), λ – thermal conductivity ($\text{MLT}^{-3}\Theta^{-1}$), $\nabla \cdot (U \cdot \tau)$ – friction work, describes work related to viscous stress, $U \cdot S_M$ – work under the influence of external sources (neglected in this case).

2.4.3 Turbulence models.

Two-parameter turbulence models are widely used because they offer a good compromise between numerical achievements and computational accuracy. Although at the same time, the two-parameter model is much more complicated than the "non-parametric" (zero equation) model, since it considers the speed and linear lengths in various transport equations.

The k- ϵ and k- ω models use the parameter gradient diffusion hypothesis to relate the Reynolds stresses with average velocity and turbulent viscosity. The turbulent viscosity is determined from turbulent velocity and linear length.

In the models of two equations, the velocity values are calculated from the kinetic energy of the turbulent flow obtained by solving the transport equation. The turbulent length is determined from 2 properties of the turbulence field, the kinetic energy of the turbulent flow, and the intensity of its dispersion. The intensity of dispersion is also determined by the solution of the transport equation.

2.4.4 k- ω model

One of the advantages of the k- ω model is the consideration in the calculations of low-rinsing flows in the surface layer. The model does not involve the use of complex non-linear functions required for the k- ϵ model, which gives a more accurate and reliable result. The low-root k- ϵ model typically requires $y^+ < 0.2$ while k- ω requires $y^+ < 2$. However, in industrial currents, even $y^+ < 2$ cannot be guaranteed in most applications. Therefore, a new method for calculating the parameters of the surface layer was developed for the k- ω model. It allows making the transition from the low-rhythm flow model to the near-wall functions smoothly. Let us briefly consider some new models for calculating the parameters of the surface layer with a transition to a turbulent flow.

The k- ω model assumes that the viscosity of a turbulent flow is related to kinetic energy and friction in a turbulent flow:

$$\mu_t = \rho \frac{k}{\omega} \quad (2.52)$$

where μ_t – turbulent viscosity ($ML^{-1}T^{-1}$), ρ – density (ML^{-3}), ω – angular velocity (T^{-1}).

2.4.5 k- ω Wilcox model

This model solves 2 transport equations: the kinetic energy equation of the turbulent flow k (2.6) and the friction in the turbulent flow ω (2.7). The stress tensor is calculated from the viscosity of the turbulent flow.

$$\frac{\partial(\rho k)}{\partial t} + \nabla \cdot (\rho \mathbf{U} k) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \nabla k \right] + P_k + P_{kb} + \beta' \rho k \omega \quad (2.53)$$

$$\frac{\partial(\rho \omega)}{\partial t} + \nabla \cdot (\rho \mathbf{U} \omega) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_\omega} \right) \nabla \omega \right] + \alpha \frac{\omega}{k} P_k + P_{\omega b} + \beta' \rho \omega^2 \quad (2.54)$$

where

$$\beta' = 0.09$$

$$\alpha = 5/9$$

$$\beta = 0.075$$

$$\sigma_k = 2$$

$$\sigma_\omega = 2$$

where k – turbulence kinetic energy per unit mass (L^2T^{-2}), μ – molecular (dynamic) viscosity ($ML^{-1}T^{-1}$), μ_t – turbulent viscosity ($ML^{-1}T^{-1}$), P_k – turbulence kinetic energy production ($ML^{-1}T^{-3}$), P_{kb} – производство выталкивающей силы, ω – угловая скорость (T^{-1}), $\mathbf{U}$ – speed vector (LT^{-1}), $P_{\omega b}$ – extra buoyant member for k - ω model.

Unknown stress tensor $-\rho \overline{\mathbf{u} \otimes \mathbf{u}}$ is defined from:

$$-\rho \overline{\mathbf{u} \otimes \mathbf{u}} = \mu_t \left(\nabla \mathbf{U} + (\nabla \mathbf{U})^T \right) - \frac{2}{3} \delta (\rho k + \mu_t \nabla \cdot \mathbf{U}) \quad (2.55)$$

where $\mathbf{u}$ – fluctuating velocity component in a turbulent flow (LT^{-1}), T – static temperature (Θ),

In addition to the independent variables, the density ρ and the velocity vector $\mathbf{U}$ are treated as known values of the Navier-Stokes equation. P_k is the derived turbulence value obtained from the following equation 25 of the k - ε model:

$$P_k = \mu_t \nabla \mathbf{U} \cdot (\nabla \mathbf{U} + \nabla \mathbf{U}^T) - \frac{2}{3} \nabla \cdot \mathbf{U} (3\mu_t \nabla \cdot \mathbf{U} + \rho k) \quad (2.56)$$

For incompressible flows $\nabla \cdot \mathbf{U}$ has a small value and the second term of the right side of the equation does not make a significant contribution. For a compressible flow $\nabla \cdot \mathbf{U}$ has large values in areas with a large divergence of speed, such as shocks.

If P_{kb} takes positive values, then the definition of pushing force is included in the equation for determining k , if its calculation function is included in CFX. For its definition are used (2.53) or (2.54):

$$P_{kb} = -\frac{\mu_t}{\rho \sigma_\rho} g \cdot \nabla \rho \quad (2.57)$$

$$P_{kb} = \frac{m_t}{\rho \sigma_\rho} \rho \beta g \cdot \nabla T \quad (2.58)$$

where g – gravity vector (LT^{-2}).

It is also included in the equation ω , if the corresponding option is enabled.

$$P_{\omega b} = \frac{\omega}{k} \left((\alpha + 1) C_3 \max(P_{kb}, 0) - P_{kb} \right) \quad (2.59)$$

where $C_3 = 1$ – dissipation vector.

If the option of taking into account the direction of flows is enabled, then equation (2.56) takes the following form:

$$P_{\omega b} = \frac{\omega}{k} \left((\alpha + 1) C_3 \max(P_{kb}, 0) \cdot \sin \phi - P_{kb} \right) \quad (2.60)$$

where ϕ – angle between speed and gravity vector.

2.4.6 Baseline k- ω (BSL k- ω)

The main problem of the Wilcox model (2.57, 2.58) is high dependence on the state of free flow. Depending on the set value ω for the input stream, there is a significant difference in the results obtained. To solve this problem, Menter proposed to combine 2 models: k- ω for the solution in the near-wall region and modified k- ϵ - away from the near-wall region. Also added the corresponding equations of transition from one model to another. Thus, the Wilcox mathematical model is multiplied by the function F_1 , and the k- ϵ model is changed by the function $1-F_1$. On the border of the boundary layer and outside it, the standard k- ϵ model is used.

Wilcox model:

$$\frac{\partial(\rho k)}{\partial t} + \nabla \cdot (\rho \mathbf{U} k) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_{k1}} \right) \nabla k \right] + P_k - \beta' \rho k \omega \quad (2.61)$$

$$\frac{\partial(\rho \omega)}{\partial t} + \nabla \cdot (\rho \mathbf{U} \omega) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_{\omega 1}} \right) \nabla \omega \right] + \alpha_1 \frac{\omega}{k} P_k - \beta_1 \rho k \omega^2 \quad (2.62)$$

where

$$\beta' = 0.09$$

$$\alpha_1 = 5/9$$

$$\beta_1 = 0.075$$

$$\sigma_{k1} = 2$$

$$\sigma_{\omega 1} = 2$$

Modified k- ϵ model:

$$\frac{\partial(\rho k)}{\partial t} + \nabla \cdot (\rho \mathbf{U} k) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_{k2}} \right) \nabla k \right] + P_k - \beta' \rho k \omega \quad (2.63)$$

$$\frac{\partial(\rho \omega)}{\partial t} + \nabla \cdot (\rho \mathbf{U} \omega) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_{\omega 2}} \right) \nabla \omega \right] + 2\rho \frac{1}{\sigma_{\omega 2} \omega} \nabla k \nabla \omega + \alpha_2 \frac{\omega}{k} P_k - \beta_2 \rho k \omega^2 \quad (2.64)$$

where

$$\alpha_2 = 0.44$$

$$\beta_2 = 0.0828$$

$$\sigma_{k2} = 2$$

$$\sigma_{\omega 2} = 2$$

Now, the equations of the Wilcox mathematical model are multiplied by F_1 , the modified equations of the k- ϵ model are $1-F_1$, and the corresponding equations for k and ω are added to formulate the BSL model. Thus, taking into account the effect of lift, the BSL model takes the form:

$$\frac{\partial(\rho k)}{\partial t} + \nabla(\rho U k) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_{ka}} \right) \nabla k \right] + P_k + P_{kb} - \beta' \rho k \omega \quad (2.65)$$

$$\begin{aligned} \frac{\partial(\rho \omega)}{\partial t} + \nabla \cdot (\rho U \omega) = & \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_{\omega a}} \right) \nabla \omega \right] + (1 - F_1) 2\rho \frac{1}{\sigma_{\omega 2} \omega} \nabla k \nabla \omega + \\ & + \alpha_3 \frac{\omega}{k} P_k + P_{\omega b} - \beta_3 \rho k \omega^2 \end{aligned} \quad (2.66)$$

The coefficient α in the equations for determining $P_{\omega b}$ replaced by α_3 .

Model coefficients are a linear combination of the corresponding base model coefficients.

$$\Phi_3 = F_1 \Phi_1 + (1 - F_1) \Phi_2 \quad (2.67)$$

where Φ_1 – represents all the constants from the original model, Φ_2 – represents constants from a modified model k- ϵ , Φ_3 – new model coefficients.

2.4.7 Shear Stress Transport (SST)

The turbulence model SST (Shear Stress Transport) is based on the k- ω model for the distribution of turbulent stress and provides more accurate predictions of the onset of separation and flow rate with adverse pressure drops.

The BSL model combines the benefits of the k- ϵ and Wilcox models. However, it still does not allow to determine the beginning of the flow separation from the smooth surface. The main reason for this is that both models do not take into account the turbulent stress in the transport equation. This results in overestimated viscosity values of the swirling flow. The correct transport equation can be obtained by

introducing a constraint to determine the turbulent viscosity:

$$\nu_t = \frac{a_1 k}{\max(a_1 \omega, SF_2)} \quad (2.68)$$

$$\nu_t = \mu_t / \rho \quad (2.69)$$

Where S – absolute value of turbulence, F_2 equal to 1 for flows in the border layers or 0 for free layers.

Also F_2 is a n interface function similar to F_1 , which imposes restrictions on (2.64) for the near-wall layer, and the underlying assumptions are not true for free vortex flows. S - invariant measure of propagation velocity.

Interface functions are important for this method and depend on the distance to the nearest surface and the flow variables.

$$F_1 = \tanh(\arg_1^4) \quad (2.70)$$

where

$$\arg_1 = \min \left(\max \left(\frac{\sqrt{k}}{\beta' \omega y}, \frac{500\nu}{y^2 \omega} \right), \frac{4\rho k}{CD_{k\omega} \sigma_{\omega 2} y^2} \right) \quad (2.71)$$

where y – distance to the nearest wall, ν kinematic viscosity and

$$CD_{k\omega} = \max \left(2\rho \frac{1}{\sigma_{\omega 2} \omega} \nabla k \nabla \omega, 1.0 \times 10^{-10} \right) \quad (2.72)$$

$$F_2 = \tanh(\arg_2^2) \quad (2.73)$$

$$\arg_2 = \max \left(\frac{2\sqrt{k}}{\beta' \omega y}, \frac{500\nu}{y^2 \omega} \right) \quad (2.74)$$

where F_1 is equal to 0, if $y^2 > 70$.

2.5 CFD model mesh parameters and boundary conditions

2.5.1 Mesh Parameters

The construction of the computational mesh to optimize the flow part of the ejector is as follows:

For CFD modeling, a tetrahedral mesh with prismatic elements was used, with the

following parameters:

1. Physics preference – CFD
2. Mesh method – Patch Independent
3. Use Advanced Size function – On: Proximity and Curvature.
4. Relevance center – Fine
5. Smoothing - high
6. Transition – Slow
7. Curvature normal angle – 18°
8. Minsize – 0.25 mm
9. Growth rate – 1.2
10. Automatic Inflation – Program Controlled
11. Inflation Option – First Layer Thickness
12. First Layer Height – 0.025 mm
13. Maximum Layers – 5
14. Growth Rate – 1.2
15. Inflation Algorithm – Pre
16. Collision Avoidance – Stair Stepping
17. Gap Factor – 0.5
18. Maximum Height over Base – 0.1
19. Growth Rate – Geometric
20. Maximum Angle – 180°
21. Fillet Ratio – 1
22. Use Post Smoothing – Yes, Smoothing Iterations – 10.
23. Shape Checking – CFD

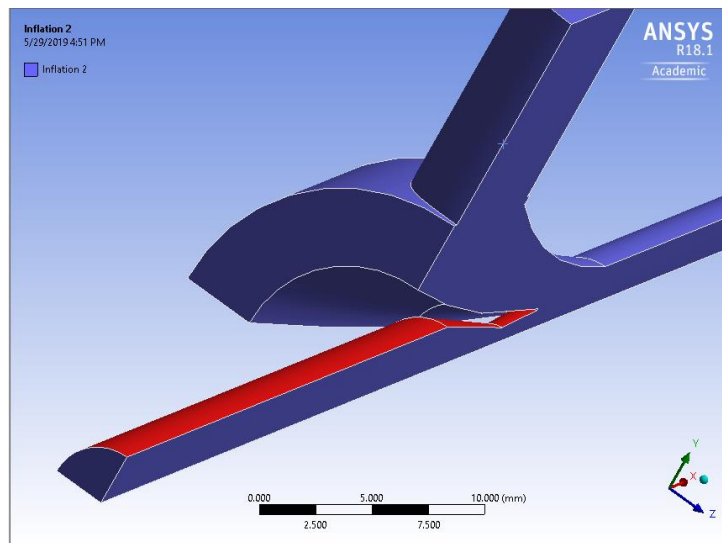
Using BodySizing for the receiving chamber, where there is an acceleration of the ejected flow and acceleration of the working flow in the diffuser part of the nozzle, as well as the mixing chamber, where compression shocks are observed, the dimensions of the mesh elements, are different from the basic dimensions indicated above.

Inflation was made for two areas: nozzle walls and walls of the suction chamber, mixing chamber, and diffuser (Fig. 2.21).

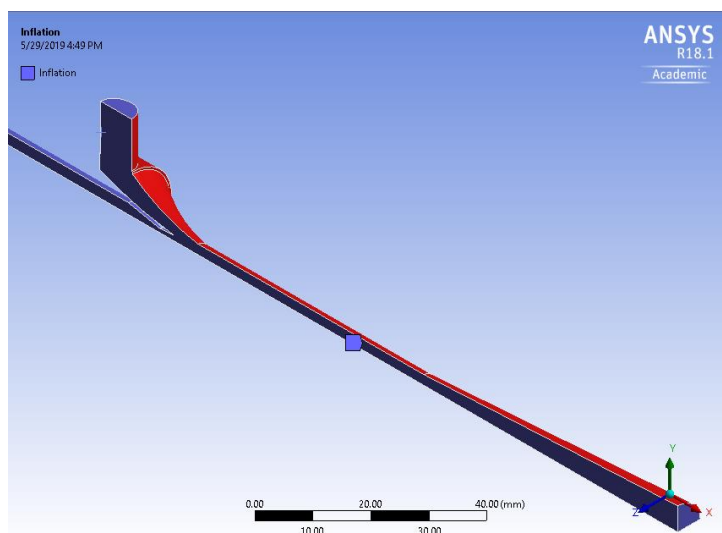
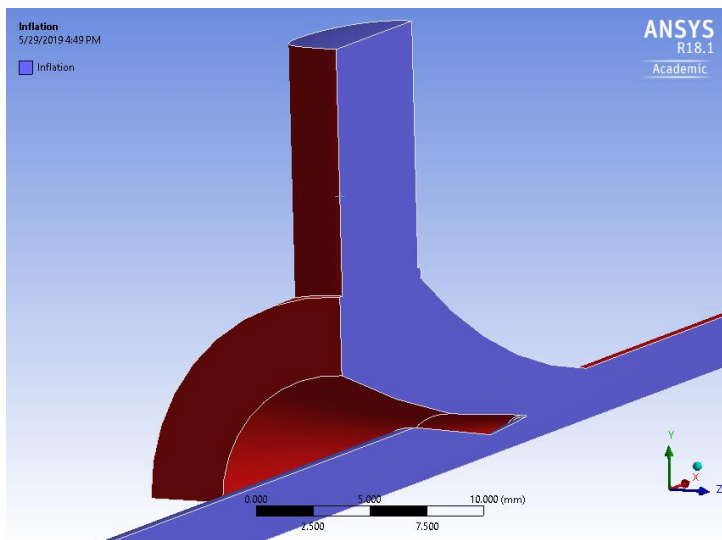
For the wall separating the ejected and the working flow at the nozzle exit, Face Sizing with Element sizing is set to 0.03 (Fig. 2.22)

The result of the mesh (Fig. 2.23): the number of points - 2723121, elements - 1684499. The minimum Aspect Ratio value is 1.16, and the maximum is 19.5, the average is 2.5.

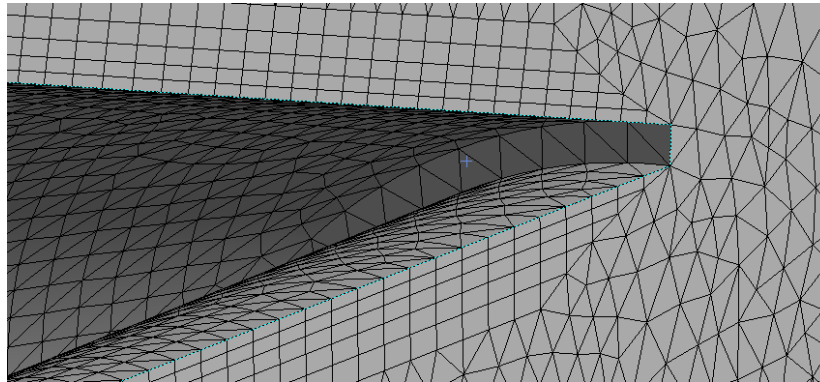
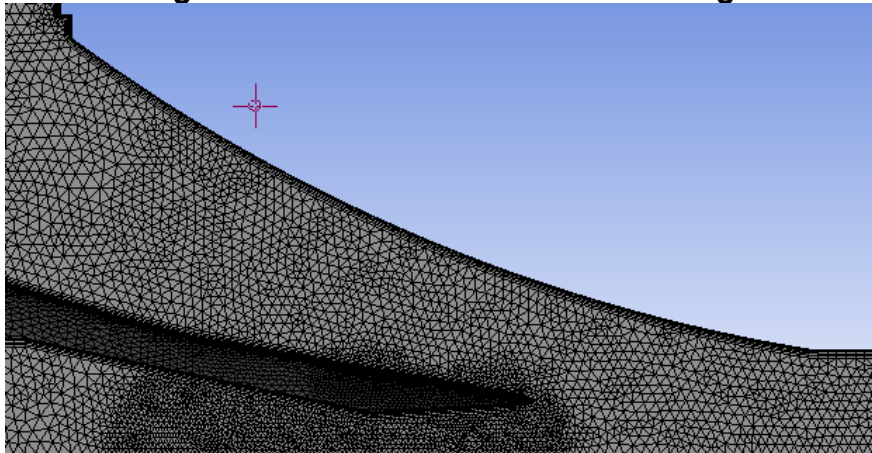
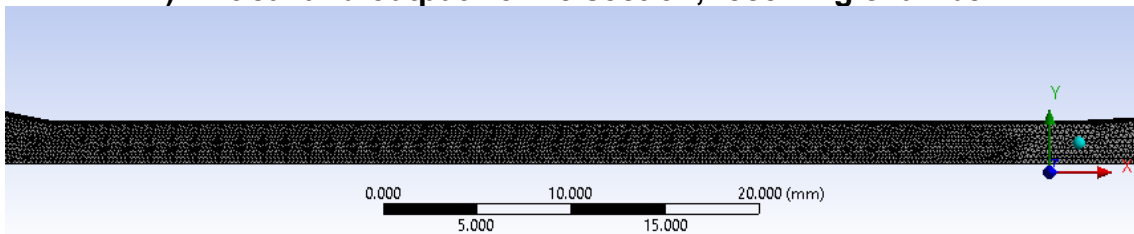
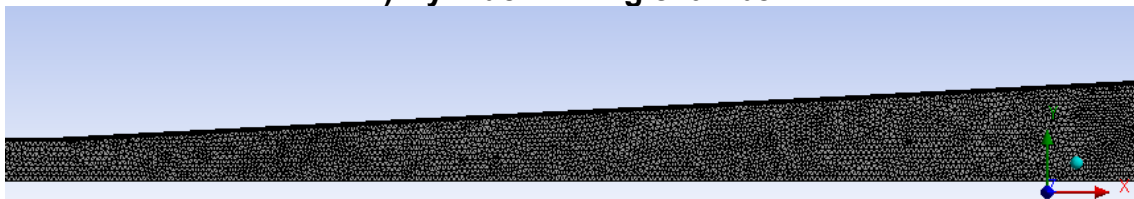
The values of Skewness lie in the range of 0.002 - 0.56; the average value is 0.18.



A) Inflation area of nozzle.



B) Inflation area of suction chamber, mixing chamber and diffuser.

Figure 2.21 Inflation areas**Figure 2.22 Area of local mesh resizing.****A) Critical and output nozzle section, receiving chamber.****B) Cylinder mixing chamber****C) Diffuser****Figure 2.23 Calculated mesh**

2.5.2 Boundary Conditions

Following parameters were specified for a CFD model:

1. CFD model designed for 1/4 of 3-D ejector model with 2 symmetry planes.
2. Working fluid properties specified as real gas. Butane is selected as constraint option. R1233zd(E) is a transport equation.

3. Fluid domain is specified as continuous fluid, non-buoyant. Reference pressure 0 [Pa]
4. Total energy is selected as a heat transfer model.
5. Wall parameters as adiabatic heat transfer, wall roughness – smooth wall.
6. Turbulence model SST.
7. Inlet boundary parameters specified as subsonic flows at static temperature, static pressure and mass fraction for both working and secondary fluids.
8. Outlet boundary contain is a static pressure.

CFD modeling was performed for single fluid ejector operating on R142b that was tested and described by Buyadgie O. (PhD thesis, 2016), pure R1233zd(E), binary Steam/Air mixture as an ideal case, and refrigerant mixtures R11/Butane and R1233zd(E)/Butane.

2.6 CFD Modeling results analysis

Detailed analysis of CFD modeling results was performed based on various graphs that represent the dependence of flow parameters along with axis position. Each graph contains 25 streamlines for primary and secondary flow.

2.6.1 Velocity and Mach number distribution.

According to flow pattern, there is no complete mixing of the flows in ejector flow part, it is inherent only by a small boundary zone, which increases approaching the exit of diffuser. Momentum transfer occurs with a more or less elastic collision, which suggests that this process is more reversible than an absolutely inelastic impact. In addition, the direct shock wave does not lead to complete mixing. It reduces the additional losses. It could be stated that the core of the cross-section remains supersonic up to entering the diffuser, where pressure restoring due to speed decrease. In order to provide a comparative analysis of single and binary fluid ejectors, CFD model of R142b, R11/Butane, Steam/Air, R1233zd(E), R1233zd(E)/Butane was developed. The Mach number of the boundary layer between the core and periphery in the mixing chamber is greater than 1 for R142b (see Fig. 2.24). In the binary fluid ejector, over the entire mixing chamber, the boundary layer is supersonic, and Mach number is almost constant (see Fig 2.25, Fig. 2.26) Mach number in binary fluid ejector on Fig. 2.28 decreases through the mixing chamber, that is similar to single fluid ejector flow pattern.

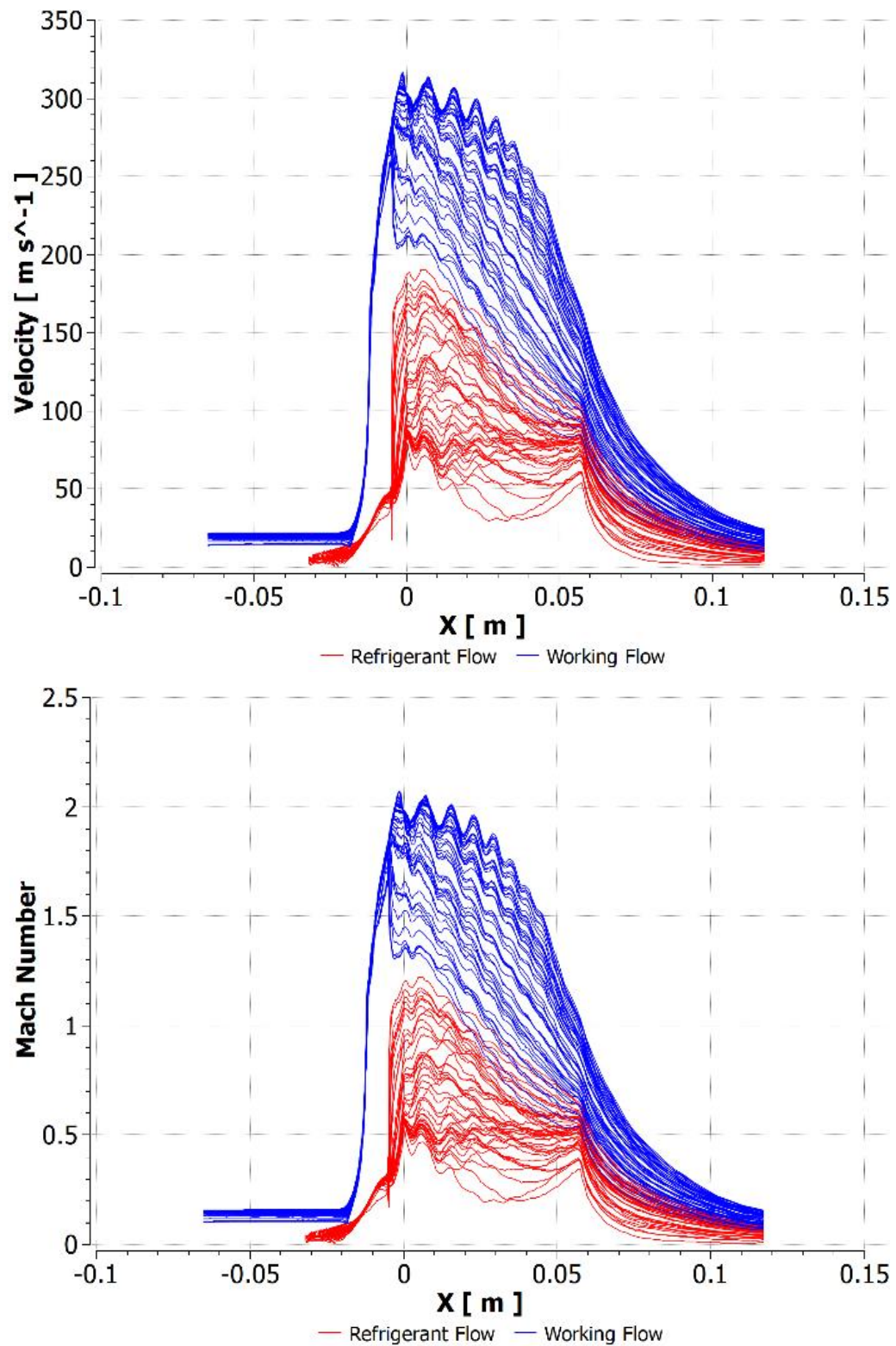


Figure 2.24 Velocity chart and Mach number chart of R142b ejector operating on $t_{\text{gen}}=85^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}\text{C}$, $t_{\text{eva}}=12^{\circ}\text{C}$.

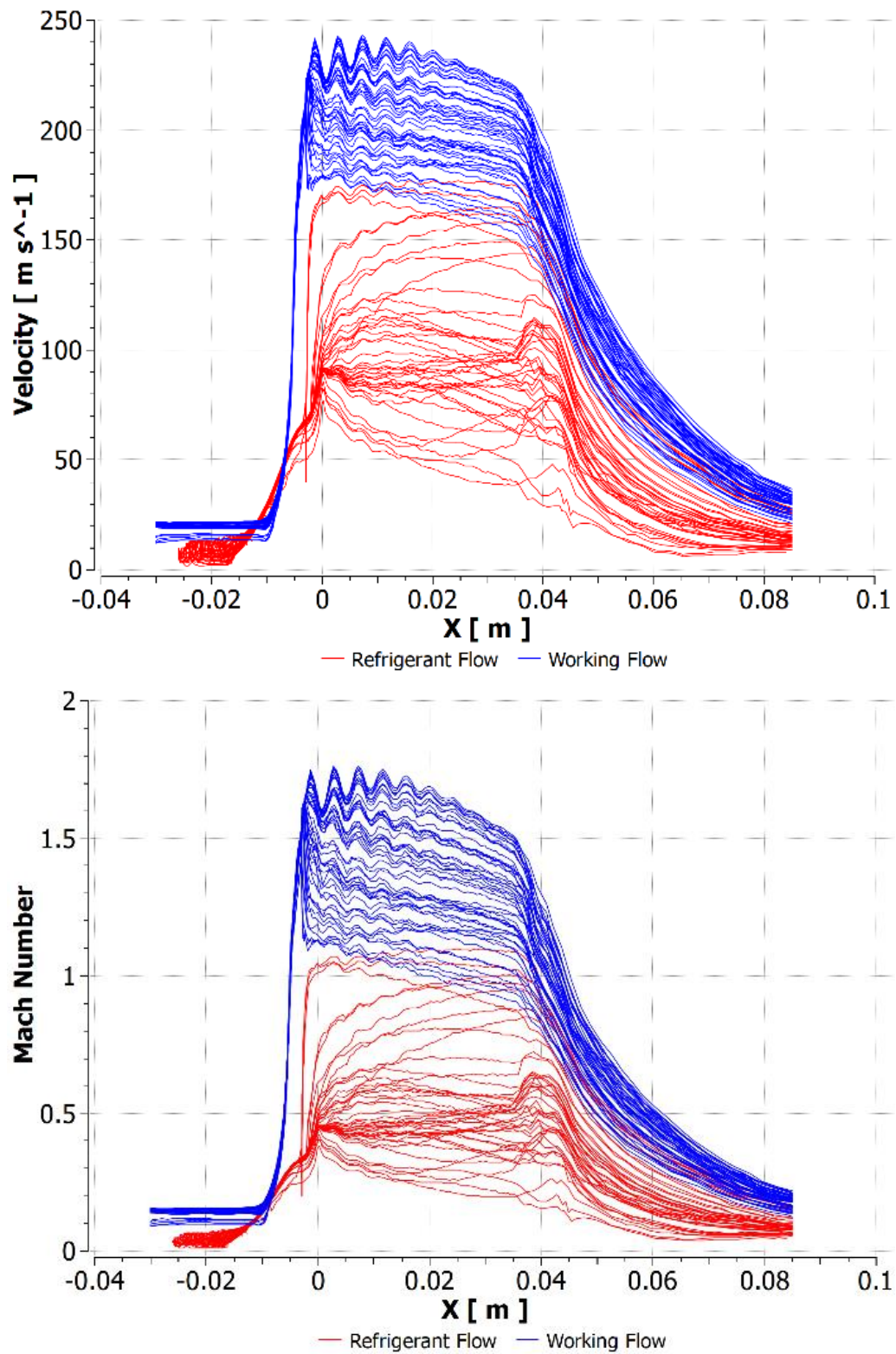


Figure 2.25 Velocity chart and Mach number chart of R11/Butane ejector operating on $t_{\text{gen}}=85^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}\text{C}$, $t_{\text{eva}}=12^{\circ}\text{C}$.

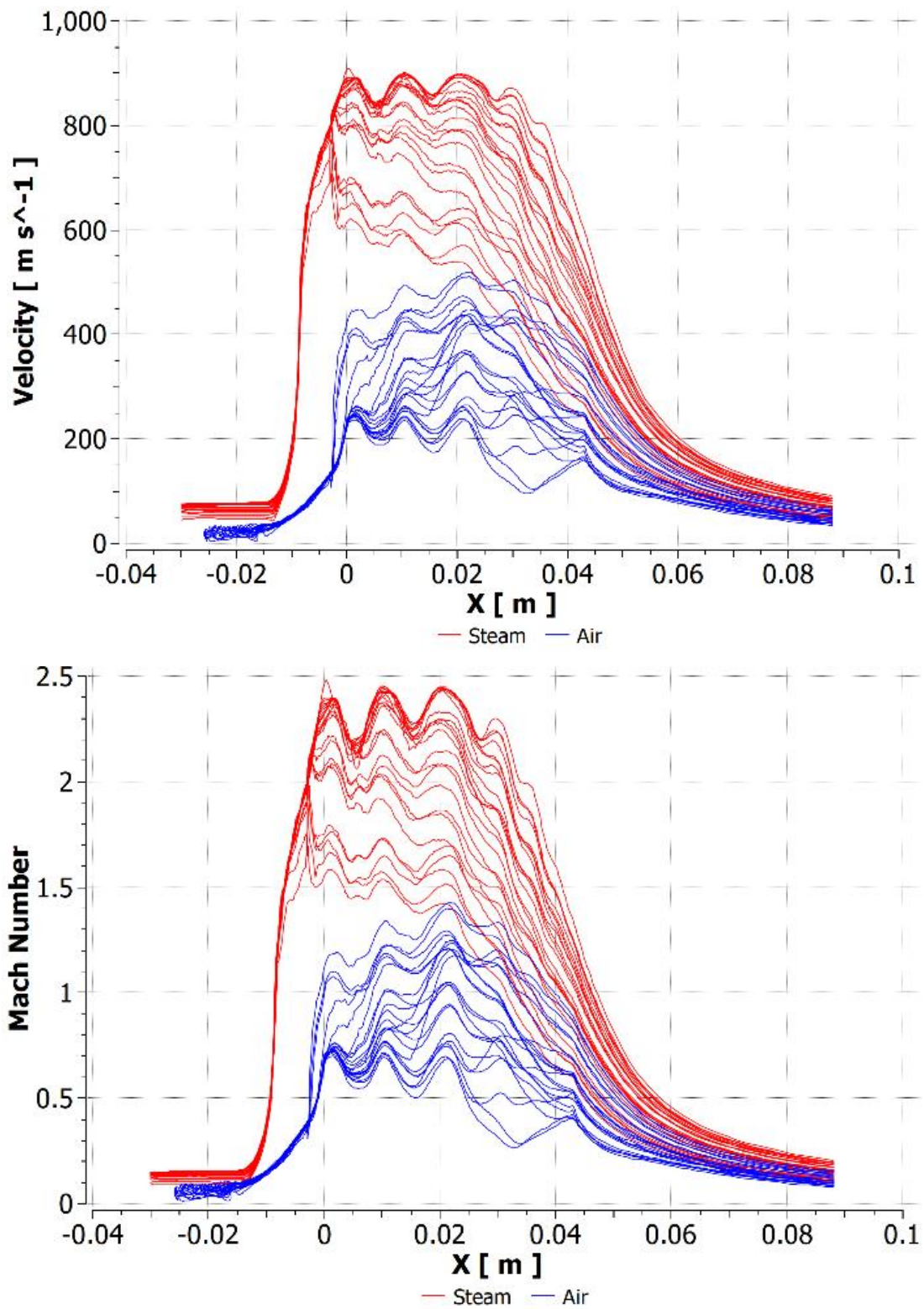


Figure 2.26 Velocity chart and Mach number chart of Steam/Air ejector operating on $t_{\text{gen}}=150^{\circ}\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^{\circ}\text{C}$, $P_{\text{eva}}=55\text{kPa}$.

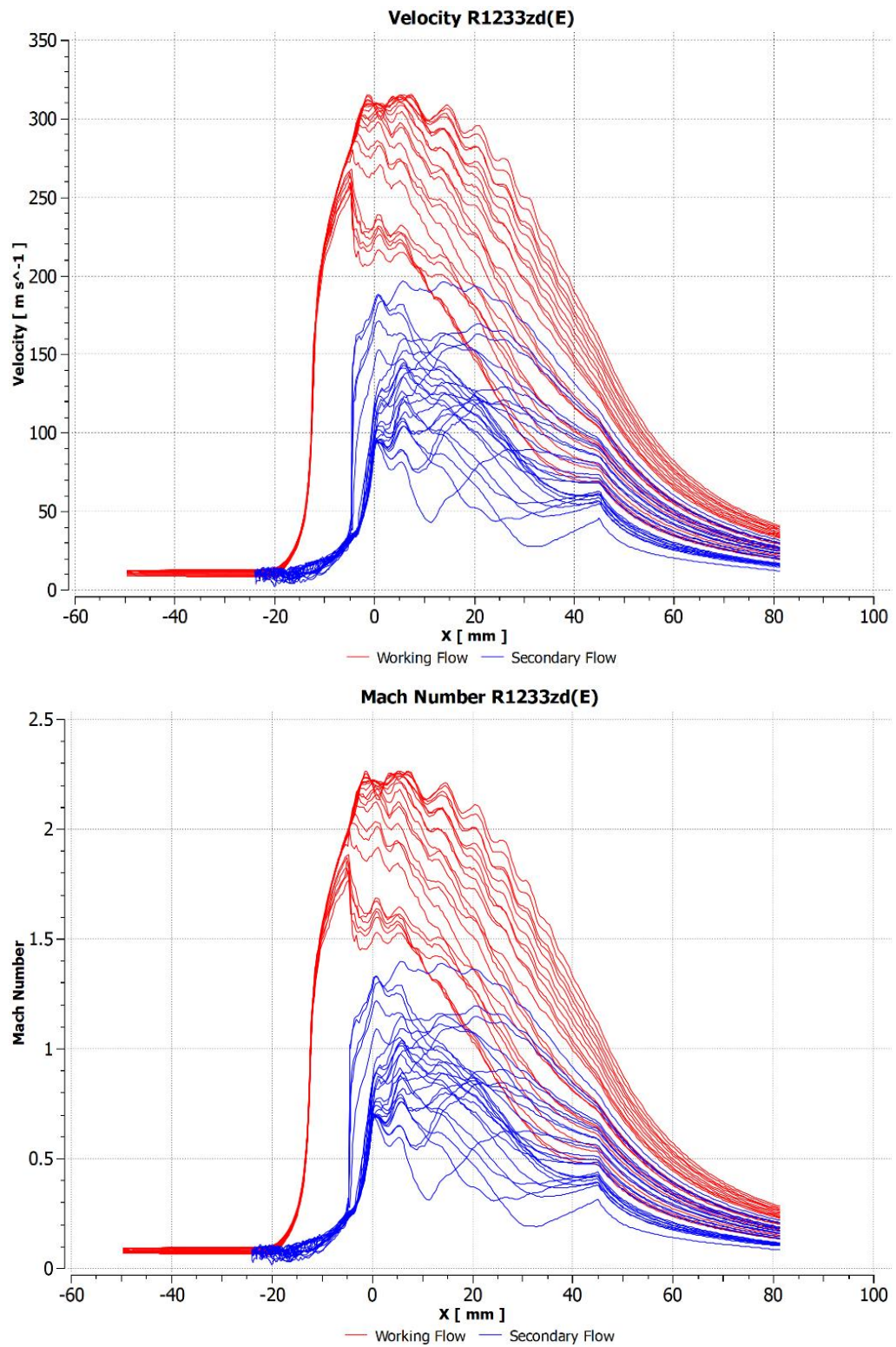


Figure 2.27 Velocity and Mach number chart of R1233zd(E) ejector operating on $t_{\text{gen}}=90^\circ$, $t_{\text{cond}}=35^\circ\text{C}$, $t_{\text{eva}}=15^\circ\text{C}$.

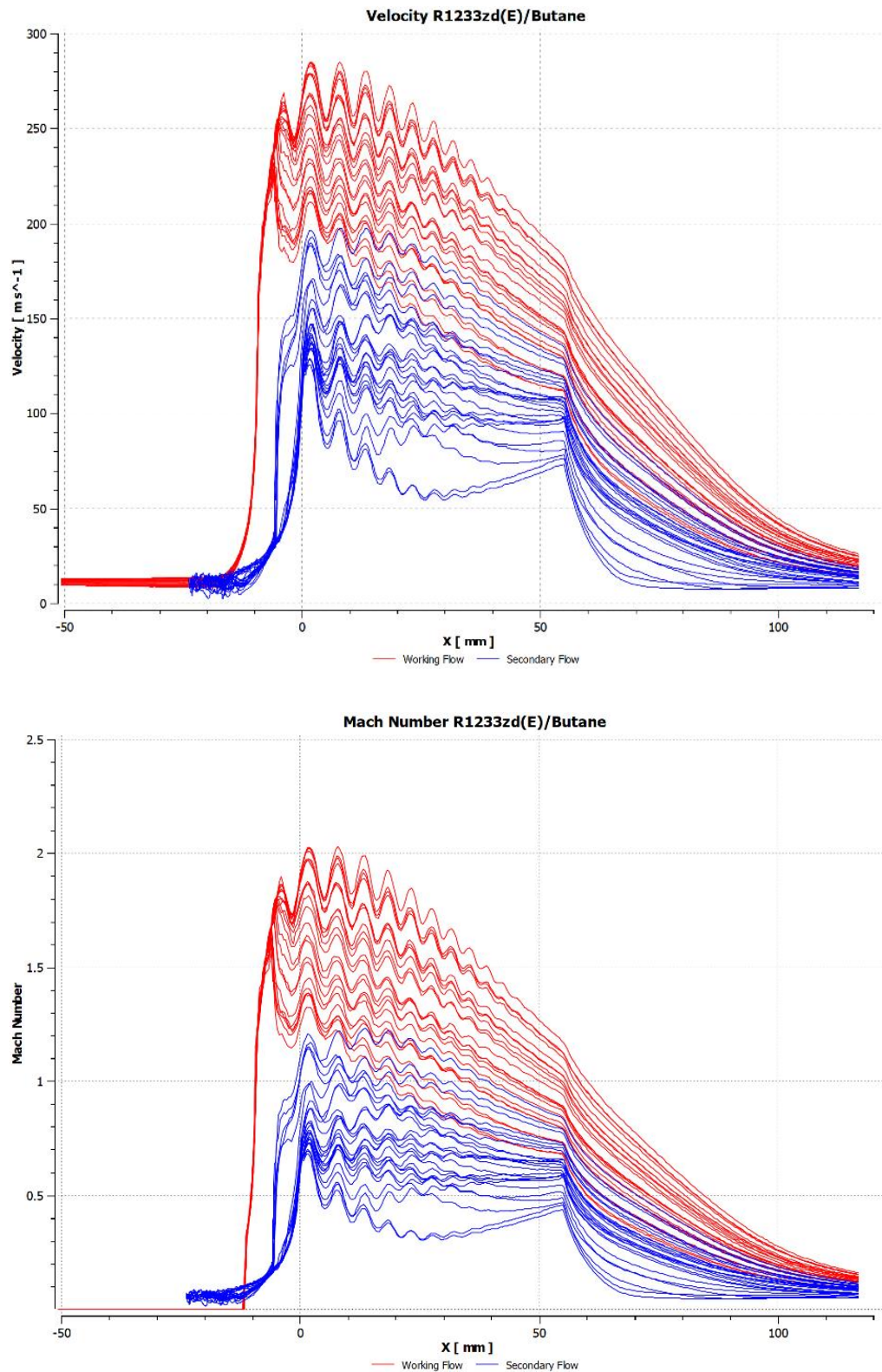


Figure 2.28 Velocity and Mach number chart of R1233zd(E)/Butane ejector operating on $t_{\text{gen}}=90^\circ$, $t_{\text{cond}}=35^\circ\text{C}$, $t_{\text{eva}}=15^\circ\text{C}$, $X_{\text{gen}}=1$, $X_{\text{eva}}=0.3$

2.6.2 Pressure distribution

The character of pressure changes along the flow part for binary and single fluid ejector differs only by quantity (see Fig 2.29-2.33). The shape of pressure curves is approximately the same. There are periodic oscillations of static pressures are observed along the mixing chamber. It can be approximated by harmonic dependence, and remaining section pressure dependence can be described by one dimensional equations.

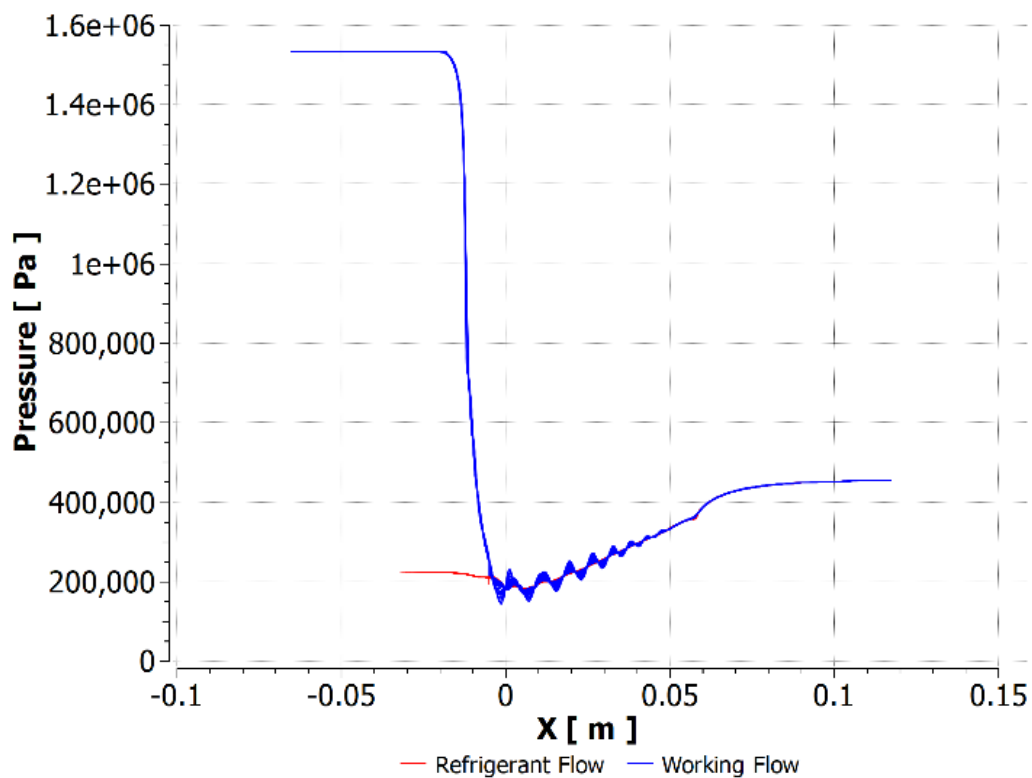


Figure 2.29 Pressure chart of R142b.

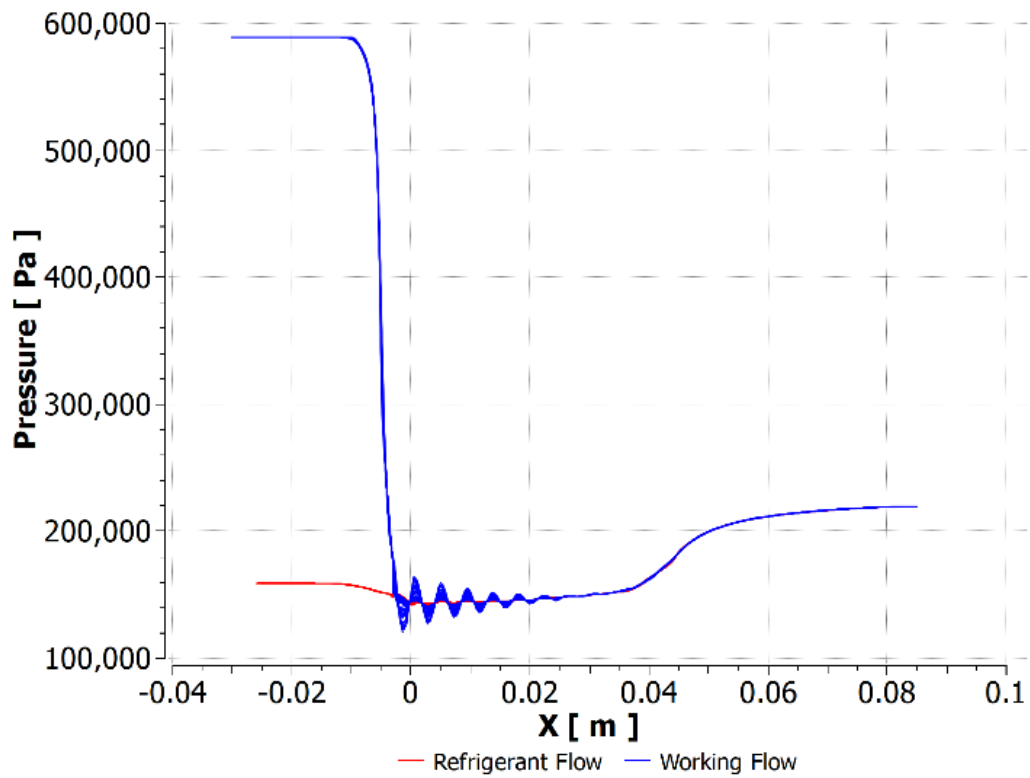


Figure 2.30 Pressure chart of R11/Butane

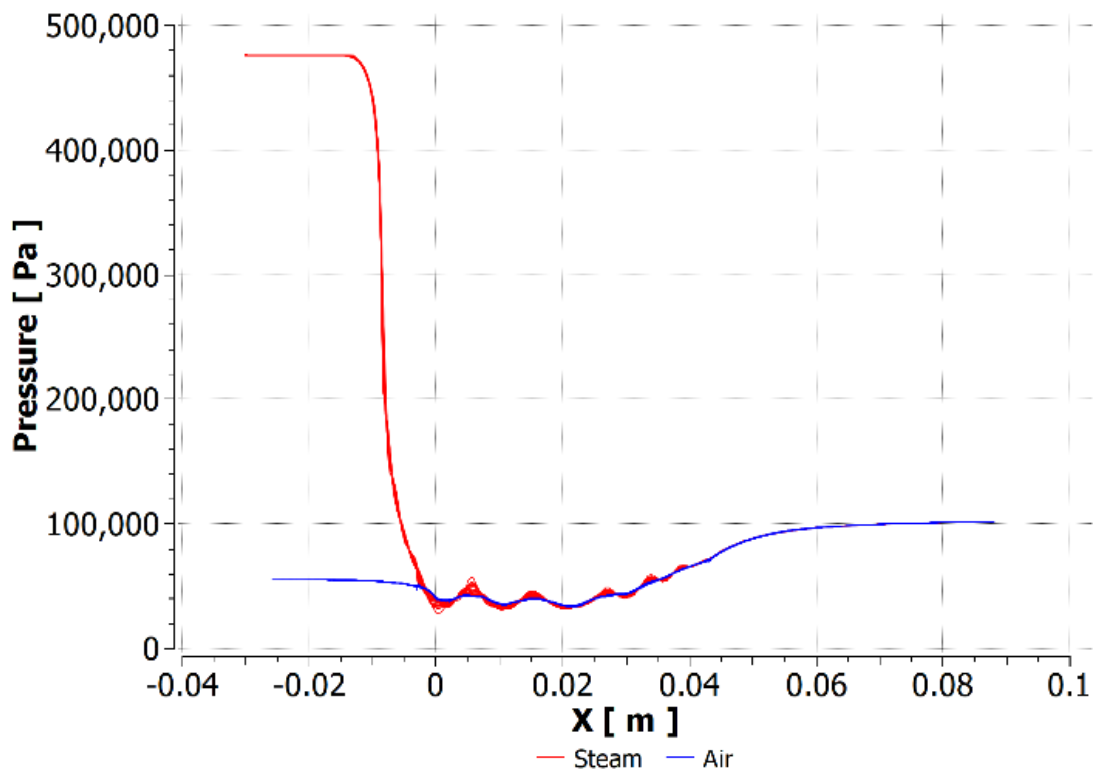


Figure 2.31 Pressure chart of Steam/Air at $t_{\text{gen}}=150^{\circ}\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^{\circ}\text{C}$, $P_{\text{eva}}=55\text{kPa}$.

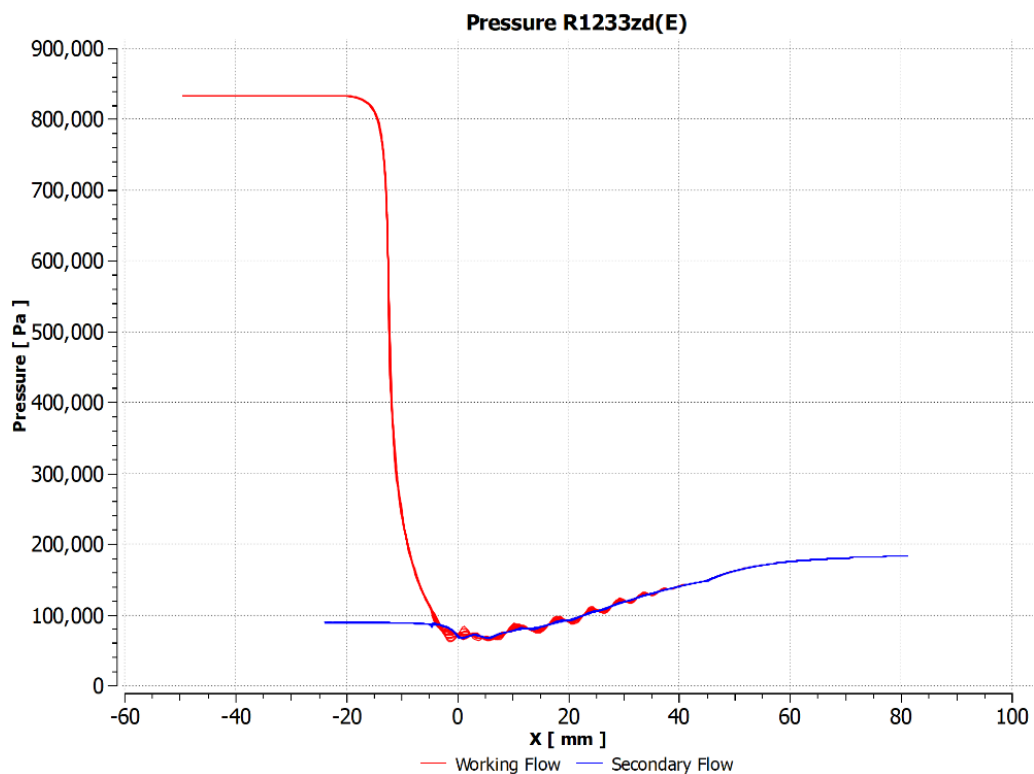


Figure 2.32 Pressure Chart of R1233zd(E)

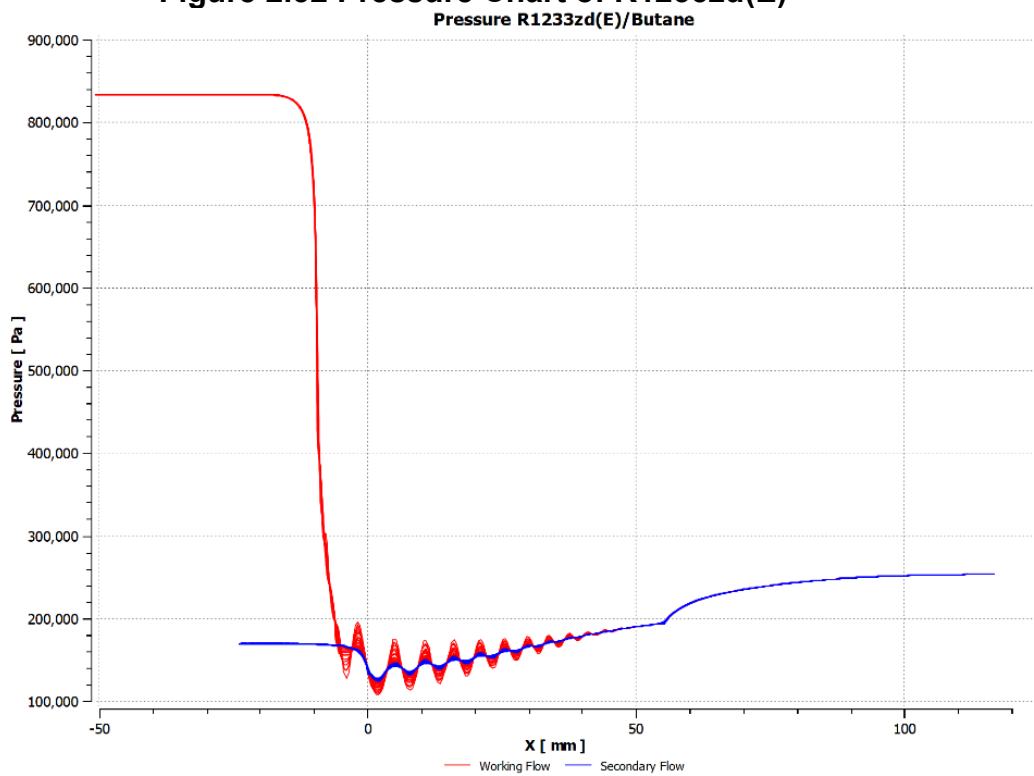


Figure 2.33 Pressure Chart of R1233zd(E)/Butane

2.6.3 Static Entropy

Large differences are observed for static enthalpy of primary flow in binary fluid and single fluid ejectors. The sharp rise of static entropy of working flow and relatively slight of entrained flow are observed for R142b (Fig. 2.34) at the inlet cross-section to the mixing chamber. Along with the mixing, chamber entropy rises, and then monotonously decreases to intermediate values almost equal to initial values. For binary fluid, sharp peaks of entropy increase observed at inlet cross-section are of mixing chamber. The entropy of entrained flow rises to conditional maximum and decreases slightly before diffuser. In the diffuser, it increases and reaches values higher than initial entropies of the flows (Fig. 2.35). This increase in entropy could be caused by the irreversibility of the mixing process and release of mixing heat. In Fig. 2.36, the entropy of the mixture practically reaches the entropy of the ejected flow but does not exceed it. It suggests that the entropy of the mixture is closer to the additive value than in the previous mixture. Generally speaking, the difference between the actual static entropy of the mixture and its additive quantity is the heat of mixing. For pure substances, the mixing process can be considered adiabatic, while for a mixture of substances it is necessary to take into account the heat of mixing, which adds additional irreversibility to the work of the ejector (Fig. 2.37 and 2.38).

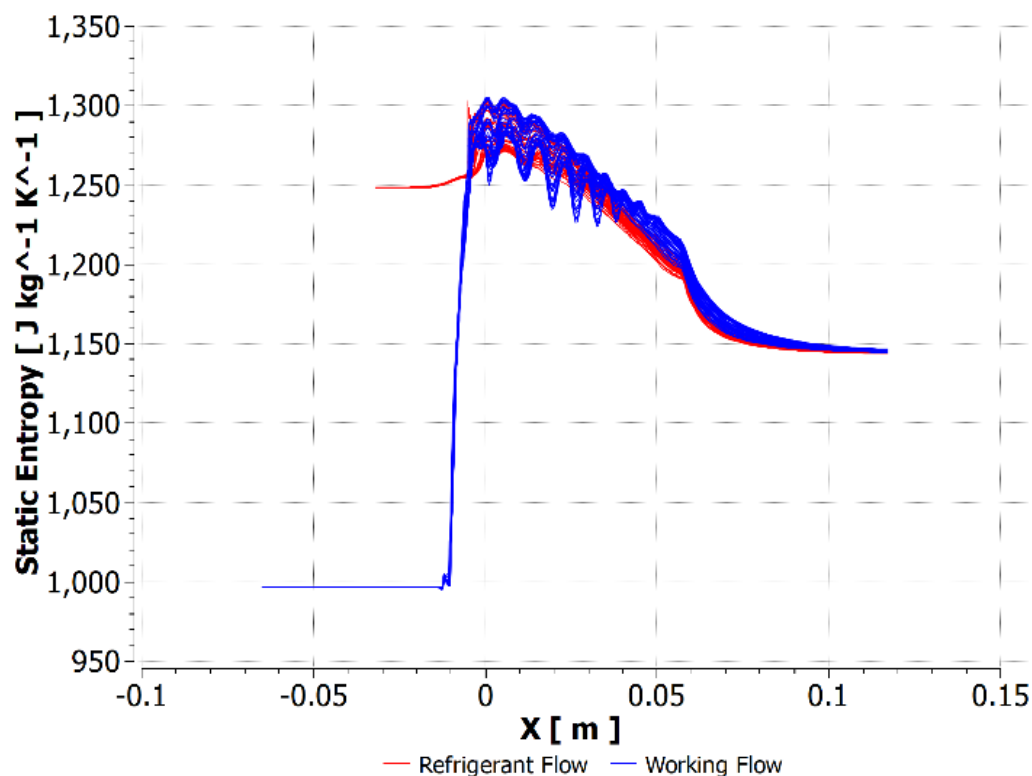


Figure 2.34 Static Entropy chart of R142b

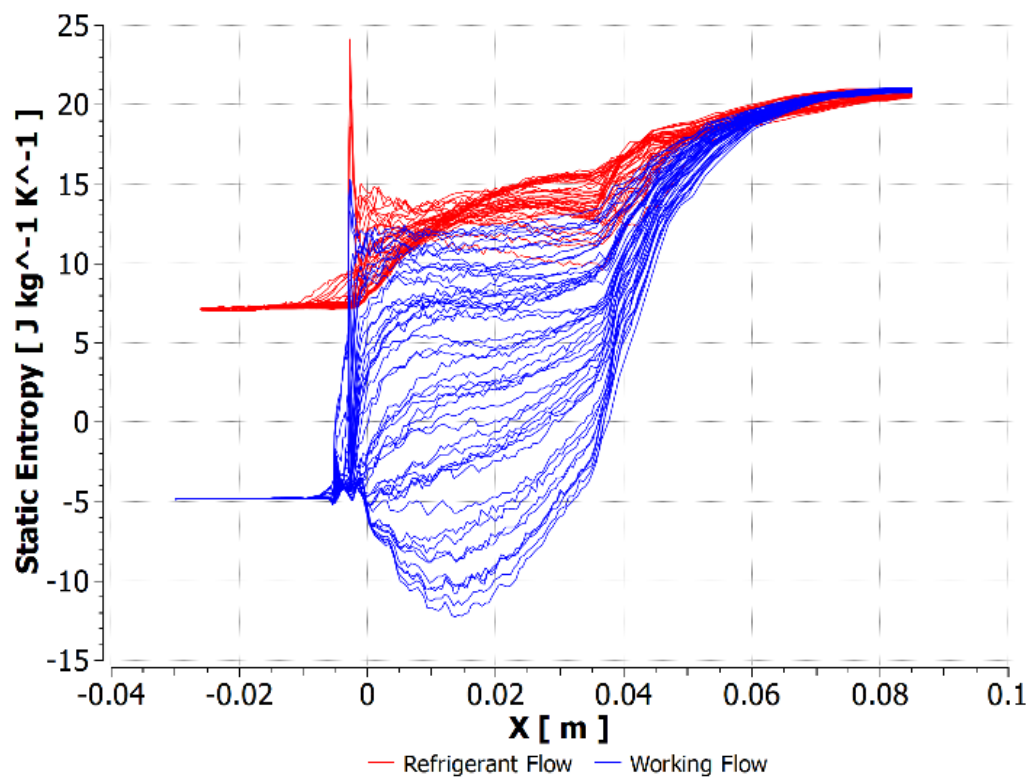


Figure 2.35 Static Entropy chart of R11/Butane

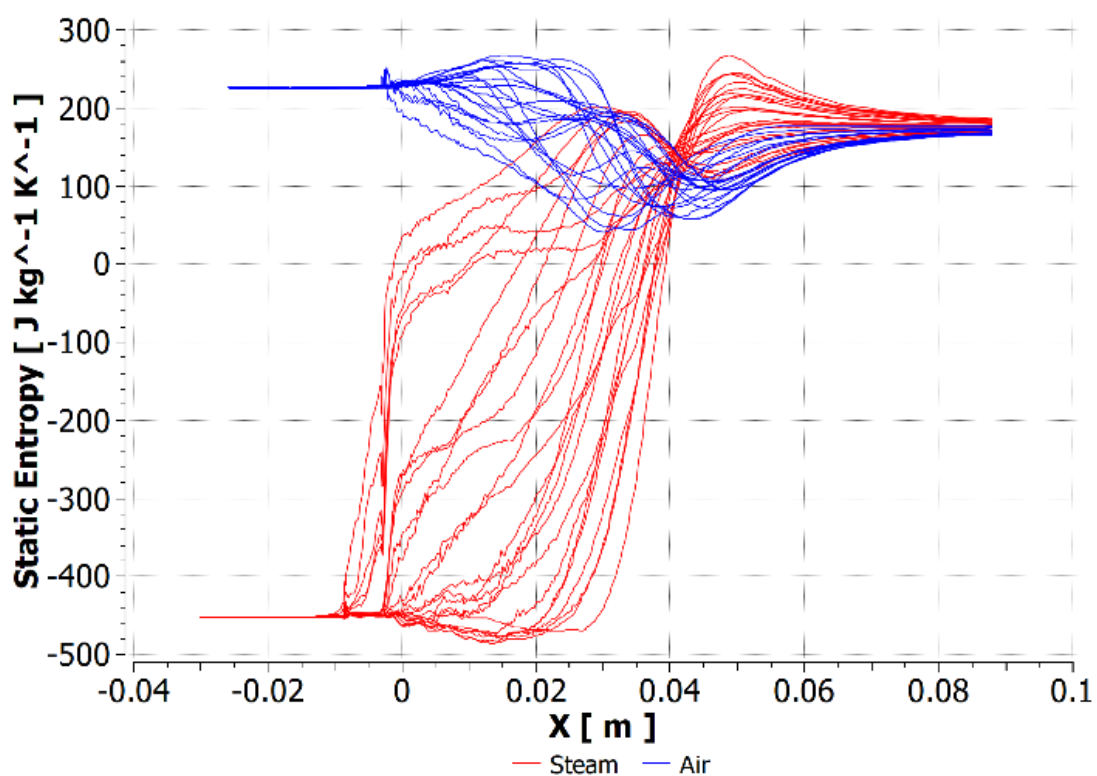


Figure 2.36 Static Entropy chart of Steam/Air

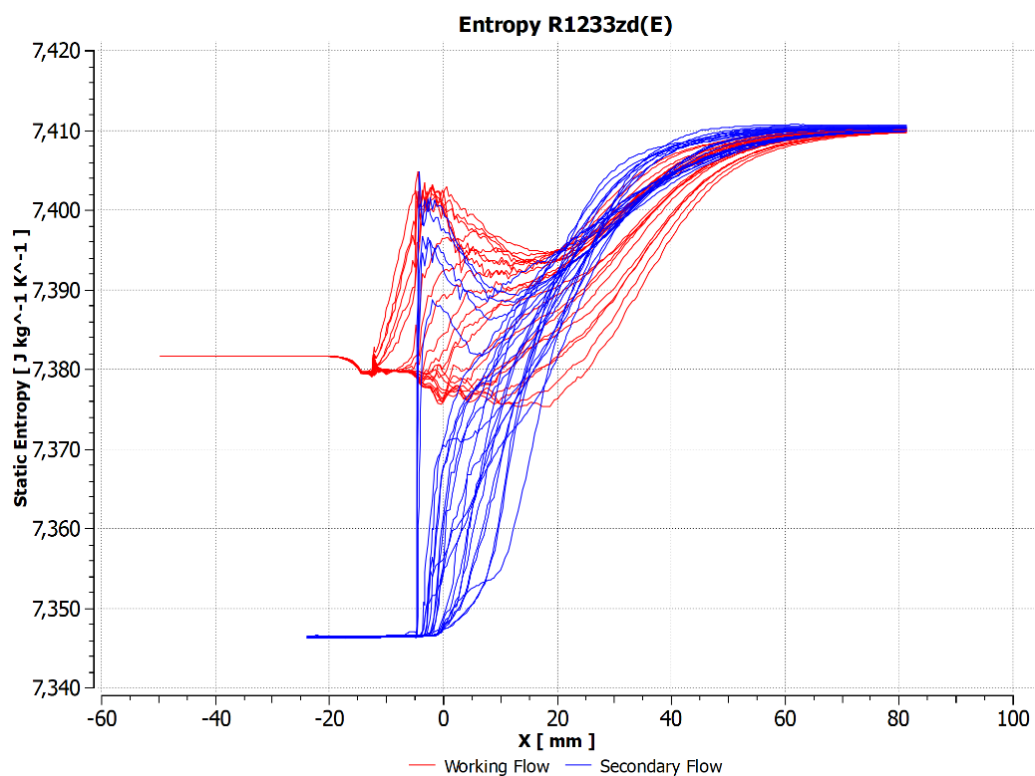


Figure 2.37 Static Entropy chart of R1233zd(E)

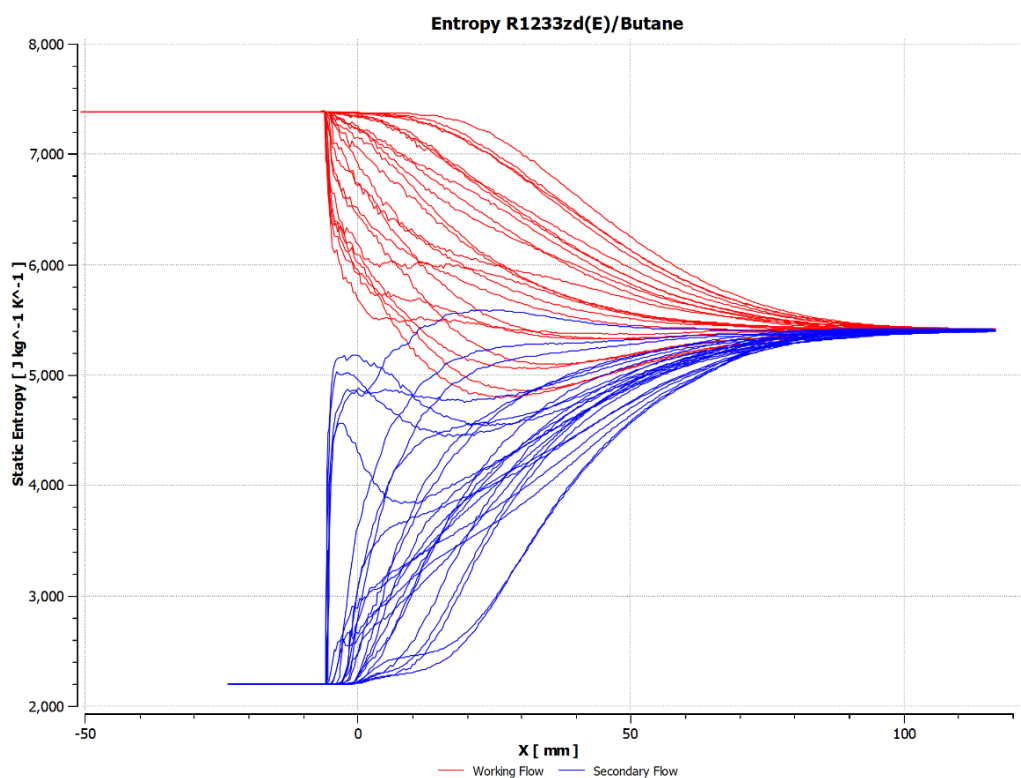


Figure 2.38 Static Entropy of R1233zd(E)/Butane

It is considered that flow processes in real ejector are irreversible, that means that entropy of the system at the end of the mixing process is significantly higher than at the beginning. However, in some cases it can be noted that entropy values at these points differ only by 1-5%, i.e., processes in ejector not only adiabatic but also isentropic. That is true for state function since changes of entropy determined by boundary conditions, not on the nature of the process. In the case of steam or steam/air mixture entropy difference is significant and reaches up to 50% (Table 2.1).

Table 2.1. Entropy values of working and secondary flows.

Fluid	U	s_{in} J/(kg K)	s_{out} , J/(kg K)
R142b	0.48	1600	1700
R11/Butane	0.55	-0.875	29.8
Steam/Air	1	-215	360
R1233zd(E)	0.575	11570	11633
R1233zd(E)/Butane	0.63	8764	8748

Providing comparison of entropy change during the mixing, it can be seen that in all cases, the entropy of working and secondary flow changes in counterflow mode, approach each other. That increases the irreversibility of one component and decreases others. It is connected to a vicious exchange between flow layers, that was observed by Lamberts et al. [22]. Additional study should be conducted on a wide range of single and binary fluids since the conclusion cannot be considered at this stage.

Entropy change is evaluated as:

$$\Delta S = \int \frac{\delta Q}{T} dt, \text{ entropy change depends on heat flows in the thermodynamic system.}$$

Fluid flow in ejector can be considered as adiabatic, but taking into account high velocities i.e., infinite small heat inleaks from the outside. However, internal heat gain processes are observed between flow and walls and in viscous friction of flow layers at different velocities. Also, the mixing process between heterogeneous fluids releases or absorbs mixing heat. Shock waves result in a transformation of kinetic energy into heat, since an impact in suction and mixing chamber can be considered as an inelastic. As a result, the influence of various factors leads to different patterns of entropy change. For a clean result of entropy change, it is necessary to investigate a broad set of parameters and fluids.

2.6.4 Density

Fig. 2.39-2.43 represents the density change in ejector working on single and binary fluid. Fig. 2.39 represents the density change in ejector working on R142b. Fig. 2.40 represents the density change in ejector working on R11/Butane. Fig. 2.41 represents the density change in ejector working on Steam/Air. Fig. 2.42 and 2.43 represent the density change in ejector working on R1233zd(E) and R1233zd(E)/Butane.

For single fluid ejector, the density is changing through the whole mixing chamber. The density values of working and refrigerant flow are almost equal. For a binary fluid ejector, the mixing process is conducted at relatively constant density, that does not exceed the density of working flow. Slight density increasing is observed from the outlet of mixing chamber during the pressure recovery process in the diffuser.

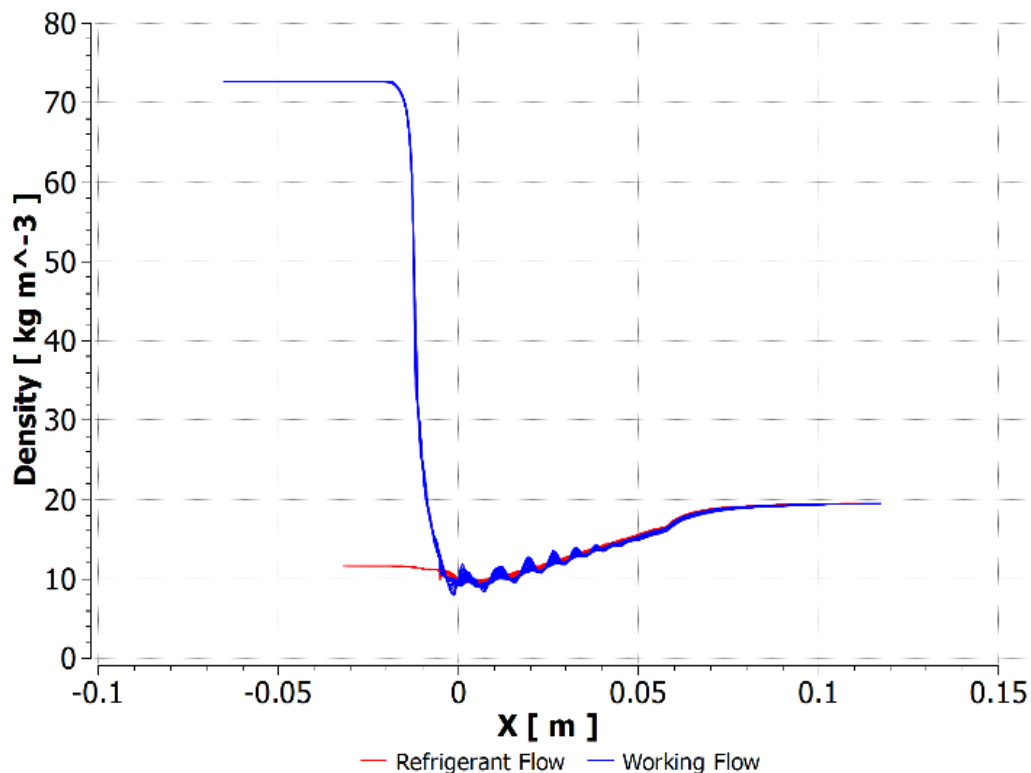


Figure 2.39 Density chart of R142b.

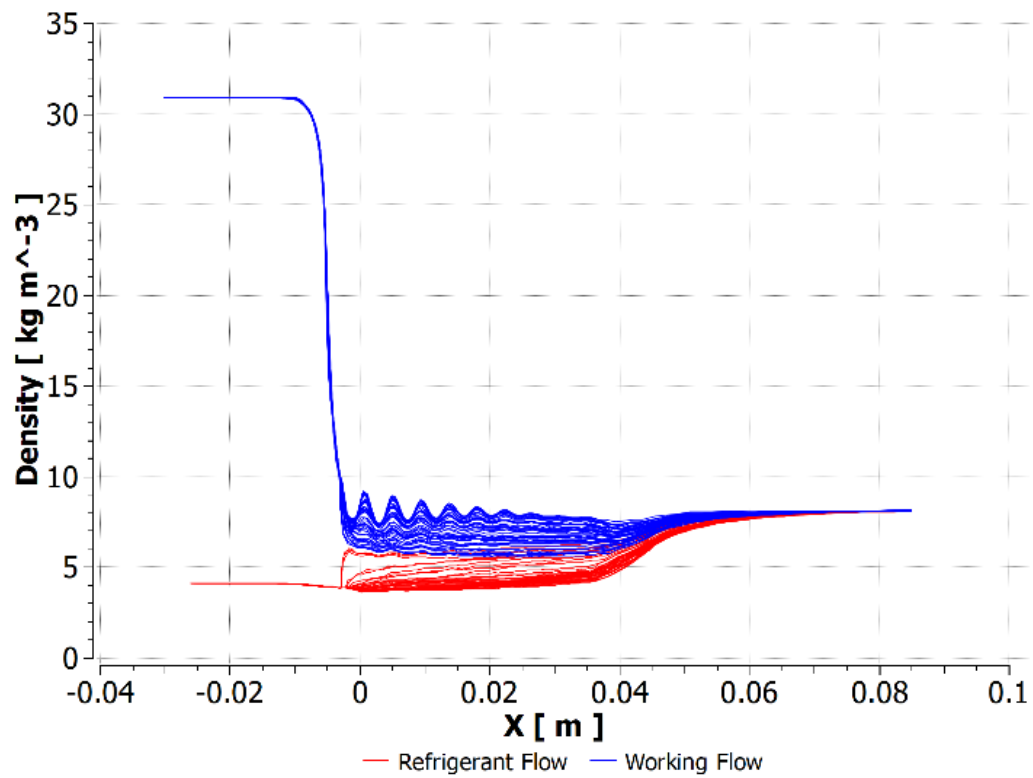


Figure 2.40 Density chart of R11/Butane .

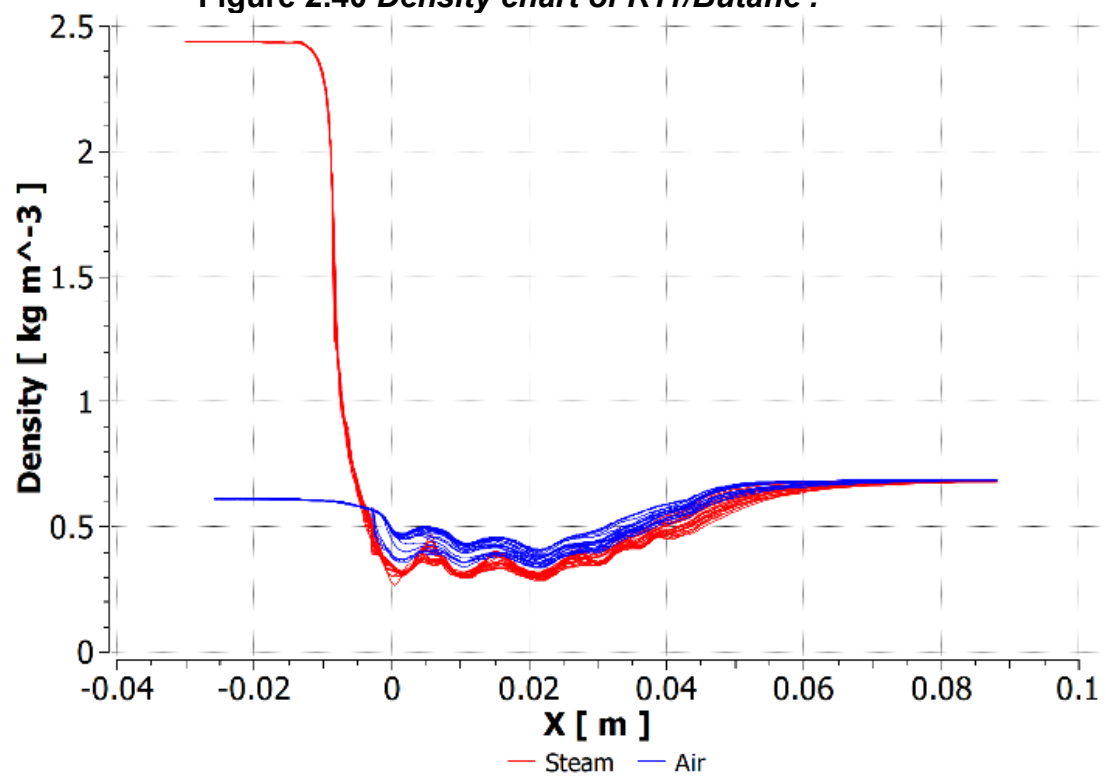


Figure 2.41 Density chart of Steam/Air

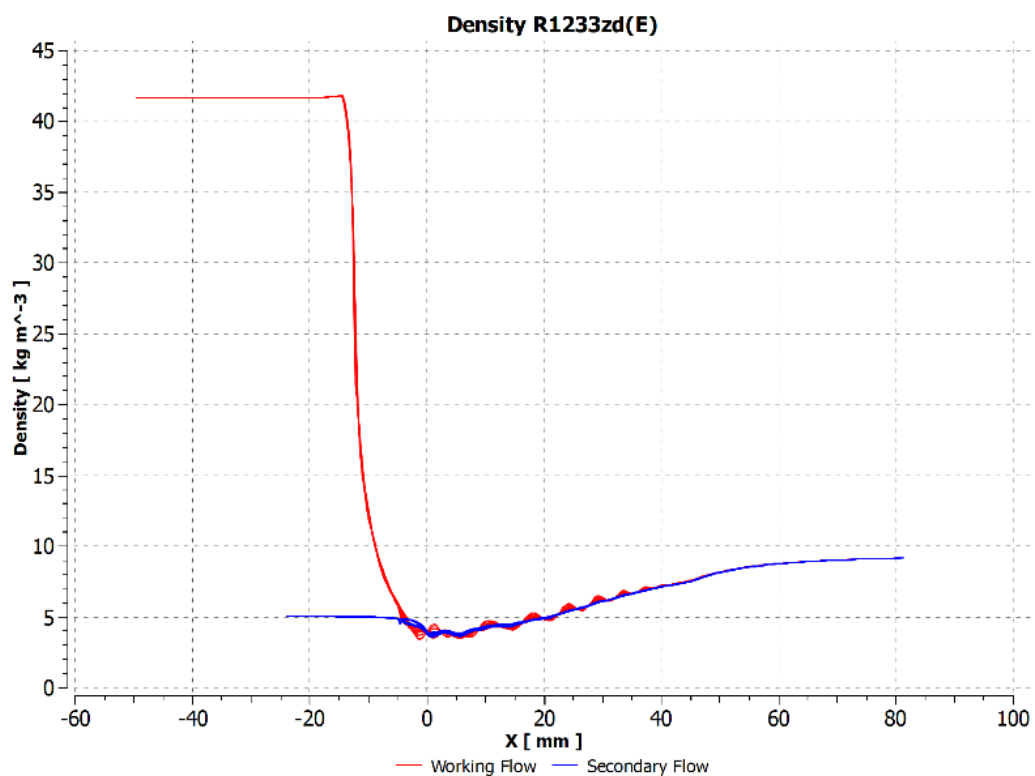


Figure 2.42 Density chart of R1233zd(E)

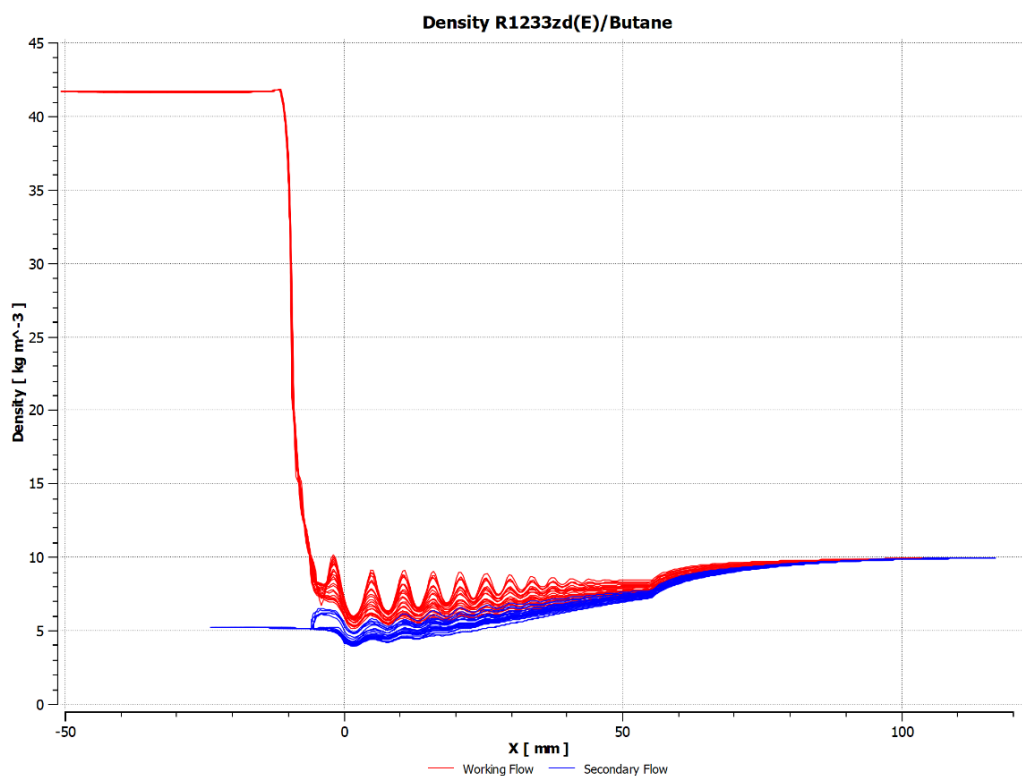


Figure 2.43 Density chart of R1233zd(E)/Butane

2.7 Off design conditions

Since the fluid separation process in the fractionating condenser is not ideal, and it is difficult to receive design mass fractions of fluids in evaporator and generator a CFD model for the various mass fractions in evaporator and vapor generator was provided. APPENDIX C represents simulation parameters and entrainment ratio for R1233zd(E)/Butane binary fluid at various mass fractions and operating parameters. Fig. 2.44 represents a depends of entrainment ratio from condensation pressure at constant $t_{\text{gen}}=90^{\circ}\text{C}$, $X_{\text{gen}}=1$ and various mass fractions of working fluid in evaporator. It is shown that increasing backpressure decreases the entrainment ratio significantly. Increasing a mass fraction of working fluid in evaporator does not allow to compensate negative effect. Increasing evaporation temperature by 3° at level $t_{\text{eva}}=18^{\circ}\text{C}$ increases entrainment ratio and allows to operates at the same capacity.

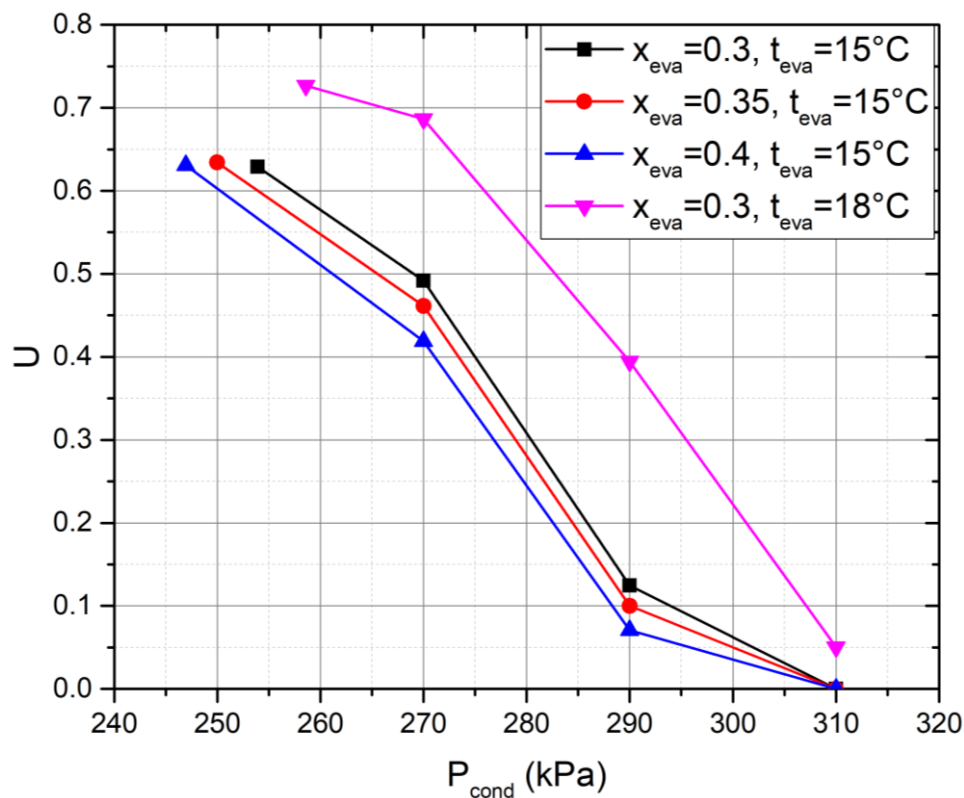


Figure 2.44 Dependens of Entrainment ratio from condensation pressure at R1233zd(E)/Butane (1/0), $t_{\text{gen}}=90^{\circ}\text{C}$ and various evaporation temperatures and mass fractions in evaporator.

Fig. 2.45 shows that adding premixtures of refrigerant fluid to vapor generator increases the efficiency of ejector by 15%. Also, it decreases critical back pressure

comparing to the pure working fluid in a generator. That allows stabilizing efficiency of ejector even at a significant increase of backpressure.

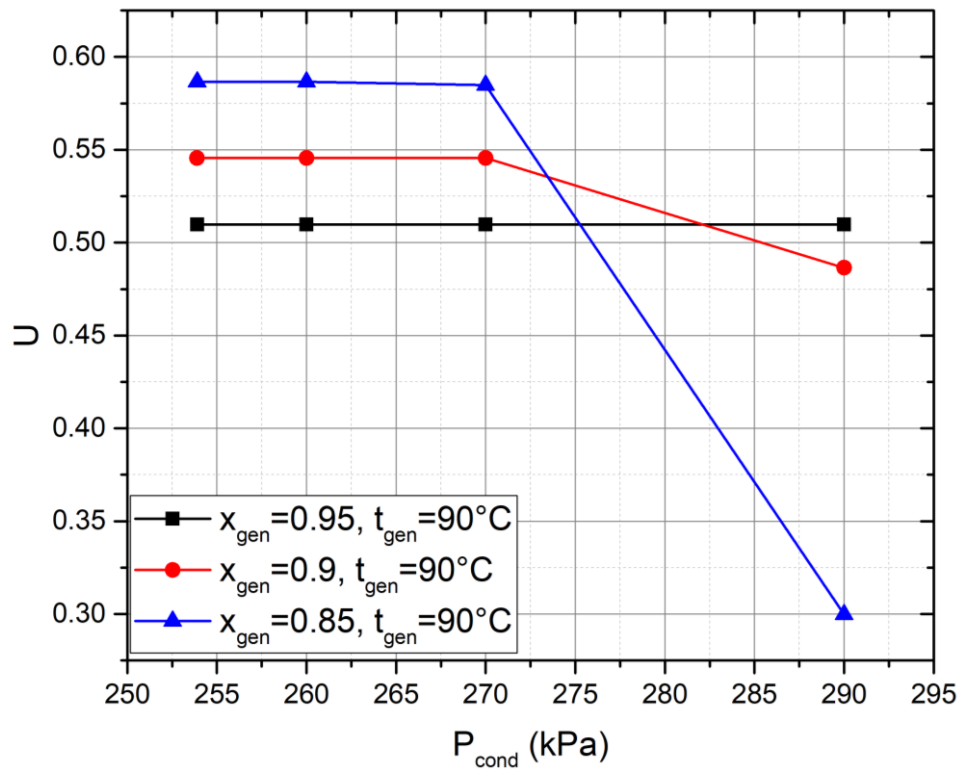


Figure 2.45 Dependence of Entrainment ratio from condensation pressure at various mass fractions in generator at constant temperature 90°C , and constant parameters in evaporator.

2.8 Results and discussions Chapter 2.

- 1 Defined main energy losses in ERS, which is caused by the need to expand active flow to lowest pressure and following compression of working and refrigerant flow to intermediate pressure due to the inelastic collision of active and passive flows.
- 2 It is shown that shock loss decreases entrainment ratio by 30-40%, and other losses by an additional 30%.
- 3 Obtained a series of curves for a number of fluids at specified parameters that shows a dependence of shock losses on the pressure in the suction chamber and entrainment ratio. Since an optimal suction pressure depends on other factors, absolute values obtained by basic calculation of energy loss is not necessary to use for ejector flow part design. However, quantitative data of the process can be one of the analysis methods for improving ejector efficiency.
- 4 Since the working flow is supersonic, and refrigerant flow is subsonic, varying critical velocities allows decreasing significantly shock losses, which is proportional to a square of flows rate difference. In order to reduce shock losses, it is necessary that the speed of sound of working flow be significantly lower than the velocity of refrigerant flow. In this case, flows will collide with a smaller velocity difference, which will reduce the loss of kinetic energy. This requirement determined a ratio of molecular masses of fluids. The working flow must have a high molecular weight, low speed of sound, and low latent heat of evaporation. Also, it should be taking into account, actual properties of mixture solubility of components, the formation of zeotropes, azeotropes or heteroazeotropes, as well as PVT data in a range of operating parameters, safety, and toxicity criteria
- 5 Described mathematical model of flows processes in a supersonic ejector takes into account vortexes formation, shock waves, allows optimizing the geometry of ejector flow part to eliminate any reverse or turbulence flows and minimize irreversible losses in the diffuser.
- 6 Analysis of ejector operating on binary fluid at off-designed conditions was provided along with an influence of premixture in evaporator and generator on ejector efficiency.

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CHAPTER 3

Chapter 3. Verification of calculation and CFD modeling results.

The demonstrated technology combines a traditional rotary dryer with a thermally driven vacuum ejector and heat pump system. The all-in-one indirect gas-fired drying system integrated with a thermally driven ejector system (TDES) offers a highly efficient and cost-effective alternative to the state-of-the-art technologies for the indirect drying and thermal processing of bulk solids, with the option of temperature profiling as well as waste heat and water recovery and reuse.

Controlled heat-input to the product is provided while a vacuum is pulled through the use of ejectors. The combination of heat and vacuum allows the product to be dried to specified remaining moisture content requirements in a shorter time. The system was designed by Wilson Engineering Technologies, Inc., a CA based engineering company with over 40-years of experience in ejector technologies and heat pumps, focusing on thermally driven ejector systems and heat pumps technology development and commercialization. Clayton Boiler, a major boiler manufacturer in CA, supplied their commercially available low-emission steam generation system and controls.

Product drying can be realized over a wide range of process temperatures and throughputs, providing reliable operation with enhanced product quality and improved energy efficiency. Employing commercially available off-the-shelf low NO_x combustion systems provides the opportunity to reduce combustion emissions in industrial and commercial drying operations.

3.1 Advanced Ejector Heat Pump Simulation and Design

3.1.1 System Specifications

The specifications for the industrial advanced drying system were provided by Wilson Engineering Technologies, Inc, as shown in Table 3.1, with input from the host site, Martin Feed, LLC. The capacity of the thermo-vacuum drying system (TVDS) is 10 tons of product per hour. It is designed for drying food wastes with an initial moisture content of 35 percent. The final product moisture required by Martin Feed is 10—12 percent. Product temperature during the drying should be less than or equal to 80°C to avoid any adverse effect on product nutrition quality. Product heating is provided by the latent heat released from the steam condensation at the specified temperature and pressure conditions as the product goes through the dryer.

Table 3.1 : Design Parameters for 10 Ton/Hour Drying Capacity (Credit: Wilson Engineering Technologies, Inc)

Specification	Value
Inlet moisture content	35%
Outlet moisture content	12%
Weight of product loaded in rotary dryer	166.67 kg per minute
Steam mass flow rate	0.726 kg per second
Steam temperature	177°C
Steam pressure	8Bar
Heat input	1984.78 kW
Heat recovery: moisture evaporation from the product	1095.52 kW
Water pump power consumption	2,5 kW
Rotary dryer drive power consumption	12 kW
2 Airlock valve drives power consumption	5kW

Project design involves the following steps:

1. Heat input and vacuum level parameters optimization, based on previous studies and available drying curves; drying time definition at various vacuum levels achieved in the rotary dryer
2. Preliminary design of the rotary dryer with a product moving shaft and heated by holo-flites
3. Preliminary design of the ejector-based vacuum system and heat pump
4. Pre-order of off-the-shelf parts and manufacture of the original parts for a TVDS under the field supervision of the project designer's team.
5. Control for installation, check-out and startup operations, and shakedown tests
6. Demonstration system design improvement and adjustments, if needed, according to the shakedown results
7. Monitoring, evaluation, and recommendation for system use; operation manual print-out

3.1.2 Process and Ejector Simulation

In the first phase of the project, Wilson Engineering Technologies, Inc designed the gas-fired thermo-vacuum system. Considering the data provided by the host site, the project team developed a heat and material balance simulation for process and instrumentation diagram (P&ID) development to account for all energy flows and product drying principles, based on the system specifications provided.

Based on the results of the simulation, a mathematical model of the vacuum

ejectors was developed that fed computational fluid dynamics (CFD) modeling of the vacuum ejector (Fig. 3.1). Wilson Engineering Technologies has a strong background and experience with fluidic ejector compression and developed the advanced heat pump design.

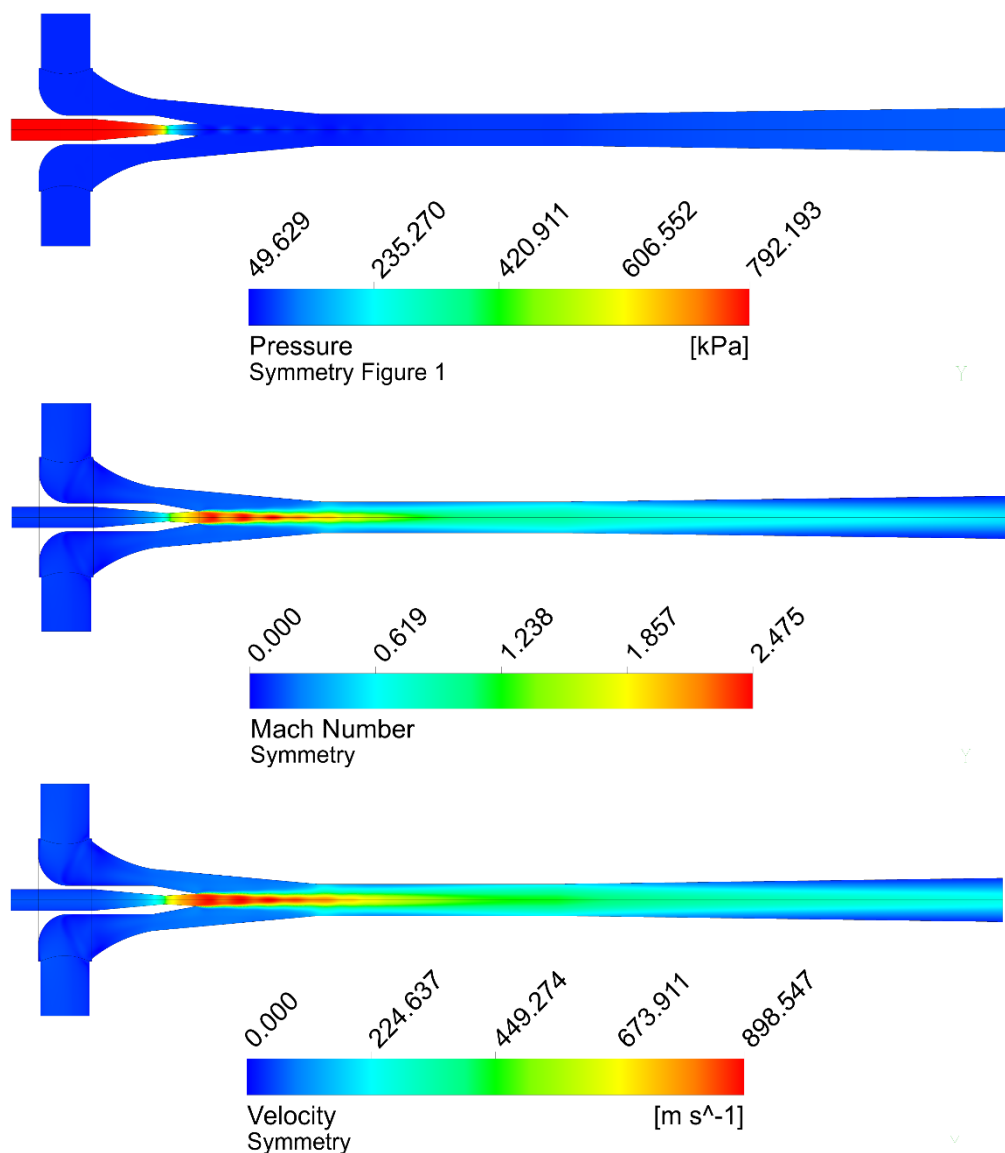


Figure 3. 1. Pressure, Much number and Velocity distribution in the Vacuum Ejector Pump (Wilson Engineering Technologies, Inc)

The thermally driven ejector system replaces the mechanical compressor used

in the traditional vapor-compression heat pump cycles and the electrical vacuum pump used in the traditional thermo-vacuum dryers. The replacement of an electrically driven compressor with a natural gas (thermally) driven ejector significantly saves on energy costs, thereby reducing greenhouse gas production.

3.1.3 Process Description

The P&ID for the advanced heat pump gas-fired thermo-vacuum drying system is shown in Appendix D. There are two closed steam/water loops: the boiler loop and the thermally driven ejector/dryer loop. The descriptions following reference the equipment numbers from the P&ID.

Boiler System

The first steam closed loop is associated with the boiler. The boiler provides the motive fluid for the ejectors and heat to the product via indirect heating. This boiler can operate at various modes of the full or partial load, which makes it possible to quickly adapt to specific drying conditions without losing the extra energy resource.

The reason this steam loop is kept separate from the primary system is to maintain the integrity of the boiler system by keeping any impurities evaporated from the product out of the boiler tubes. The boiler is supplied with soft water provided to meet the requirements of its safe and efficient operation, the stock of which is replenished from purchased containers.

The boiler (B-001) produces saturated steam at 177°C and nominal 896kPa. The steam is condensed in a shell and tube condenser-evaporator (HE-001) as it vaporizes the motive steam for the ejectors, which are on the tube side of HE-001. The condenser-evaporator is a vertical straight-tubes single-pass shell-and-

tube apparatus. Process water boils in the tube bundle, and steam from the boiler condenses in the shell side. The condensate from HE-001 goes to a counter-flow plate heat exchanger HE-002, which is used to preheat the motive fluid to bubble point temperature. It reduces irreversibility in the heat exchange process and ensures the operation of the condenser-evaporator without pressure and level fluctuations. The sub-cooled condensate is circled back to B-001, where it is again heated and evaporated.

Thermal Driven Ejector System

The second steam loop contains the thermally driven ejector system, steam tank, and the dryer. In this loop, there are certain impurities of salts, fats, and dissolved gases present. The condensate, before pumping to the heat exchanger HE-002, is preliminarily filtered from impurities, particles and, sometimes, fats that should be separated and returned to the product. Bubble point condensate enters the tube side of HE-001 and exits as saturated steam at 792.9kPa (175°C). The steam is directed to the nozzles of eight parallel-connected ejectors (EJ-001, -002, -003, -004, -005, -006, -007, -008). This steam represents the motive flow for the ejectors to entrain the evaporated moisture from the product and recompress it, thereby recovering its useful heat. All ejectors remove the steam-air mixture from the holo-flite®, which is released from the heated product. Thus, the ejector captures heat with a temperature of 70-80°C at a pressure of 45—70 kPa and converts it into heat at a temperature of 95—110°C and pressure of 100—115 kPa, i.e., works as a heat pump.

Additionally, a vacuum is pulled in the dryer, intensifying the product drying process and shortens the time of drying 2-5 times compared to the ambient

pressure drying. The steam enters the ejector at about 792.9 kPa (175°C). After passing through the expanding nozzle, the pressure is reduced to about 55-69kPa, entraining moisture from the product. The combined steam and moisture pass through the ejector, exiting at ambient pressure and with a few degrees of superheat (~110°C). Each ejector can be independently switched on to provide an additional drying capacity depending on the level of the initial product's moisture content.

The useful, but already low-grade steam exiting EJ-001,-002, -003, and -004 is directed for heating the product in the dryer (D-001), through the dryer jacket and hollow screws flights. This heat is transferred directly to the product as it comes in contact with the flites and inner jacket surfaces while the product is moved through the dryer. If additional ejectors EJ-005,-006, -007, and -008 are operating, their exiting steam flow is directed to the steam tank (ST-001). The steam condensing in the dryer jacket and hollow screws (holo-flites) heats up the product to increase the rate of evaporation, after the heat sink condenses so the condensate, exiting the dryer, flows to the steam tank ST-001 by gravitational forces.

All uncondensed steam and air that was ejected from the dryer volume are vented out through the stack. The follow-on engineering of the demonstrated technology should consider additional condenser to recover the vented steam and utilize it for other services relevant to the application or the site operation. This condensate could be combined with other excess water streams, processed, and utilized for irrigation, livestock, or other needs. Condensate is collected at the bottom of ST-001 and is pumped (WP-001) through a water filter (F-001) to be

first heated up to boiling point temperature at HE-002, and then evaporated in HE-001.

Product Dryer

The product is moved through the dryer by the holo-flights and changes its moisture content from high to low value. The wet product, delivered by auger from the remote hopper, enters the top of the dryer through an infeed rotary airlock valve (RO-001) that supplies feed into the vacuumed dryer with a negligible portion of air penetration. In the dryer volume, the product's temperature is raised to about 82°C due to the indirect contact with the heated holo-flites and jacket walls as it is continuously pushed through the dryer. Supplied heat and produced vacuum of 34.5-41.4 kPa) result in intensive evaporation of product moisture. The dryer should be operated fully loaded to ensure full contact of the product with the jacket and holo-flites' walls surface of the dryer, which is heated by the condensing steam. The hot and dry product exits the rotary dryer at the bottom side on the other end of the dryer via the outfeed rotary airlock valve. The moisture of the product is controlled through the speed of the holo-flites rotation at which it is moved through the dryer, and monitored by product humidity transmitters at the entrance and exit of the dryer.

3.1.4 Working Fluids and Operational Parameters

The primary working media in the current drying system are water and steam. When the process is just started, and vacuum is yet to be stabilized in the holo-flite drying chamber, the amount of air will prevail in the amount of steam. However, later, since air infiltration to the drying chamber is minimal, the steam-water component of the steam-air mixture increases. As the product moves along the holo-flite at nominal or lower initial moisture content, the added mass in the

ejectors' decreases. It automatically decreases the pressure inside the drying chamber. A further decrease in pressure stimulates more intensive removal of moisture, which can even lead to product overdrying. When such a situation arises, the steam supply to the ejectors' nozzle temporarily stops; all other parts will be stopped automatically in a short time, rated for 10—20 seconds. If the initial moisture content in the product exceeds the designed values, and the pressure in the drying volume does not reach the designed value of 45-60 kPa at the exit of the dryer, - operators can reduce the rotary speed of the holo-flite motor and evacuate maximum moisture from the product. In extreme cases, when this does not solve the problem, the product to be discharged from the dryer and delivered back to the feeding hopper.

Listed below are the results of the calculation of the entrainment ratio, the coefficient of performance (COP), the geometric characteristics of the flow part of the ejector with the working steam flow of 1kg /s, as well as the values of thermodynamic functions of the working, ejected, and mixed flows:

The operation of the steam-air ejector heat pump in a set temperature range is characterized by high-performance results: low pressures in the steam generator and condenser, explosion and fire safety, and non-toxicity and total environmental safety (ODP [Ozone Depletion Potential] = 0; GWP [Global Warming Potential] = 1). In this case, most of the water and steam circulate in a closed loop. Excessive water extracted from the products during the drying process can be accumulated and used for the necessary technological purposes. The boiler loop with soft water circulation is isolated from the loop of the condensate circulation, which significantly reduces the operating costs for water treatment and protects the

boiler from contamination. Incomplete heat recovery results in some losses, which are small due to high heat transfer coefficients in both the condensation process and the evaporation in the steam generator that is also determined by the properties of water as a highly efficient heat carrier.

Compared to hydrocarbon-based low-boiling substances, water has a higher viscosity, which simplifies the requirements for seals at joints and minimizes leakages. Also, due to the availability of water and its safety, small leaks are acceptable and quickly replenished from the accumulated reservoirs.

3.1.5 Integration Features

This thermal vacuum method of drying the product is based on the principle of operation of an ejector heat pump.

An ejector heat pump consumes high potential heat from a gas boiler, takes heat from a source with a low temperature (the steam-air mixture from the product being dried), and releases heat of the intermediate potential in an amount equal to the sum of the heat removed from the boiler and the product. Since the demo thermo-vacuum drying system is designed for heat flows entering ejectors to be equal ($COP=1$), then at the ejectors' output, the amount of heat is doubled compared to the heat generated by the boiler. Half of the output goes to heat the product, and the other half remains unused in the current stage of the technology development that leaves additional opportunities for further development and optimization.

The drying process is a non-stationary process. It is explained by the fluctuation in moisture content as the product moves along the holo-flite drying chamber. Local areas of steam-air compartment above the product also have different

moisture content, which may contribute to the migration of airflow along with the product. In the continuous process of product infeed, it is not possible to reduce the pressure above it; - space is fixed. It is not possible to divide this space into individually sealed compartments due to the continuous product inside the dryer chamber. If the drying is carried out discretely or the drying volume is designed as several consecutive chambers, isolated from each other, then a deeper vacuum drying can be achieved in each of them. At the same time, the pressure can vary up to 0.1 bar only. Unlike operation in the off-design conditions, when a decrease in the entrainment ratio is a consequence of a change in operating parameters, in this case, a decrease in the flow rate of the ejected flow may cause a decrease in the suction pressure, while the ejector continues to operate at the limiting mode. Thus, the modeling of the ejector for the conditions of maximum initial moisture content of the product allows, without changing of the geometry of the flow part, to work in optimal conditions at the limit mode at any values of the initial moisture content lower than the maximal. It means that in the ejector with any parameters, there will be no locking of the cross-section of the mixing chamber, reverse flows, i.e., unproductive losses. Fig. 3.2 represents the dependence of evaporation temperature, evaporation pressure, and ejector outlet temperature on entrainment ratio.

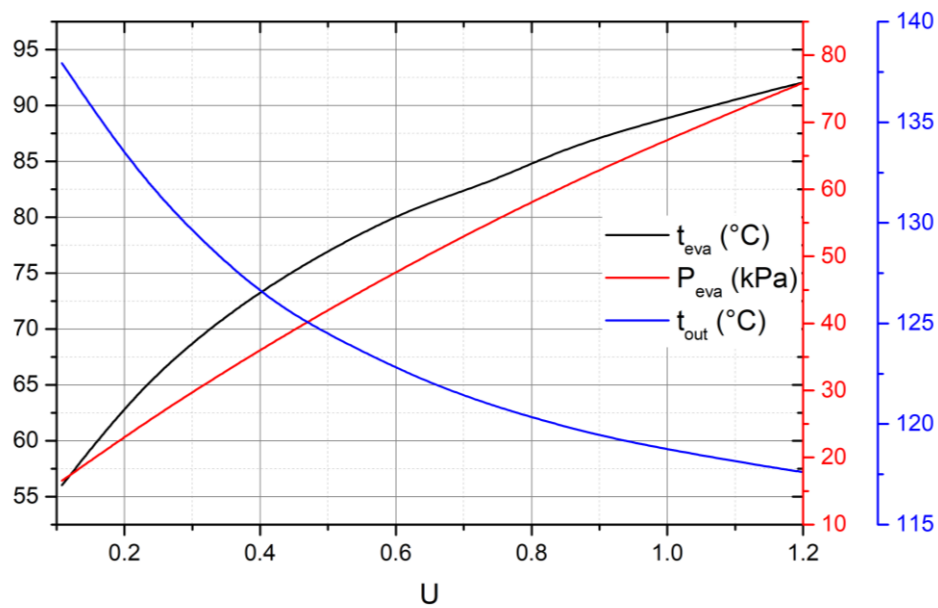


Figure 3.2 Evaporation Temperature, Pressure, and Ejector Outlet Temperature vs Entrainment Ratio (Wilson Engineering Technologies, Inc)

The final installation of thermal vacuum drying is the layout of the following main components:

- Clayton steam boiler with a capacity of 200 BHP of heat, which produces high pressure steam and temperature to provide heat generation of the working steam.
- Heat exchange unit used to generate steam, consisting of counter flow plate condensate pre-heater and a vertical shell-and-tube steam generator-condenser developed by Wilson Engineering Technologies, Inc.
- Holo-flite® with 2 horizontal screws manufacturer by the “Denver HollowFlite,” modified to the objectives of this project.
- A set of 8 Wilson Engineering Technologies, Inc designed ejectors connected in parallel into groups of 1, 2, and 4
- The product infeed unit, consisting of a hopper, inclined auger and electric motor

- The product outfeed unit, consisting of an inclined auger, feeding the product into the dried product storage area.
- Set of 2 rotary air-lock valves in between both augers and holo-flite.
- Steam-water tank, from which condensate is pumped and filtered before entering to the heat exchange unit of the steam generator

A free-standing panel controls the operation of the drying unit. It is mounted according to the schematic diagram developed by Wilson Engineering Technologies, Inc and assembled by Spurt. The operation of the boiler is controlled independently.

3.1.6 Performance Evaluation

The demonstration system installation scheme allows varying the performance in a reasonably wide range by changing the speed of the auger supplying product through a change in the frequency in the range of 75 to 10 Hertz (Hz).

The thermal load was regulated by several ejectors switched on and the adjustable pressure on the suction line of the ejectors. The boiler regulates its performance by the amount of heat consumed in the production of steam in the shell and tube heat exchanger.

In cases where the initial humidity of the product has a maximum design value of 35 percent, the product feed rate is minimal and corresponds to the minimum engine speed, i.e., minimum current frequency. The maximum performance of the product is 366 lb per minute. In this case, the final moisture content of the product is 12 percent. Accordingly, with a lower initial moisture content of the product, the product performance can be increased in conformity with the amount of moisture that must be removed from the product to fixed final moisture content.

Table 3.2 presents the values of the maximum mass productivity of the installation and the speed of rotation of the holo-flite, depending on the initial humidity of the product.

It should be noted that the GFTD installation designed and demonstrated during this project is not intended for drying of over-wetted product, as there is an increased adhesion of the product (consisting mainly of carbohydrates) to the surface of the holo-flite®, producing an over-dried crust which is required to be removed for optimal operation. Therefore, during the pre-commercial engineering and thermo-vacuum technology implementation, the product should be either pre-dried (or pressed) to the level of at least 50 percent moisture content, or the heating surfaces should be coated with non-sticking material to prevent undesirable depositions.

Table 3.2 Mass Productivity of the Dryer at Various Initial Moisture Levels of the Product (Credit: Wilson Engineering Technologies, Inc)

	Initial moisture, %	Product flow, kg/min	Frequency, Hz
1	35	166	22
2	32	192.3	26
3	30	211.8	31
4	27	254	38
5	25	293.5	42
6	20	476.7	46
7	17	762.95	55

3.2 System Installation.

Upon completion of the series of pre-shipment tests, evaluations, and inspections, all the components of the demonstration system were delivered to the participating host site for final assembly and installation. In addition to typical

inspections, the rotary heater components were pressure tested before shipment to the site. The team and local mechanical installation contractors performed extensive field engineering. The host site provided the space for the unit; however, it was not able to provide the required utilities, namely power, water, and gas. No construction permit was required, as a skid-mounted approach was elected for the demonstration unit. Fig. 3.3-3.4 shows installation and system assembly.



Figure 3.3: System Mechanical Installation at Martin Feed, LLC in Corona, California (GTI)



Figure 3.4:Overall View of Thermo-vacuum Drying System Installed.

3.2.1 Utilities

The host site did not have hookups for power, water, gas, and compressed air. A generator was procured for startup, shakedown, and demonstration periods, and natural gas was supplied by a mobile natural gas supplier, Ultimate CNG. Soft water totes were procured for charging the system for a startup. A mobile air compressor was rented to supply air for instrumentation.

3.2.2 Steam Generator

Clayton Industries was selected to provide the steam generator for the system. This steam generator provides steam for indirect heating of the motive fluid to drive the ejectors.

3.2.3 Airlocks

Prater's rotary airlocks were used for feed handling to charge and discharge the product into the system that is under vacuum. The airlock at the product inlet side feeds the product into the vacuum in a metered manner while maintaining the pressure differential through an airlock seal, thereby preventing loss of vacuum

and temperature in the system. The same airlock operation was established at the product outlet. APPENDIX D includes photographs of the airlock used in the demonstration unit.

3.2.4 Rotary Holo-flite®

Metso's rotary holo-flite® was specified for the continuous integrated heating and transportation of the product. Fig. 3.5 shows an overall view of the holo-flite® configuration by which product can be moved through a trough. Multiple shafts can be integrated for larger feed rates. It is comprised of a jacketed cylinder containing screw conveyors. The rotary transport unit selected for the project has two screws, which enables transport of the required 100 tons per day of product requested by the host site.

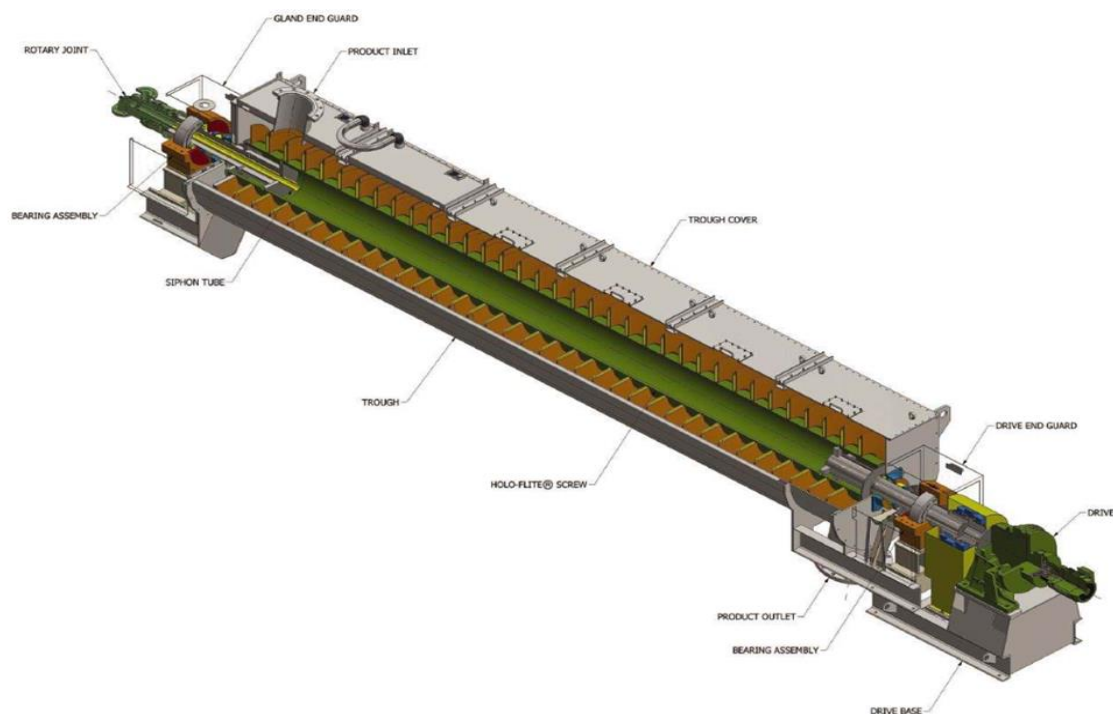


Figure 3. 5:Generic Holo-flite® Illustration (Metso)

This is an established thermal drying technology that has been used in various industries for over 60 years, including food processing, petrochemical processing,

mining applications, and waste applications. The benefit of this technology is that it continuously conveys product through a trough via rotating screws while the product is indirectly heated as it comes into contact with the hollow flights and shaft. It is an indirect thermal heating/drying system. The inside of the shaft of the screw is hollow, allowing for the flow of thermal fluid, as seen in Figure 3.6. For this application, steam is used as the thermal fluid. The trough and screws are constructed of stainless steel.

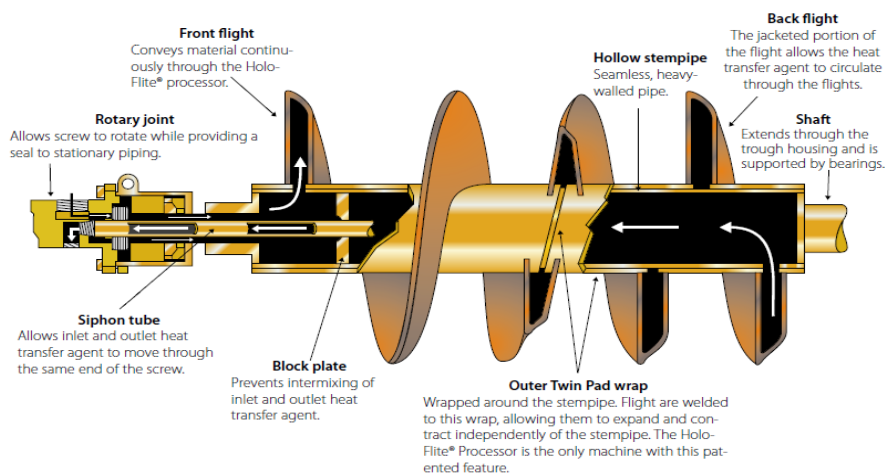


Figure 3.6: Rotary Holo-flite® (Metso, manufacturer)

3.2.5 Ejectors

As described previously, the ejectors create a dynamic vacuum in the dryer chamber and act as a fluidic compressor for the advanced heat pump. The specifications were provided for the motive and ejected flow-operating regimes. Fig. 3.7 shows the vacuum ejector assembly and Fig. 3.8 shows the assembly of the ejector-based system.

3.2.6 Measurement Sensors and Control Panel

The combustion controls were integrated into the packaged boiler unit supplied by Clayton Industries. In the commercial implementation of the system, the boiler

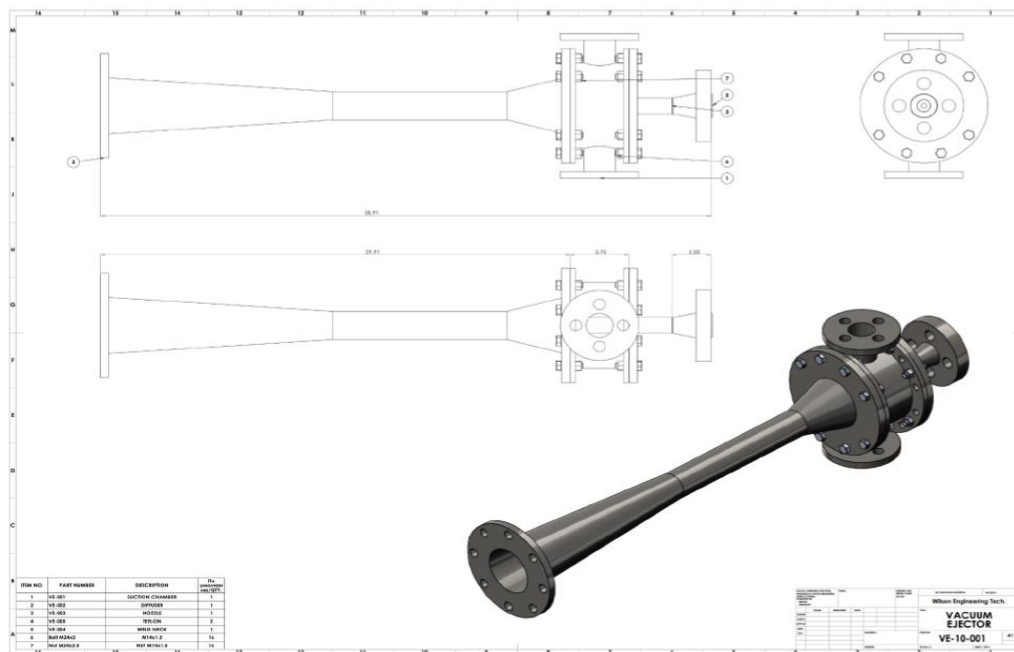


Figure 3.7: Vacuum Ejector Assembly (Wilson Engineering Tech., Inc)



Figure 3. 8: Assembly of Ejector-Based System (GTI)

controls will be fully integrated with the steam controls of the ejector system.

The dryer controls for the demonstration effort were developed by Spurt Electric, Inc. The controls are used for GFTD motor control of the various motor components and ejector solenoid valves to maintain the desired production rate through the dryer, condensate recirculation, vacuum level, and heat level in the dryer. These are described in more detail in APPENDIX E.

The overview screen is shown in Fig. 3.9 presents an overall picture of the GFTD, including temperatures, pressures, and rotation speed of the inlet/outlet airlocks and holo-flite® screws.

The solenoid valves control screen (Fig. 3.10) allows the operator to open and close valves controlling the level of vacuum in the dryer and to direct steam to the dryer jacket and flites to apply heat to the product.



Figure 3.9: Control System Overview Screen (Left: before ejectors start; right: at ejectors operation)

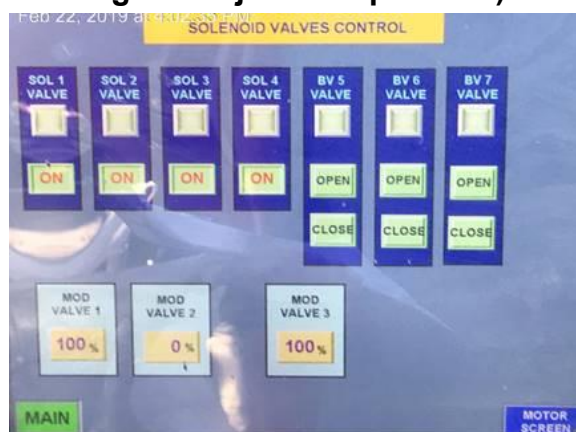


Figure 3.10: Solenoid Valves Control Screen (Credit: GTI)

The motor control screen allows the operator to adjust the speed of the airlocks at the inlet and outlet of the dryer and to change the holo-flite® speed, which is a variable that can affect the level of drying of the product during the test run.

The full control system was not in place for the demonstration system. The

completed system will have feedback from the humidity sensors at the inlet and outlet of the dryer to control the speed of the system and the vacuum pulled, and to automatically adjust to meet product moisture targets.

3.3 Testing Results

The main achievement of the system tests is a demonstration of the high level and quality of the calculation, design, and manufacturing of the ejectors, which consistently produced the calculated parameters and automatically shifted to the designed limit load corresponding to the lower suction pressure. The results of the runs have successfully proved that the demonstrated technology can evacuate the moisture from the product with simultaneous product heating and heat pumping effect for efficient drying of the product (Table 3.3).

Table 3.3 Experimental results of thermo-vacuum system testing with 6 ejectors operation

№	Tested Parameter	Test 1	Test 2
		Vacuum 54 kPa	Vacuum 44kPa
1	Product flow rate, kg/min	166	190.9
2	Initial moisture content, %	22.39	18.61
3	Drying time, min	8	8
4	Weight of removed moisture, kg	173.76	129.23
5	Remaining moisture content, %	9.3	10.15
6	Entrainment ratio, kg/kg	0.714	0.531
7	Heat factor	0.678	0.502
8	Motive steam flow rate for 1 ejector, kg/s	0.08452	0.08452
9	Total motive steam flow rate, kg/s	0.507	0.507
10	Total heat input, kW	1223	1223
11	Gas flow rate, kg/min	1.623	1.623
12	Volumetric gas flow rate, m ³ /min	2.325	2.325
13	Combustion heat, kW	1609.5	1609.5
14	Boiler efficiency, %	76.2	76.2

3.3.1 Fuel Efficiency and Emissions

A Clayton, Model EG204-FMB, boiler equipped with a low NO_x burner was used to generate steam during the drying process. The boiler is rated at 81.5 percent efficient. Emissions testing was performed on the boiler to measure emissions of NO_x, CO, and oxygen (O₂) and to demonstrate compliance with the requirements of SCAQMD Permit to Operate and Rule 1146. The average measured CO concentrations were below the quantifiable range of the reference method during each test.

Testing was conducted while the boiler was operated at high, mid, and low firing rate conditions. Results are summarized in Table 3.4. These measurements were taken during the initial startup of the unit and were not repeated during performance testing.

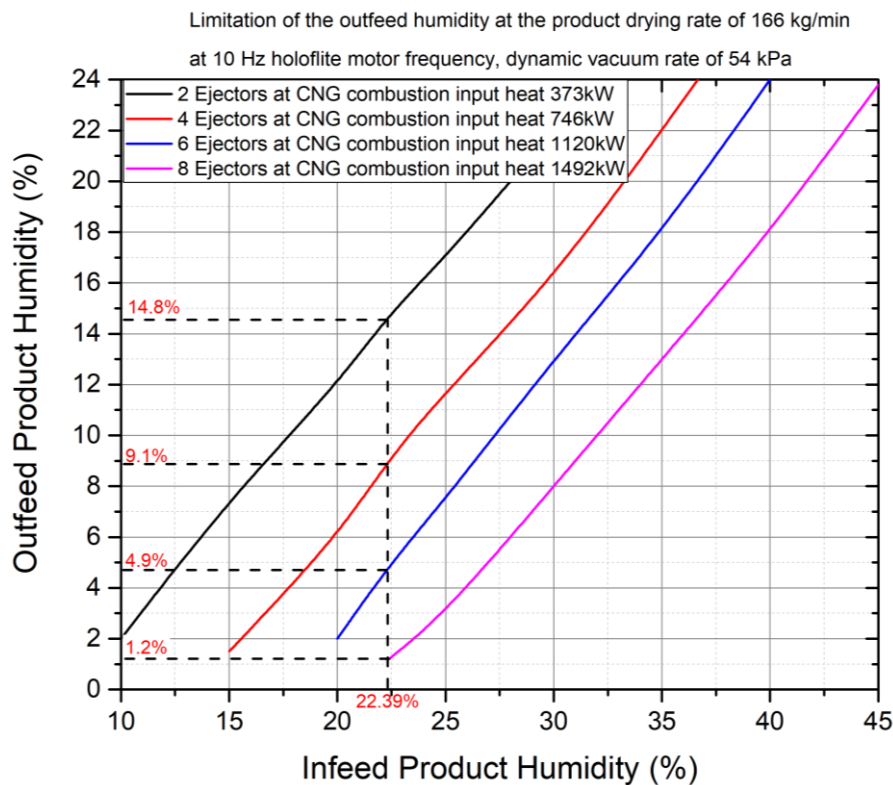


Figure 3.11 Combustion heat input vs remaining moisture content in the product after GFTVD (Wilson Engineering Technologies, Inc)

Table 3.4 Boiler Emission Summary (Tetra Tech Inc)

Parameter	Units	100 Percent Load	50 Percent Load	25 Percent Load
O ₂	%	11.30	11.24	11.76
CO ₂	%	5.58	5.54	5.24
NO _x	ppm@3%O ₂	7.53	7.26	6.91
	kg/hr	0.0276	0.0149	0.0068
CO	ppm@3%O ₂	<18.6	<18.5	<19.6
	kg/hr	<0.0417	<0.0231	<0.0118

Boiler emissions are compliant with SCAQMD emissions requirements.

3.3.2 Energy Use Summary

The energy use for the thermo-vacuum system was calculated using direct measurement data and operational data, summarized in Table 3.5.

Table 3.5 : Energy Use Summary (Tetra Tech Inc)

Source	kW
Boiler	1223
Steam tank loss	584.9
Condensate return line	33.2
Portable generator	60 kVA

Table 3.6: Moisture Analysis

Sampling	Sample Location	Sample Time	Average Moisture, (%)
1	In	11:09	22.39
See Figure 28	Out*	-	14.8 (a) 9.1 (b) 4.9 (c) 1.2 (d)
2	In	14:51	18.61
See Figure 28	Out*	-	10.4 (a) 4.6 (b) 0.5 (c) n/a (d)

*Sample moisture at the system outlet has been calculated based on the measured vacuum level, heating input and number of operated ejectors (a – 2, b – 4, c – 6, d - 8). Actual measurements of outlet moisture were negatively affected by adverse weather conditions during the test and excluded from reasonable consideration. Credit: GTI

3.3.3 Moisture

Moisture analysis was performed onsite using Method ASTM D2216 – 10. The drying time used in the analysis was set at 43°C to avoid burning the sample during the moisture analysis process. Results from the analysis are summarized in Table 3.6.

3.4. Results

The overall aim of this project was to design and demonstrate a high-productivity integrated gas-fired drying technology of superior energy efficiency and benefits, including reduced gas consumption and an accelerated drying process. This system has demonstrated a promising performance at the laboratory scale. Additionally, the main achievement of the demonstration system was the design and manufacturing of the ejectors that are key components of the technology; during the performance testing these ejectors produced the calculated parameters and automatically shifted to the designed limit load corresponding to the lower suction pressure. The pressure measurements demonstrated the ability of the designed system to evacuate the moisture from the drying volume with simultaneous product heating and heat pumping effect for an efficient drying process.

Effectiveness and efficiency of the drying process are characterized by the drying time, energy consumption, and capital and operating costs, as well as by the product quality and environmental compliance. The thermo-vacuum process significantly improves the operation's drying time and energy consumption, and

provides favorable environmental impact to the community.

The project demonstrated the designed performance of the ejector system for product throughput of 166.6 kg per minute (~10 ton per hour). The ejectors evacuated about 38.96 kg per minute of air-moisture, where the air mass portion was under 1.5 percent. However, taking into account the minor leakages in the sealed chambers, the nominal moisture evacuation rate by ejectors should be 21.54 kg per minute to provide the dried product moisture content at the designed level of 12—15 percent.

The parametric optimization of the drying process by considering the product type, throughput variations, and vacuum dynamics are the subject of follow-on efforts. In order to dry product from 35 percent to 12 percent moisture content, there is a need to remove 38.1kg of moisture per minute. For that purpose, it is necessary to heat the product by providing 1583 kW. The removal of the evaporated moisture would require additional heat for blowing 70.8-99.1 m³/h of air at a temperature of 100—130°C in the amount of 2930-4400 kW. Therefore, the basic estimate indicates a required natural gas consumption of 529.16-633 m³/h.

The technology demonstrated under this project requires only 1963.5kW (198.2-226.53 m³/h) of heat for optimal ejector network operation. Due to the heat pumping arrangement of energy transformation, such a thermal input is sufficient to generate and sustain a dynamic vacuum at the designed level, as well as for heating the drying product to the designed temperature. Therefore, the thermo-vacuum system has a strong potential to reduce gas consumption by 61-65 percent for the same drying product throughput.

As to primary energy consumption, the demonstrated thermo-vacuum system

differs from the state-of-the-art equipment by mostly pumping power that was 8-15 kW, while the state-of-the-art drying equipment requires 5-6 kW recirculating pumps and over 20kW to power the air fans. Thus, the thermo-vacuum system demonstrated an obvious reduction in primary energy consumption by at least 40 percent.

Table 3.7 Comparative summary

	State-of-the-Art	GFTVD
Operating pressure, kPa	101.325	44.8-101.325
Operating temperature, °C	100-130	65-82
Drying rate (time), min	8-12*	4-8**
Natural gas consumption, kW	5275-6450	1465.3-2051.5
Primary energy consumption, kW	25-26	8-15
Conversion Efficiency, %	60	75-80***

Credit: Wilson Engineering Technologies, Inc

*depending on infeed product's moisture content and flow rate of the hot air blown through the product at the given temperature (80,000m³/h ~ 12 min, 120,000m³/h ~8 min);

**depending on infeed product's moisture content and rotation frequency of the holo-flite motor (10-75Hz or 9-35rpm)

CHAPTER 4

Chapter 4. Exergy analysis of BERS.

4.1 Introduction

Comparison of the energy characteristics always shows the advantages of the cycles that consumes electric power. That is not relevant because the value of electric power and low-grade heat are incommensurable. During energy performance evaluation, the power consumption cycles take into account production and transmission of electricity from the low-grade heat that is consumed by the heat utilizing cold generators. In this case, the selection of cold generator is determined by operational and cost characteristics [1-3]

Using the concept of exergy makes it possible to define the influence of imbalance of thermodynamic processes on energy conversion efficiency, i.e., allows to take into account second law of thermodynamics and isolate part of the energy that cannot be used due to the gas dynamic phenomena, friction, and heat transfer. This approach makes it possible to analyze the degree of thermodynamic excellence of the system's components and does not require a preliminary evaluation of the entire system. Therefore, exergy analysis becomes more relevant and useful [4-7].

An ideal cycle that does not take into account losses and exergy dissipation, exergy COP always equals to 1 and does not depend on the cold production cycle, i.e., it is the same for absorption and ejector cycles. The differences appear during the evaluation of internal and external irreversibility [8]. First is related to exergy dissipation inside a cycle. Second is related to exergy loss by energy exchange with external sources, i.e., final temperature difference during heat exchange, for

example.

Also, type, thermodynamic properties, and phase of the working fluid affect a real cycle efficiency. For example, comparing ideal Carnot cycle in two phase area with Rankine cycle taking into account that expansion work of liquid or consumed work for liquid compression is defined by eq 4.1.

$$E_{liq} = \int VdP \quad (4.1)$$

, is lower than the expansion or compression work of vapor Eq. 5.2.

$$E_{vap} = \int PdV \quad (4.2)$$

As a result, replacing the compressor with a pump or expander with a throttling valve in direct and reverse Rankine cycle gains an efficiency. In the first case, the required compression work decreases significantly. In the second case, loss of expansion work in the isentropic expansion is replaced by adiabatic throttling is more than compensated by simplicity and price of throttling valve comparing to the expander. There is a compromise solution to use an ejector as an expansion unit. That return to cycle a part of expansion work.

Ejector Refrigeration cycle combines a direct and reverse Rankine cycles, that means they use advantages of both cycles.

Energy analysis of ERS becomes prevalent in the last years. Articles provide an analysis of the influence of dissipation and exergy loss in various components of convenient ERS, including mechanical pump [6, 7, 9-12]. At the same time, it does not take into account a cavitation and leakage loss of liquids, which are most of the refrigerants based on halogen-substituted hydrocarbons of high to low pressures. Taking into account that exergy destruction, the pump is one of

the significant losses in ERS.

4.2 Exergy Analysis of the Binary ERS.

The exergy efficiency of a closed thermodynamic system is defined by eq 4.3.

$$\eta_e = \frac{E_{total}}{E_{Input}} = 1 - \frac{\sum \Delta E}{E_{Input}} \quad (4.3)$$

Since consumed exergy should create exergy of the product obtained, i.e., cold and also cover all the losses and exergy destruction that occurs during the actual processes in the cycle and interaction with external sources.

Exergy of cold is useful exergy produced in ERS. Consumed exergy is the energy of heat. It is known that exergy of cold is a minimal work required for its production at a condition that all processes in the refrigeration cycle are reversible, i.e., corresponding to Carnot Cycle.

Exergy of heat is maximum work produced from consumed heat, assuming that processes in Carnot power cycle are reversible.

$$E_{total} = G_{eva} \left(\frac{T_{eva}}{T_{amb}} - 1 \right) \quad (4.4)$$

$$E_{Input} = G_{gen} \left(1 - \frac{T_{amb}}{T_{gen}} \right) \quad (4.5)$$

Comparative analysis of exergy efficiencies of single and binary fluid ERS at various fluids and operating parameters represents an advantage of binary fluid ERS.

Work on improvement of heat utilizing systems at the beginning of the 20th century led to the appearance of several disparate concepts, such as "performance" and "usability of work production". In 1955 Yugoslav scientist Z.

Rant proposed the term "exergy," that was quickly adopted in European literature. Ya. Shargut in 1956 developed a theory of "Zero State" (zero-energy state), i.e., equilibrium state of the system with the environment. This theory allows defining a dependence between exergy and traditional analysis of physical systems. Based on this theory, it is possible to give a general algorithm for exergy evaluation of various systems.

Exergy definition allows evaluating the nonequivalence of different types of energy. So, despite a large amount of heat in the environment, its technical suitability is zero and it is necessary to spend some energy for its application (for example using a heat pump). In this regard, methods of COP evaluation of combined production of heat and other types of energy based on a simple addition of the energy of heterogeneous fluxes (for example, heat flux and electricity at CHP, cold at different temperature levels or produced at the expense of energy sources of different value).

Exergy analysis of technical systems allows in some cases to make more accurate conclusions of systems degree of perfection comparing to efficiency evaluation based on energy balance. For example, COP of the boiler defined on energy balance is high and reaches 90%. Taking into account exergy loss at irreversible heat exchange shows that efficiency is around 50%.

It is interesting to evaluate exergy characteristics of cold generators, that provides complete information of thermodynamic perfection of cycles, as well as analysis of exergy losses and methods of loss reduction.

Exergy analysis can be made in 2 steps. First, define the value of exergy COP of a cycle and provide a comparison with other cycles in order to provide a selection

of the most suitable cold generator. Second, provide an analysis of exergy destruction in every systems component. That allows to define critical parts and provide measures for their improvement. Successful implementation of tasks to reduce exergy losses for each cycle allows to return to the first stage and clarify the new ratio of efficiency of the cycle.

Energy destruction and loss in ERS units are defined by Guye-Stodola approach. For BERS exergy destruction in the fractionating condenser is required to be defined. Also, in the fractionating condenser, the water is heated by 20-30°C higher than ambient temperature. The exergy of heat provides additional useful exergy produced in a cycle, except for cold production. Fractionating condenser operates as a heat pump. The variable temperature in the fractionating condenser contributes to the reversibility process.

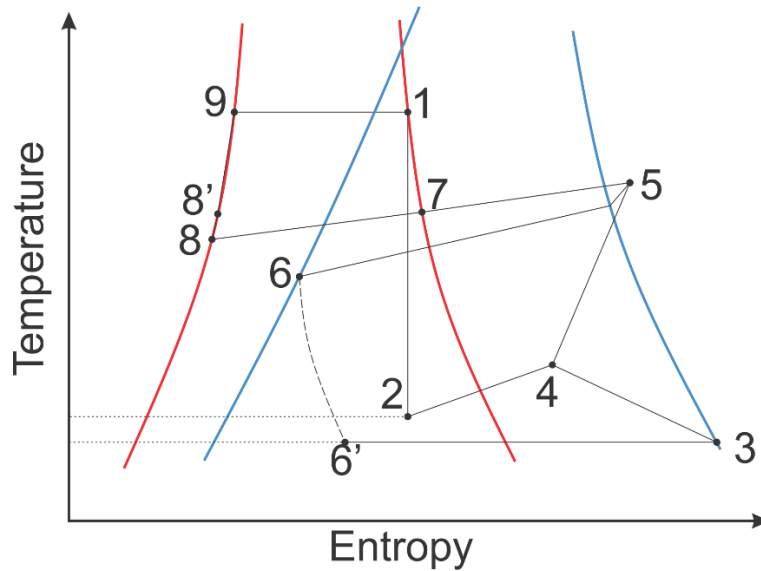


Figure 4.1 T-S diagram of processes in BERS.

Exergy destruction in BERS is defined by following equations:

Ejector

$$D_{ej} = h_1 + Uh_3 - (1+U)h_{amb} - T_{amb} \left((1+U)s_5 - s_1 - Us_3 \right) \quad (4.6)$$

Condenser

$$D_{cond} = U(h_5 - h_6 - T_{amb}(s_5 - s_6)) \quad (4.7)$$

Generator

$$D_{gen} = T_{amb}(s_1 - s_{8'}) + T_{gen} / T_{amb}(h_1 - h_{8'}) \quad (4.8)$$

Evaporator

$$D_{eva} = T_{amb}(s_3 - s_{6'})U + T_{amb} / T_{eva}(h_3 - h_6)U \quad (4.9)$$

Fractionating Condenser

$$D_{Fr} = Uh_6 + h_7 - (1+U)h_5 - (UT_{Amb}(s_5 - s_6) + T_{Amb}(s_5 - s_8)) \quad (4.10)$$

Thermopump

$$D_{th} = G_{vap}((h_1 - h_{8'}) - T_{8'} / T_{amb}(s_1 - s_{8'})) \quad (4.11)$$

In binary fluid ejector, additional irreversibility appears during the mixing process when mixing heat is generated. In this case, mixing heat from one side increases the temperature of the compressed mixture by increasing the temperature of heated water. From the other side, it decreases entrainment ratio by decreasing the specific density of the binary fluid. The influence of this factor on exergy balance is small and requires additional research. One solution to this issue is a heat removal by water jacket on the ejector diffuser part.

4.3 Energy Comparison of VCRS and Single/Binary BERS.

It is believed that the vapor compression refrigeration system (VCRS) has high exergetic COP. It is true if assumed that the degree of thermodynamic perfection is calculated as the ratio of the actual COP divided by COP of Carnot cycle [9,13,14]. In this case, the difference between the VCRS and JTT are significant

in favor of VCRS. For example, VCRS working on R142b with the evaporation temperature of $t_{eva}=12^{\circ}\text{C}$ and condensation temperature of $t_{cond}=35^{\circ}\text{C}$ has exergetic COP equal to 0.59 while ERS that operates at the same temperatures has exergetic COP equal to 0.226, which is 2.5 times lower. However, this type of comparison is incorrect because VCRS cycle work is generated from the cycle with much higher temperature level. With proper and even comparison, the gap between COPs will be much shorter. Furthermore, JTT can take the heat of condensation from distillate vapor, which is obtained from the saline or wasted water. This can significantly improve the heat source exergy, which can be represented by solar thermal collectors.

In order to determine exergetic COP of VCRS let's assume that the work on cold production for VCRS is obtained from the heat of the same temperature level as in ERS vapor generator. In this case, the difference of exergy COP between VCRS and ERS becomes shorter. Actual COP of VCRS is determined as the ratio of the effective COP to actual Rankine cycle thermal COP in this temperature range. In this case, the loss of exergy in the heat exchangers and other elements are not included (Eq. 4.12):

$$COP_{VCRS} = \eta_{exp} \varepsilon_{CRS} = \frac{N_{exp}}{Q_{gen}} \frac{Q_{eva}}{N_{comp}} \quad (4.12)$$

Actual COP of single and binary fluid ERS is determined as the product of entrainment ratio, calculated using in-house code, and the ratio of the specific cooling capacity to the heat consumption of the vapor generator (Eq. 4.13):

$$COP_{ERS} = \frac{Q_{eva}}{Q_{gen}} = U \frac{q_{eva}}{q_{gen}} \quad (4.13)$$

The program algorithm is described in [15, 16]. There is a Carnot cycle thermal

coefficient in the denominator of the COP equation for every cooling system. It is the product of the thermal COP of Carnot cycle on the Carnot cycle refrigeration COP (Eq. 4.14):

$$COP_{Carnot} = \eta_{Carnot} \varepsilon_{Carnot} = \left(1 - \frac{T_{cond}}{T_{gen}}\right) \left(\frac{T_{eva}}{T_{cond} - T_{eva}}\right) \quad (4.14)$$

Fig. 4.2 shows the dependence of exergetic COP from the generation temperature for different binary mixtures with the constant condensation and evaporation temperatures of $t_{cond}=35^{\circ}\text{C}$ and $t_{eva}=12^{\circ}\text{C}$. As can be seen from the graph, the mixtures exergetic COP decreases with the generation temperature increase. This is due to the fact of the predominance of Carnot cycle COP growth corresponding to the real COP growth. VCRS show overall high exergetic COP which, in some cases, lower than binary fluid ERS exergetic COP.

Fig. 4.3 shows the dependence of exergetic COP for heat-driven VCRS and single/binary fluid ERS from evaporation temperature for two-generation temperatures of 85°C and 90°C . For heat-using VCRS this figure shows constant growing line with the 10% increase of COP, and for the JTT there is more or less delineated maximum near the evaporation temperature of $-5^{\circ}\text{C} - 0^{\circ}\text{C}$. The differences in exergetic COP of VCRS and single fluid ERS with high evaporation temperature is about 60-70%, and in case of binary fluid ERS, for some evaporation temperatures, its exergetic COP exceeds VCRS COP in about 6%. High evaporation temperature regime shows an increased exergetic COP that favors VCRS systems.

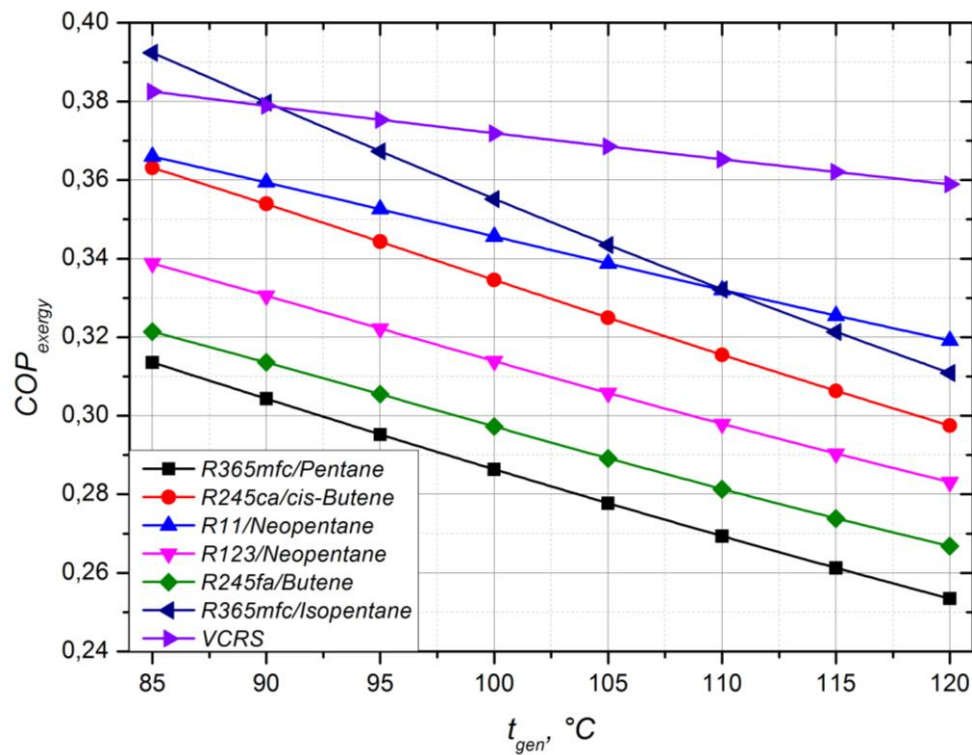


Figure 4.2 Dependence of exergetic COP from generation temperature.

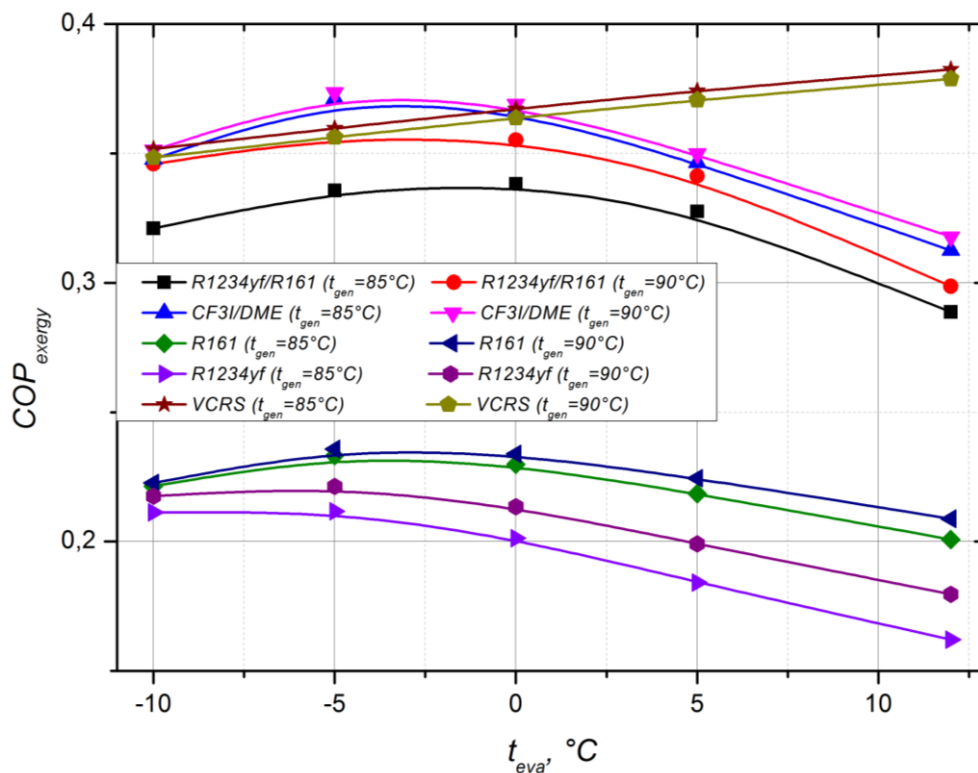


Figure 4.3 Dependence of exergetic COP from the evaporation temperature.

4.4 Heat driven jet thermo-transformers exergetic balances

Previously completed schemes of ERS exergetic balances running on R-12 in air conditioning mode and using electricity driven pump showed that the main exergy losses occur in the steam generator and refrigerant pump. Although the quantitative pump consumes only about 5-10% of the electricity, accordingly to generator heat load, the exergy losses in it reach 30-40% from the ERS condenser losses (Eq. 4.15).

$$E_{in} = E_{out} + \sum D \quad (4.15)$$

$$E_{in} = E'_{cond} + E'_{gen} \quad (4.16)$$

$$E_{out} = E_{eva} + E''_{gen} + D_{ej} + D_{e.v.} + D_{pump} + D_{gen} + D_{cond} + D_{eva} \quad (4.17)$$

Using a thermopump for the ERS scheme can significantly reduce these losses. The evolution of ERS exergetic COP growth can be traced from the water steam ERS to refrigerant- lever-thermopump ERS [18]. Exergetic COP of water steam ERS was about 3%, because of scheme components: electrically driven pump, main ejector, used for compressing the vapor coming from the evaporator, and two or three-stage auxiliary ejectors that served for air removal. Refrigerant ERS studied by L.S. Krasnyuk [19], already provided exergetic COP about 9.4%.

The experimental ERS prototype running on R142b and using gravitational type thermopump with the parameters of $t_{gen}=85^{\circ}\text{C}$, $t_{cond}=35^{\circ}\text{C}$ and $t_{eva}=12^{\circ}\text{C}$ has shown the exergetic COP about 11.8%. In this case, the ejector exergy losses totaled 0.222kW per 1kW of cooling capacity, i.e., more than half of the losses in the system (Table 4.1).

Exergy loss in the thermopump is comparable to the loss of exergy in the throttle valve, which is the lowest in the system.

Table 4.1 Component exergy losses in a single fluid ERS (R142b)

$t_{eva}, ^\circ\text{C}$	D_{ej}, W	$D_{e.v.}, \text{W}$	D_{pump}, W	D_{gen}, W	D_{cond}, W	D_{eva}, W
16	181.1	4.58	6.93	34	109.5	14.3
14	200	5.63	7.69	36	121.5	14.5
12	222.4	6.81	8.56	38	135.2	14.7
10	248.7	8.13	9.54	41	150.7	14.9
8	279.9	9.58	10.66	44	168.5	15.1
6	316.7	11.18	11.96	48	188.9	15.3

Table 4.2 Component exergy losses in a BERS (R11/Butane)

$t_{eva}, ^\circ\text{C}$	D_{ej}, W	$D_{e.v.}, \text{W}$	D_{pump}, W	D_{gen}, W	D_{cond}, W	D_{eva}, W
16	71.5	0.74	1.92	19.4	144	14.3
14	73.3	1.09	2.08	21	145	14.5
12	76.4	1.49	2.26	22.8	147	14.7
10	79.7	1.97	2.46	24.7	148	14.9
8	84.3	2.52	2.68	26.9	150	15.1
6	89.7	3.14	2.92	29.3	152	15.3

Table 4.3 and 4.4 represents exergy loss in single fluid ejector operating on R1233zd(E) and binary fluid R1233zd(E)/Butane at $t_{gen}=90^\circ\text{C}$, $t_{cond}=35^\circ\text{C}$ and various evaporation temperatures.

Table 4.3 Component exergy losses in a single fluid ERS (R1233zd(E))

$t_{eva}, ^\circ\text{C}$	D_{ej}, W	$D_{e.v.}, \text{W}$	D_{pump}, W	D_{gen}, W	D_{cond}, W	D_{eva}, W
16	175.55	4.001	3.872	52.575	78.06	14.289
15	188.259	4.487	4.142	55.353	83.520	14.388
14	201.80	5.00	4.42	58.307	89.276	14.48
12	235.85	6.146	5.11	65.58	103.15	14.69
10	273.13	7.426	5.862	73.57	118.1	14.890
8	320.152	8.854	6.774	83.54	136.56	15.107

**Table 4.4 Component exergy losses in a single fluid ERS
(R1233zd(E)/Butane)**

$t_{eva}, ^\circ\text{C}$	D_{ej}, W	$D_{e.v.}, \text{W}$	D_{pump}, W	D_{gen}, W	D_{cond}, W	D_{eva}, W
15	83.932	5.415	6.406	50.85	184.705	21.0159
14	92.926	5.28	6.65	52.862	184.55	21.02
12	112.08	5.062	7.244	57.514	186.47	21.04
10	133.33	4.838	7.899	62.720	188.587	21.05
8	156.350	4.615	8.612	68.395	190.66	21.07
6	181.89	4.392	9.4060	74.723	192.79	21.09

Fig. 4.4 shows the exergy flow diagram for binary ERS that works on R11+R600 mixture. The main exergy losses in the system occur in ejector and condenser. Exergy losses in the ejector were about 0.076kW per 1kW of cooling capacity else being equal (Table 4.2). This is almost three times lower than the previous case, a single-fluid ERS. Exergy COP of such binary ERS is about 25.5%.

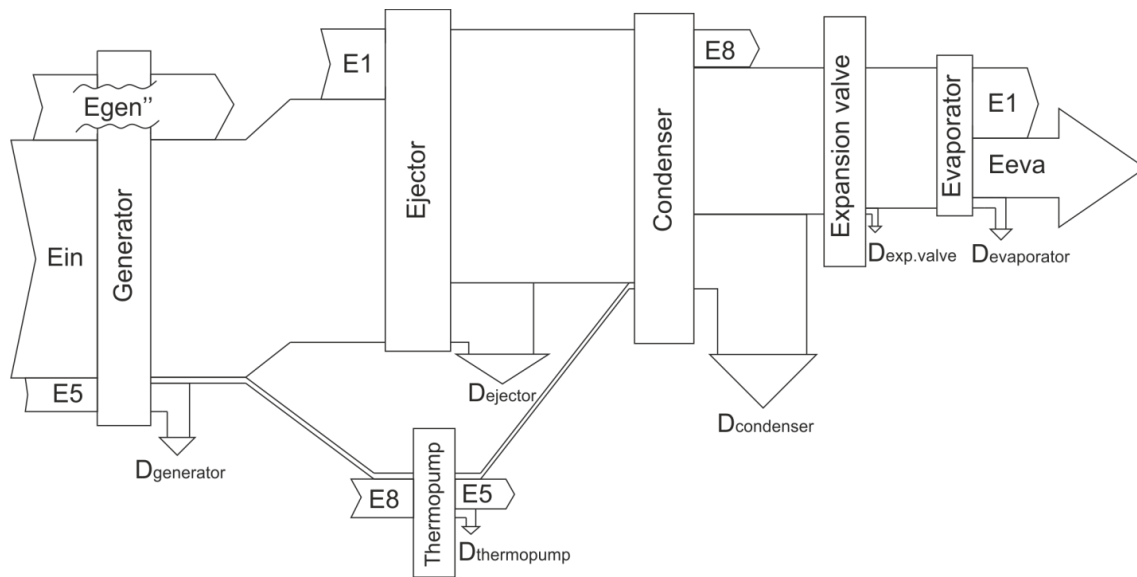


Figure 4.4 The scheme of exergetic flows in BERS. E1 – exergy flow from evaporator to ejector; E5 – exergy flow from thermopump to vapour generator; E8 – exergy flow into thermopump.

It should be noted that the loss of exergy in thermopump is increasing with the lowering of temperature level. This is due to the fact of thermopump COP reduction, associated with the density difference between the liquid and vapor refrigerant (Eq. 4.18):

$$COP_{pump} = 1 - \frac{\rho_{vap}}{\rho_{liq}} \quad (4.18)$$

COP of thermopump for R142b in air-conditioning mode is 0.935.

4.5 Results and discussion on Chapter 4.

Exergy analysis of the heat-driven compressor systems and ejector refrigeration systems showed that the binary fluid ERS have exergetic COP approaching very close to exergetic COP of the compressor systems, and in some cases exceeds them. Multicomponent refrigerant auto-cascades at different compositions show reduced exergetic COP at higher generation temperatures and are inferior to the

vapor-compression systems on about 15-40% depending on the composition and final evaporation temperature.

Analysis of exergy destruction in various units of ERS operating on single and binary fluids, it should be noted that exergy destruction in binary fluid ejector is lower than in single fluid. That is caused by the velocity difference of working and secondary fluid, i.e., processes in binary fluid ejector are more reversible.

Processes in expansion valve and thermopump almost similar in the binary and single fluid systems. Higher exergy destruction in binary fluid system comparing to single fluid is caused by higher temperature difference during the condensation process, i.e., condensation process flows in a wide range of temperatures. Thus, it is reasonable to provide exergy loss evaluation in fractionating condenser varying ambient temperature. In this case, losses decrease significantly.

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CONCLUSIONS

Conclusions

In this thesis, four main tasks were performed:

- 1 CFD modeling of the binary fluid ejector, operating in air conditioning mode was completed and verified;
- 2 Development of schematic solutions for air conditioning in various climate zones, as well as heating and their integrated solutions, were presented;
- 3 Theoretical analysis of the influence of thermophysical properties of refrigerant and their mixtures on ejector efficiency in limit and off-design conditions was discussed;
- 4 Experimental and industrial verification of quality and accuracy of ejector modeling was provided.

Statement of the first task is dedicated by the need to design and manufacture ejectors with rigorous requirements to minimize losses. It could not be provided only by calculations based on empirical dependencies and assumptions that distorted real flow pattern, shock waves, turbulence zones, and other sources of losses.

k-w SST turbulence model was used for modeling since it provides accurate results as for the free flow (core flow) and flows near to the wall. The difference between calculated and modeling results do not exceed 2.5%, that fits into experimental error, while ejectors manufactured according to the k-e model provides difference up to 10%.

During working on the thesis, more than 20 schematics were developed for various operating conditions, including thermovacuum drying.

Developed air-conditioning schematic solutions are aimed for an application not only in a residential area but also for commercial and industrial application. Also, the possibility of renewable and alternative heat source application was considered.

Restrict requirements for environmental protection, especially for ozone layer destruction and increasing greenhouse effect, lead to the exclusion of many available and affordable refrigerants and defined a search for new solutions. Therefore, for industry development, it is necessary to study new promising substances that have now appear and need to be promoted on the market. This work was focused on environmentally safe refrigerants and their compositions, such as R1233zd(E)/Butane, that can be used in a wide range of working temperature as an air-conditioning and heat pump system.

The energy efficiency of BERS is close to vapor compression systems at the same time this durability and reliability over exceeded that of vapor compression by 2-3 times, and price lower by 30-50%. Provided exergy analysis shown that BERS exergy efficiency close to the vapor compression system and significantly exceeds single fluid ERS.

Theoretical and experimental study of ERS and BERS allows solving their stable functioning at variable operating conditions. Developed compensation approach and algorithm for automatic control system makes it possible to reduce the influence of operating factors. In primary respect, this problem was solved by modeling off-design conditions operations. An interesting fact that varying mass fractions are an additional degree of freedom that allows stabilizing the operating parameters of the BERS.

As a result of theoretical study, a set of parameters for binary fluid selection was developed. That expands an area of application of ejector thermotransformers, which have not only good efficiency and reliability, but also environmentally safe.

APPENDIXES

APPENDIX A. Refrigerant Safety Properties.

FLUID NAME	M [KG/KMOL]	T _{NB} [C]	T _{CRIT} [C]	P _{CRIT} [MPa]	ODP	GWP	CAS REGISTR Y NUMBER	ASHRAE 34 SAFETY GROUP	NFPA 704
Butane	58.12	-0.49	151.98	3.80	0	4	106-97-8	A3	1/4/0/
Cyclohexane	84.16	80.71	280.45	4.08			110-82-7		1/3/0/
Cyclopentane	70.13	49.26	238.57	4.57			287-92-3		1/3/0/
Cyclopropane	42.08	-31.48	125.15	5.58			75-19-4		1/4/0/
Decane	142.28	174.12	344.55	2.10			124-18-5		1/2/0/
Diethyl ether	74.12	34.45	193.55	3.64		4 ± 2	60-29-7		1/4/1/
Dimethyl carbonate	90.08	90.11	283.85	4.91			616-38-6		3/3/0/
Dimethyl ether	46.07	-24.78	127.23	5.34		1	115-10-6	A3	2/4/1/
Dodecafluoro-2-methylpentan-3-one	316.04	49.05	168.66	1.87	0	1	756-13-8		3/0/1
Dodecane	170.33	216.29	384.95	1.82			112-40-3		1/2/0/
Ethanol	46.07	78.42	241.56	6.27			64-17-5		2/3/0/
Ethylbenzene	106.17	136.16	343.97	3.62			100-41-4		2/3/0/
Heavy water	20.03	101.39	370.70	21.67			7789-20-0		
Heptane	100.20	98.38	266.98	2.74			142-82-5		1/3//
Hexane	86.18	68.71	234.67	3.03			110-54-3		/3/0/
Isobutane	58.12	-11.75	134.66	3.63	0	3	75-28-5	A3	0/4/0/

FLUID NAME	MOLAR MASS [KG/KMOL]	NORMAL BOILING TEMP. [C]	CRITICAL TEMP. [C]	CRITICAL PRESSURE [MPA]	ODP	GWP	CAS NUMBER	ASHRAE 34 SAFETY GROUP	NFPA 704
Isobutene	56.11	-7.00	144.94	4.01			115-11-7		2/4/1/
Isohexane	86.18	60.21	224.55	3.04			107-83-5		1/3/0/
Isooctane	114.23	99.21	270.85	2.57			540-84-1		1/3/0/
Isopentane	72.15	27.83	187.20	3.38			78-78-4	A3	1/4/0/
MD2M	310.69	194.36	326.25	1.23			141-62-8		2/2/1
MD3M	384.84	229.87	355.21	0.95			141-63-9		
MD4M	458.99	259.57	380.05	0.88			107-52-8		
MDM	236.53	152.53	290.94	1.42			107-51-7		
Methanol	32.04	64.48	240.23	8.22		2.8	67-56-1		1/3/0/
Methylcyclohexane	98.19	100.86	299.05	3.47			108-87-2		2/3/0/
Mm	162.38	100.25	245.55	1.94			107-46-0		2/3/1
M-xylene	106.17	139.06	343.74	3.53			108-38-3		2/3/0/
Neopentane	72.15	9.50	160.59	3.20			463-82-1		2/4/0/
Nonane	128.26	150.76	321.40	2.28			111-84-2		1/3/0/
Octane	114.23	125.62	296.17	2.50			111-65-9		1/3/0/

FLUID NAME	MOLAR MASS [KG/KMOL]	NORMAL BOILING TEMP. [C]	CRITICAL TEMP. [C]	CRITICAL PRESSURE [MPA]	ODP	GWP	CAS NUMBER	ASHRAE 34 SAFETY GROUP	NFPA 704
O-xylene	106.17	144.37	357.11	3.74			95-47-6		2/3/0/
Pentane	72.15	36.06	196.55	3.37	0	4 ± 2	109-66-0	A3	1/4/0/
Perfluorobutane	238.03	-2.26	113.18	2.32			355-25-9		
Perfluoropentane	288.03	29.75	147.41	2.05			678-26-2		
Propane	44.10	-42.11	96.74	4.25		3.3	74-98-6	A3	2/4/0/
Propylcyclohexane	126.24	156.71	357.65	2.86			1678-92-8		
Propylene	42.08	-47.62	91.06	4.56		1.8	115-07-1	A3	1/4/1/
Propyne	40.06	-25.14	129.23	5.63			74-99-7		1/4/3/
P-xylene	106.17	138.32	343.02	3.53			106-42-3		2/3/0/
R11	137.37	23.71	197.96	4.41	1	4750	75-69-4	A1	
R113	187.38	47.59	214.06	3.39	0.85	6130	76-13-1	A1	
R114	170.92	3.59	145.68	3.26	0.58	10000	76-14-2	A1	
R115	154.47	-39.22	79.95	3.13	0.57	7370	76-15-3	A1	
R12	120.91	-29.75	111.97	4.14	0.82	10900	75-71-8	A1	
R1216	150.02	-30.34	85.75	3.15			116-15-4		
R123	152.93	27.82	183.68	3.66	0.01	77	306-83-2	B1	1/0/1

FLUID NAME	MOLAR MASS [KG/KMOL]	NORMAL BOILING TEMP. [C]	CRITICAL TEMP. [C]	CRITICAL PRESSURE [MPA]	ODP	GWP	CAS NUMBER	ASHRAE 34 SAFETY GROUP	NFPA 704
R1234YF	114.04	-29.49	94.70	3.38	0	4	754-12-1	A2	
R1234ZE	114.04	-18.97	109.36	3.63	0	6	29118-24-9		
R124	136.48	-11.96	122.28	3.62	0.02	609	2837-89-0	A1	
R125	120.02	-48.09	66.02	3.62	0	3500	354-33-6	A1	
R134A	102.03	-26.07	101.06	4.06	0	1430	811-97-2	A1	
R141B	116.95	32.05	204.35	4.21	0.12	725	1717-00-6	A2	2/1/0
R142B	100.50	-9.12	137.11	4.06	0.06	2310	75-68-3	A2	2/4/0/
R143A	84.04	-47.24	72.71	3.76	0	4470	420-46-2	A2	
R152A	66.05	-24.02	113.26	4.52	0	124	75-37-6	A2	
R161	48.06	-37.55	102.10	5.01			353-36-6		2/4/0/
R21	102.92	8.86	178.33	5.18	0.04	151	75-43-4	B1	
R218	188.02	-36.79	71.87	2.64	0	8830	76-19-7	A1	
R22	86.47	-40.81	96.15	4.99	0.04	1810	75-45-6	A1	
R227EA	170.03	-16.34	101.75	2.93		3220	431-89-0	A1	
R236EA	152.04	6.17	139.29	3.42	0	1410	431-63-0		
R236FA	152.04	-1.49	124.92	3.20	0	9810	690-39-1	A1	
R245CA	134.05	25.26	174.42	3.94	0	726	679-86-7		3/4/0
R245FA	134.05	15.14	154.01	3.65	0	1030	460-73-1	B1	

FLUID NAME	MOLAR MASS [KG/KMOL]	NORMAL BOILING TEMP. [C]	CRITICAL TEMP. [C]	CRITICAL PRESSURE [MPA]	ODP	GWP	CAS NUMBER	ASHRAE 34 SAFETY GROUP	NFPA 704
R32	52.02	-51.65	78.11	5.78	0	675	75-10-5	A2	1/4/0
R365MFC	148.07	40.19	186.85	3.27	0	794	406-58-6		0/3/1
R40	50.49	-23.98	133.66	3.80	0.02	13	74-87-3	B2	2/4/0/
RC318	200.04	-5.97	115.23	2.78	0	10300	115-25-3	A1	
RE143A	100.04	-23.58	104.77	3.64			421-14-7		
RE245CB2	150.05	5.61	133.66	2.89			22410-44-2		
RE245FA2	150.05	29.25	171.73	3.43			1885-48-9		3/0/0
RE347MCC	200.05	34.20	164.55	2.48			375-03-1		
TRIFLUOR IODOMET HANE	195.91	-21.86	123.29	3.95			2314-97-8		
WATER	18.02	99.97	373.95	22.06	0	0.2 ± 0.2	7732-18-5	A1	

APPENDIX B. Criteria of fluids selection for BERS

Working fluid	Refrigerant fluid	U	$T_{nb, wf}/T_{nb, wf}$	$T_{crit, wf}/T_{crit, rf}$	Z_{wf}/Z_{rf}	M_{rf}/M_{wf}	r_{rf}/r_{wf}	$(P_{gen, wf} \rho_{gen, wf}) / (P_{eva, rf} \rho_{eva, rf})$
CF3I	R161	0.134	1.0667	1.0563	0.7655	0.2453	5.8439	69.5830
CF3I	DME	0.114	1.0118	0.9902	0.7304	0.2352	6.8261	202.2084
CF3I	R152a	0.134	1.0087	1.0260	0.7402	0.3371	4.8140	141.2644
DME	Propane	0.161	1.0750	1.0824	0.7754	0.9572	1.3092	23.4111
MM	Hexane	0.459	1.0922	1.0214	0.9578	0.5307	1.8860	120.5965
R11	Isobutane	0.416	1.1356	1.1552	0.9329	0.4231	2.2652	22.3038
R11	Butane	0.345	1.0887	1.1082	0.9176	0.4231	2.4656	51.6841
R114	DME	0.211	1.1142	1.0461	0.8406	0.2695	4.3705	41.9625
R114	Isobutane	0.210	1.0587	1.0270	0.8276	0.3401	3.5633	97.2745
R114	Butane	0.155	1.0150	0.9852	0.8141	0.3401	3.8785	225.4114
R114	neopentane	0.104	0.9791	0.9656	0.8077	0.4221	3.2502	384.9804
R123	Isobutane	0.379	1.1514	1.1202	0.9172	0.3801	2.4613	22.3013
R123	Butane	0.320	1.1038	1.0746	0.9022	0.3801	2.6790	51.6781
R123	R21	0.387	1.0672	1.0119	0.8889	0.6730	1.6987	61.3140
R123	neopentane	0.278	1.0648	1.0532	0.8951	0.4718	2.2450	88.2611
R1234yf	Propane	0.213	1.0547	0.9945	0.5607	0.3867	5.5656	102.4496
R1234yf	DME	0.145	0.9811	0.9188	0.5353	0.4040	6.4739	316.6449

Working fluid	Refrigerant fluid	U	$T_{nb,wf}/T_{nb,rf}$	$T_{crit,wf}/T_{crit,rf}$	Z_{wf}/Z_{rf}	M_{rf}/M_{wf}	r_{rf}/r_{wf}	$(P_{gen,wf}\rho_{gen,wf})/(P_{eva,rf}\rho_{eva,rf})$
R1234YF	Isobutane	0.124	0.9321	0.9020	0.5271	0.5097	5.2782	734.0246
R1234ZE	DME	0.188	1.0234	0.9554	0.6675	0.4040	4.2169	162.7486
R1234ZE	Isobutane	0.162	0.9724	0.9380	0.6572	0.5097	3.4381	377.2726
R161	Propane	0.272	1.0196	1.0146	0.6182	0.9175	1.9244	67.0795
R161	DME	0.187	0.9485	0.9374	0.5902	0.9586	2.2385	207.3254
R21	DME	0.406	1.1355	1.1276	0.9058	0.4476	2.2735	20.0556
R21	Isobutane	0.355	1.0788	1.1071	0.8919	0.5647	1.8536	46.4916
R21	Butane	0.267	1.0343	1.0620	0.8773	0.5647	2.0175	107.7337
R227EA	DME	0.142	1.0340	0.9364	0.6206	0.2709	7.0447	224.2610
R236FA	DME	0.210	1.0940	0.9942	0.7549	0.3030	4.3036	75.5017
R236FA	R152a	0.282	1.0906	1.0302	0.7651	0.4344	3.0351	52.7461
R236FA	Isobutane	0.155	1.0394	0.9761	0.7433	0.3823	3.5088	175.0228
R236FA	R21	0.189	0.9635	0.8817	0.7204	0.6769	2.4216	481.1986
R236FA	Butane	0.106	0.9965	0.9364	0.7312	0.3823	3.8192	405.5754
R245CA	Isobutane	0.371	1.1411	1.0975	0.8953	0.4336	2.1514	27.3726
R245CA	Butane	0.300	1.0940	1.0528	0.8806	0.4336	2.3417	63.4297
R245CA	R21	0.364	1.0577	0.9913	0.8676	0.7678	1.4848	75.2568
R245CA	neopentane	0.244	1.0553	1.0319	0.8737	0.5382	1.9623	108.3317

Working fluid	Refrigerant fluid	U	$T_{nb,wf}/T_{nb,rf}$	$T_{crit,wf}/T_{crit,rf}$	Z_{wf}/Z_{rf}	M_{rf}/M_{wf}	r_{rf}/r_{wf}	$(P_{gen,wf}\rho_{gen,wf})/(P_{eva,rf}\rho_{eva,rf})$
R245FA	DME	0.337	1.1607	1.0669	0.8686	0.3437	2.9068	23.5221
R245FA	Isobutane	0.282	1.1029	1.0474	0.8552	0.4336	2.3699	54.5273
R245FA	Butane	0.218	1.0573	1.0048	0.8413	0.4336	2.5795	126.3546
R245FA	R21	0.241	1.0223	0.9461	0.8288	0.7678	1.6356	149.9145
R245FA	Neopentane	0.160	1.0199	0.9848	0.8347	0.5382	2.1616	215.8011
R365MFC	Butane	0.487	1.1492	1.0820	0.9214	0.3925	2.3364	25.8703
R365MFC	R245fa	0.565	1.0869	1.0769	0.9127	0.9053	1.2387	41.1105
R365MFC	Isopentane	0.299	1.0411	0.9992	0.9001	0.4872	2.2080	184.4995
R365MFC	Pentane	0.205	1.0134	0.9793	0.8958	0.4872	2.3368	365.1592
RC318	R152a	0.193	1.0724	1.0051	0.7354	0.3302	4.4913	82.7676
RC318	Isobutane	0.159	1.0221	0.9524	0.7145	0.2906	5.1923	274.6407
RC318	Butane	0.105	0.9799	0.9136	0.7028	0.2906	5.6516	636.4172

APPENDIX C. CFD modeling report data

R1233zd(E)

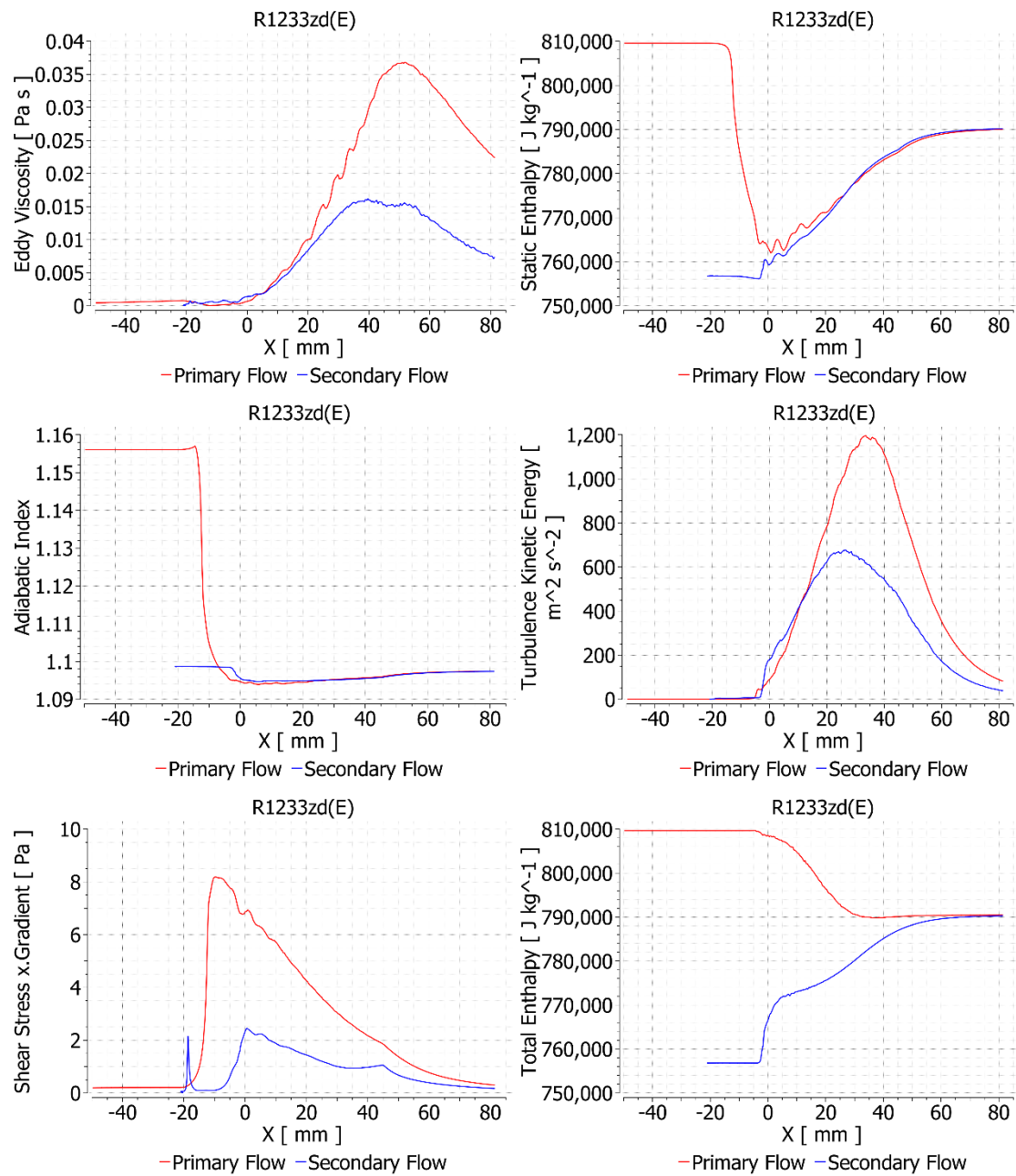


Figure C.1.1 Chart of Total Enthalpy, Static Enthalpy, Adiabatic Index, Turbulence Kinetic Energy, Shear Stress, Eddy Viscosity of R1233zd(E) ejector operating on $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

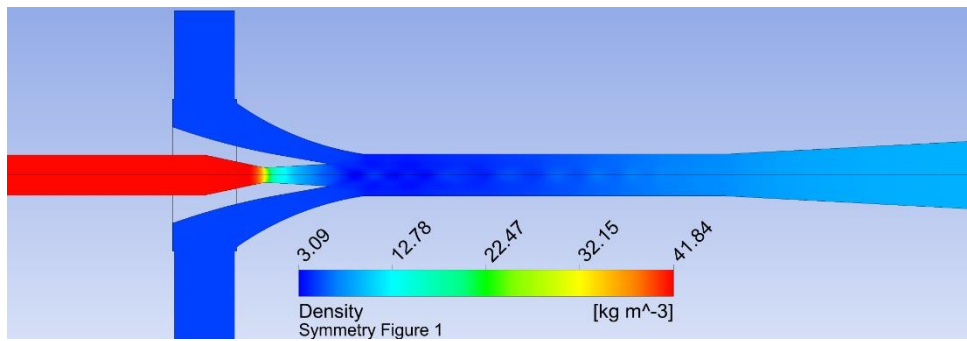


Figure C.1.2. Density distribution in ejector operating on R1233zd(E) at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

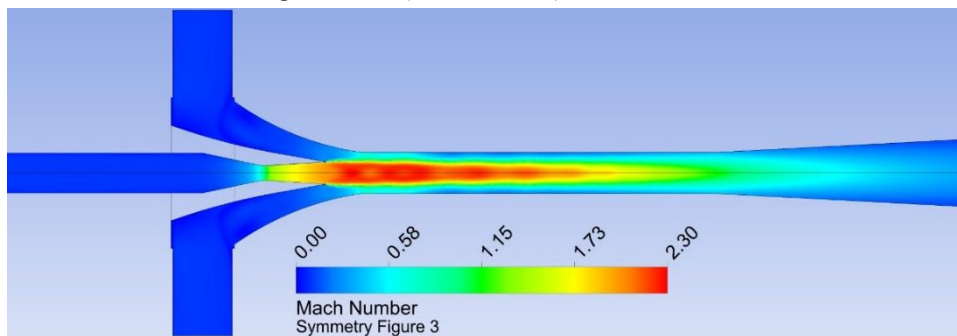


Figure C.1.3. Mach distribution in ejector operating on R1233zd(E) at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

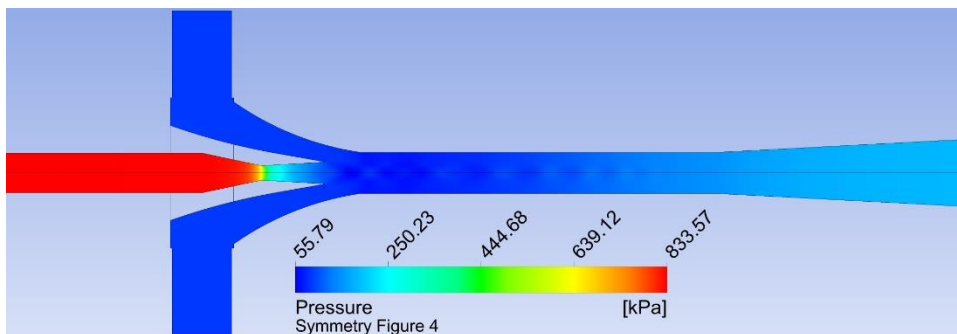


Figure C.1.4. Pressure distribution in ejector operating on R1233zd(E) at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

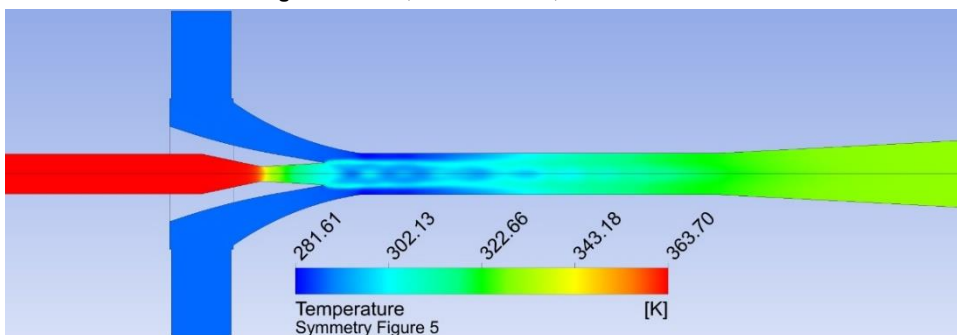


Figure C.1.5. Temperature distribution in ejector operating on R1233zd(E) at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

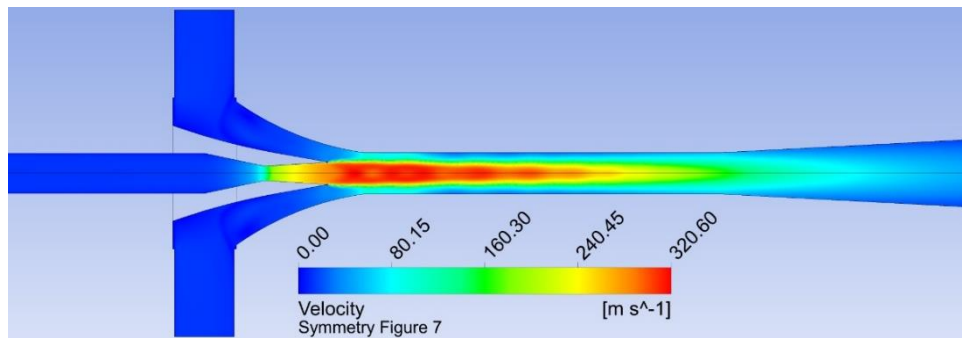


Figure C.1.6. Velocity distribution in ejector operating on R1233zd(E) at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

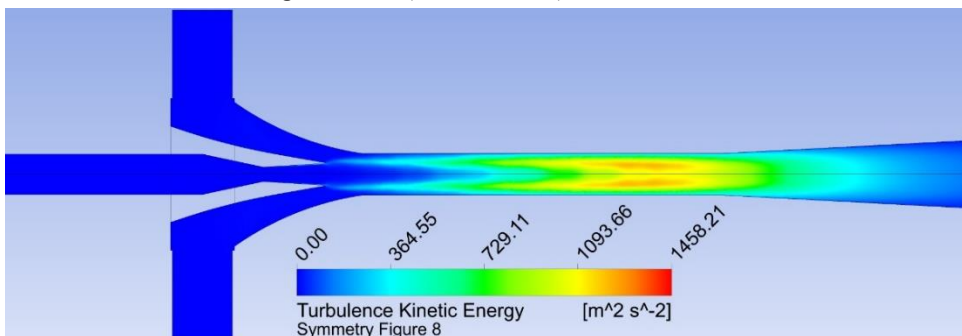


Figure C.1.7. Turbulence kinetic energy distribution in ejector operating on R1233zd(E) at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

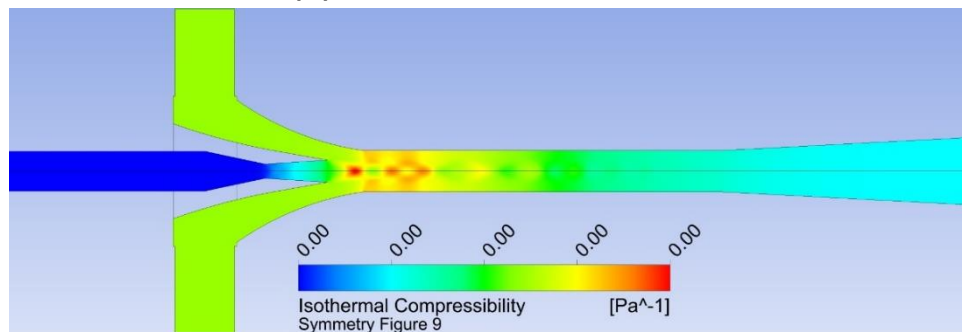


Figure C.1.8. Isothermal Compressibility distribution in ejector operating on R1233zd(E) at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

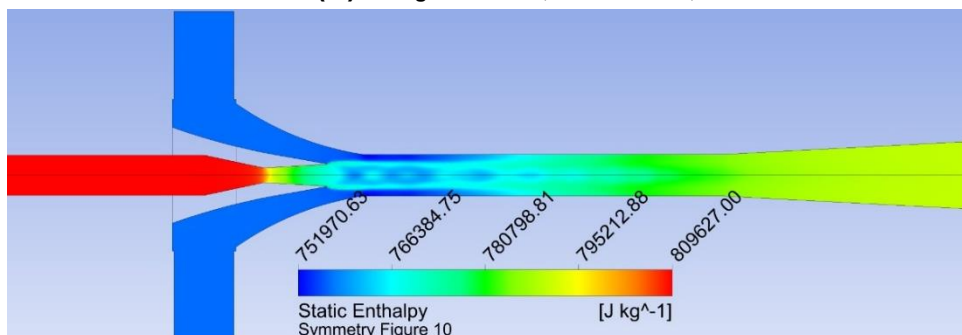


Figure C.1.9. Static Enthalpy distribution in ejector operating on R1233zd(E) at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

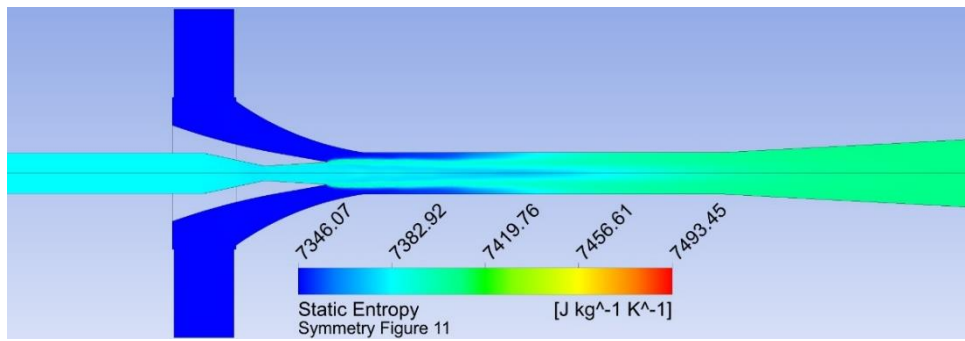


Figure C.1.10. Static Entropy distribution in ejector operating on R1233zd(E) at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

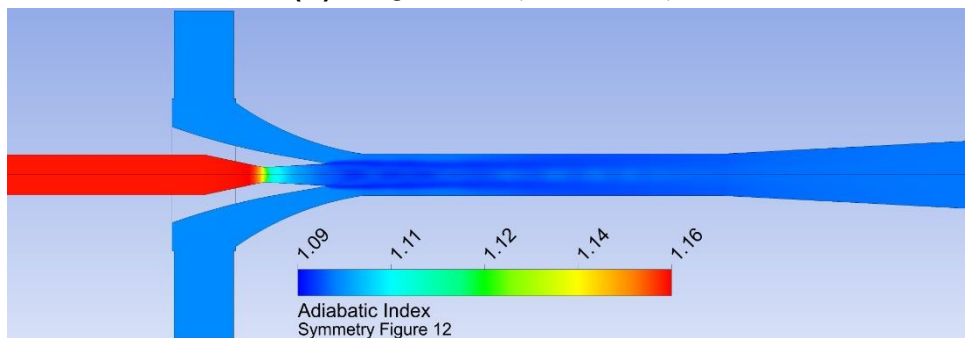


Figure C.1.11. Adiabatic Index distribution in ejector operating on R1233zd(E) at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

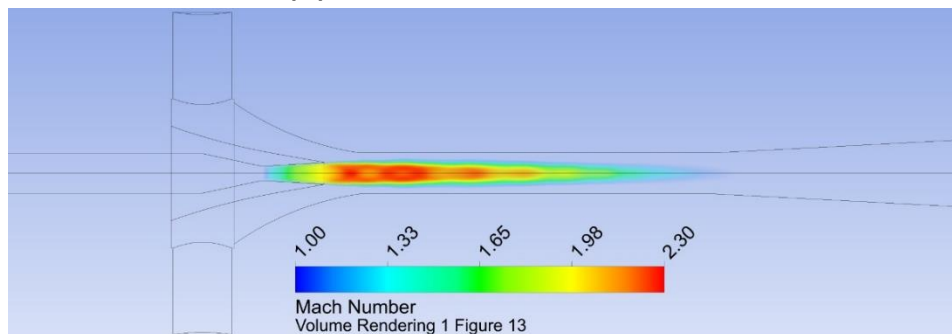


Figure C.1.12. Area of Mach Number ($M>1$) distribution in ejector operating on R1233zd(E) at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

R1233zd(E)/Butane

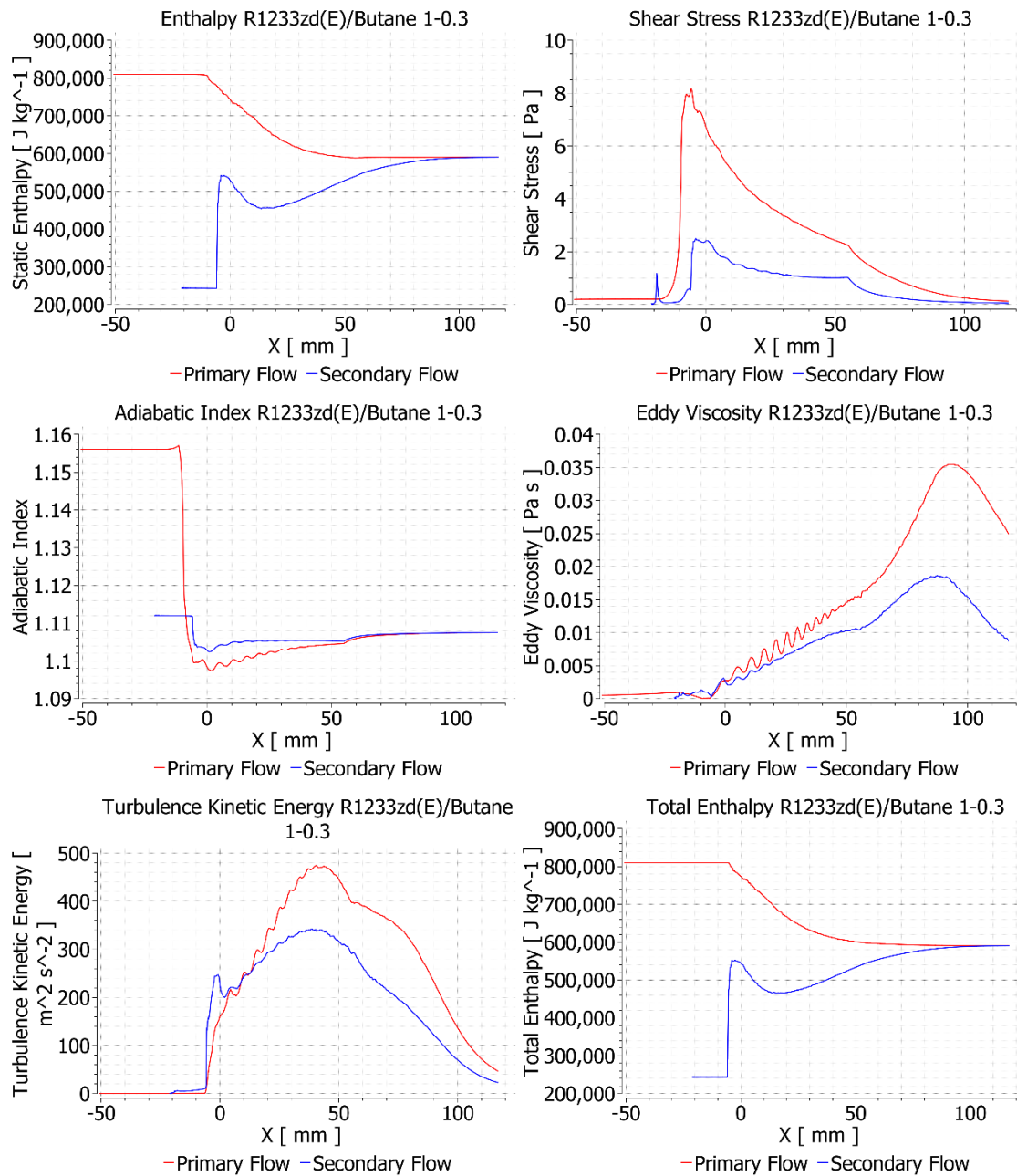


Figure C.2.1 Chart of Total Enthalpy, Static Enthalpy, Adiabatic Index, Turbulence Kinetic Energy, Shear Stress, Eddy Viscosity of R1233zd(E)/Butane ejector operating on $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}\text{C}$, $t_{\text{eva}}=15^{\circ}\text{C}$

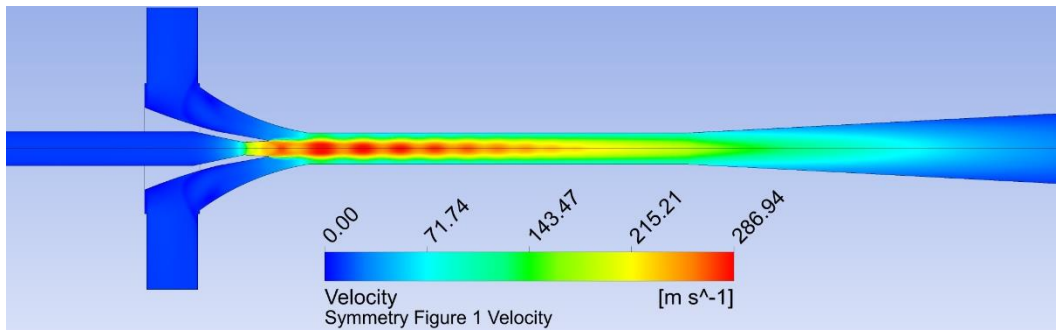


Figure C.2.2. Velocity distribution in ejector operating on R1233zd(E)/Butane at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

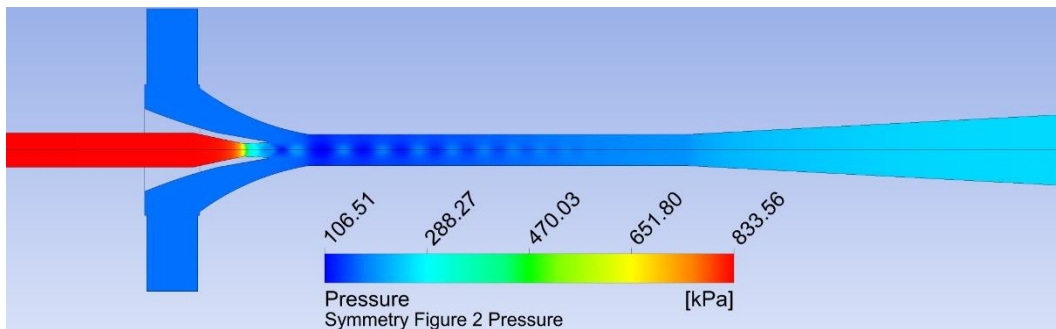


Figure C.2.3. Pressure distribution in ejector operating on R1233zd(E)/Butane at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

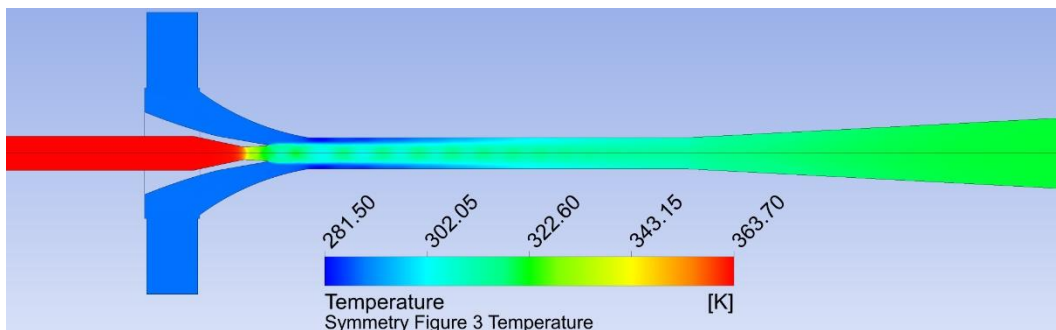


Figure C.2.4. Temperature distribution in ejector operating on R1233zd(E)/Butane at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

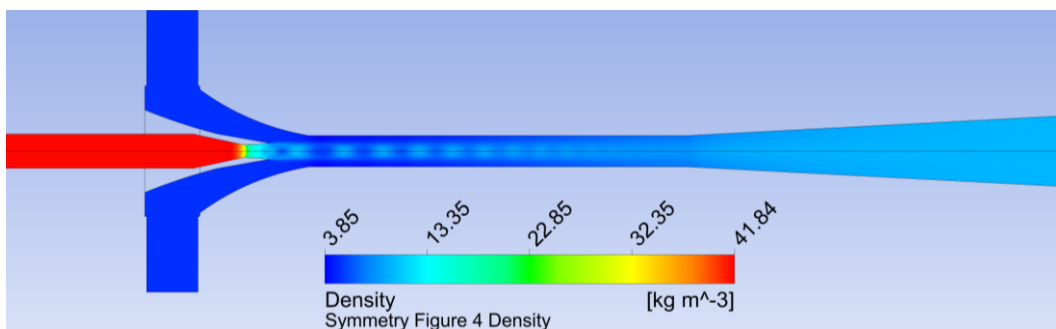


Figure C.2.5. Density distribution in ejector operating on R1233zd(E)/Butane at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

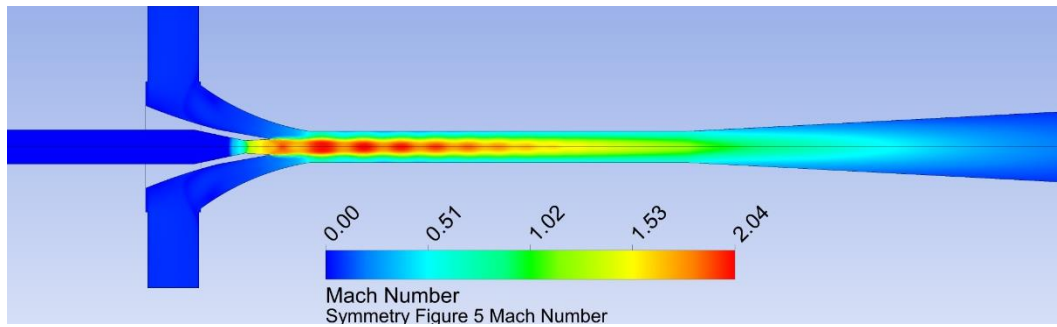


Figure C.2.6. Mach Number distribution in ejector operating on R1233zd(E)/Butane at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

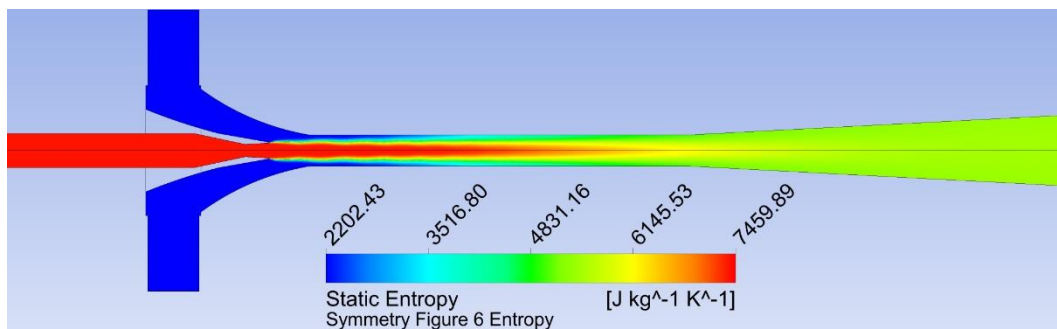


Figure C.2.7. Static Entropy distribution in ejector operating on R1233zd(E)/Butane at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

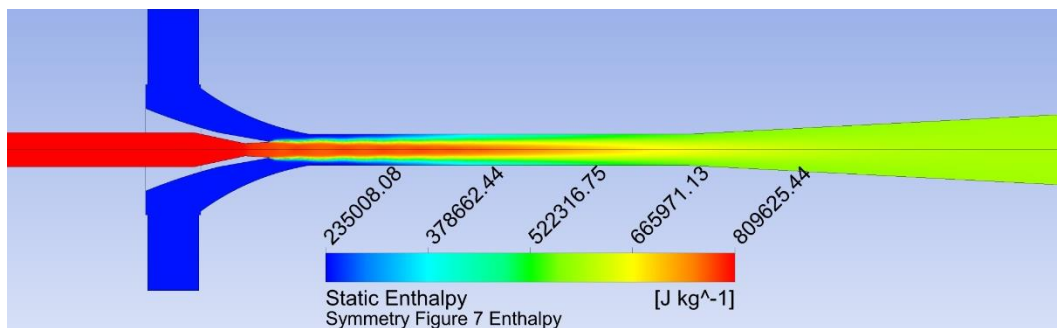


Figure C.2.8. Static Enthalpy distribution in ejector operating on R1233zd(E)/Butane at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

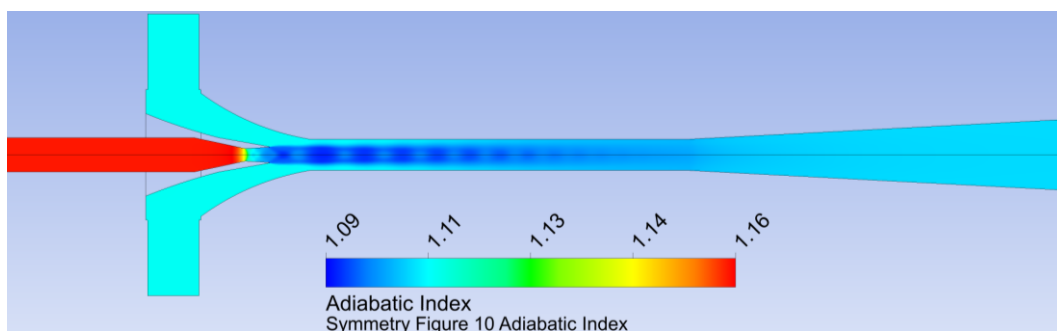


Figure C.2.9. Adiabatic Index distribution in ejector operating on R1233zd(E)/Butane at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

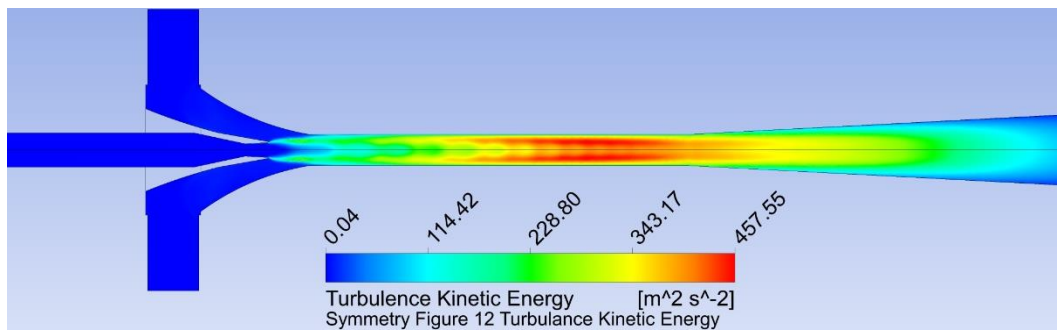


Figure C.2.10. Turbulence Kinetic Energy distribution in ejector operating on R1233zd(E)/Butane at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

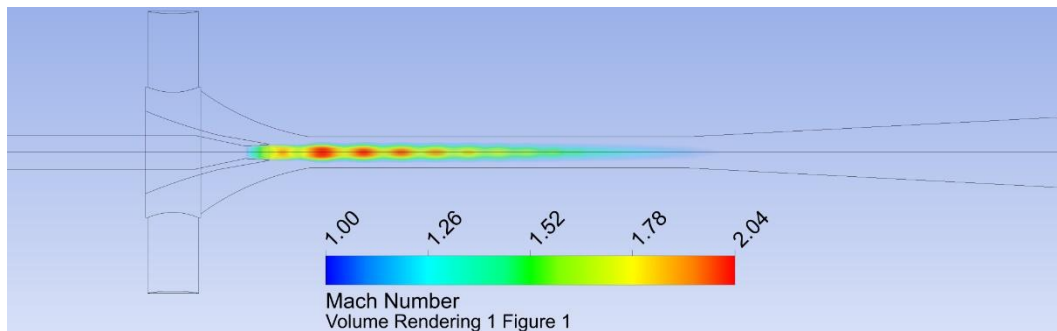


Figure C.2.11. Area of Mach Number ($M>1$) distribution in ejector operating on R1233zd(E)/Butane at $t_{\text{gen}}=90^{\circ}\text{C}$, $t_{\text{cond}}=35^{\circ}$, $t_{\text{eva}}=15^{\circ}\text{C}$

Steam/Air

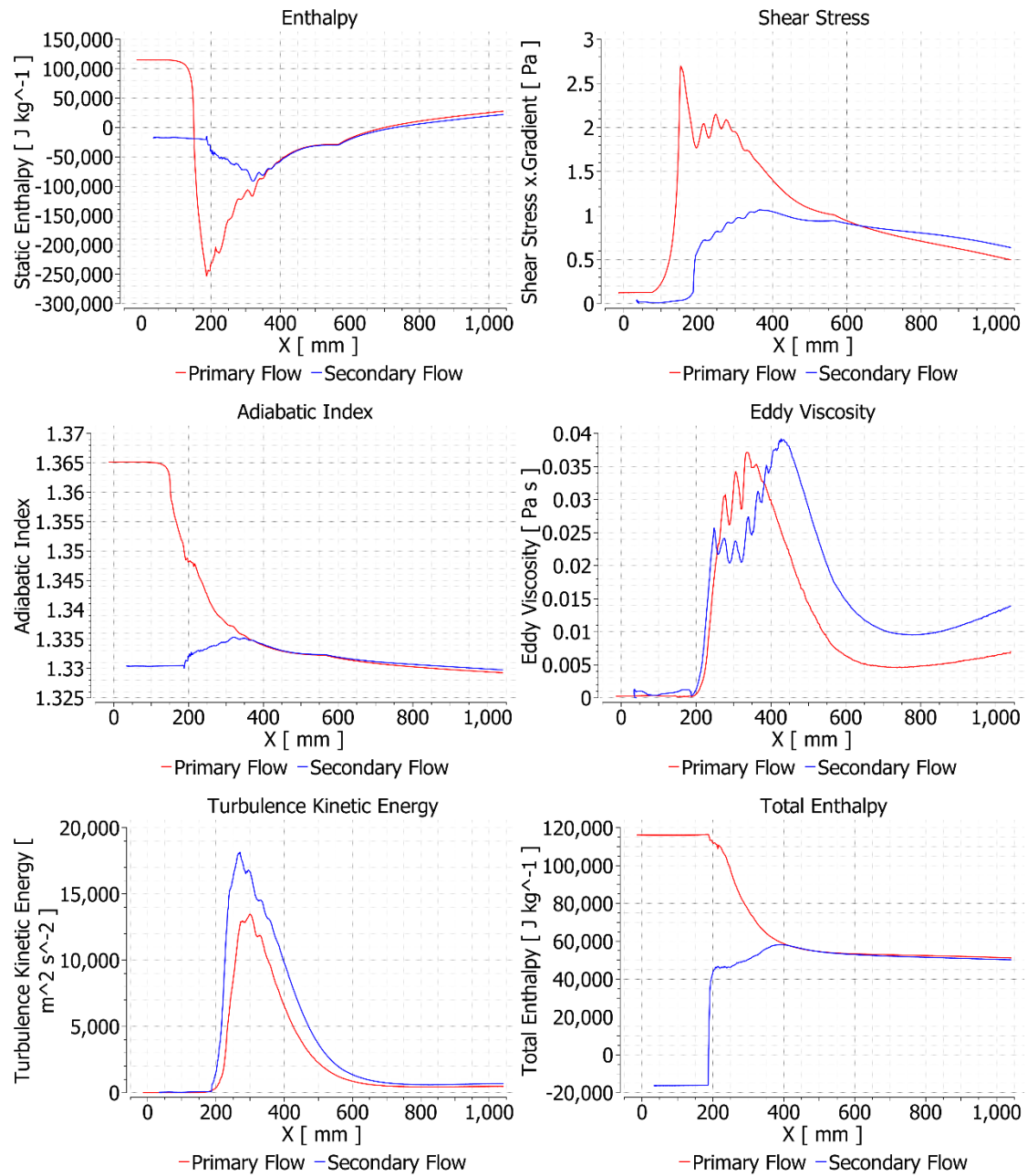


Figure C.3.1 Chart of Total Enthalpy, Static Enthalpy, Adiabatic Index, Turbulence Kinetic Energy, Shear Stress, Eddy Viscosity of Steam/Air at $t_{\text{gen}}=150^\circ\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^\circ\text{C}$, $P_{\text{eva}}=55\text{kPa}$.

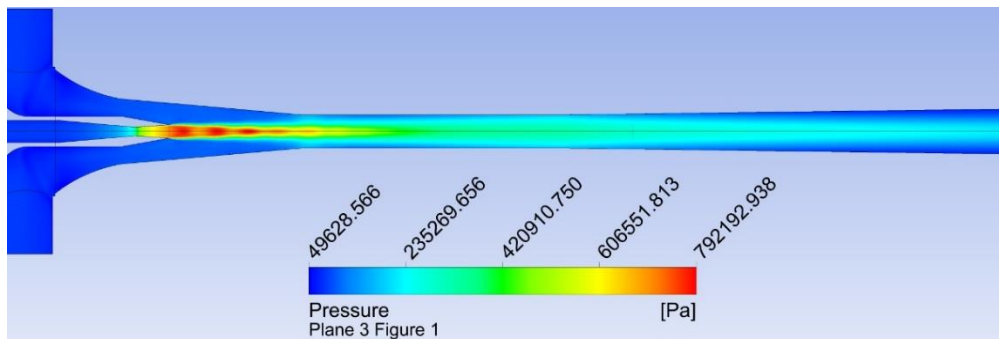


Figure C.3.2. Pressure distribution in ejector operating on Steam/Air at $t_{\text{gen}}=150^{\circ}\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^{\circ}\text{C}$, $P_{\text{eva}}=55\text{kPa}$.

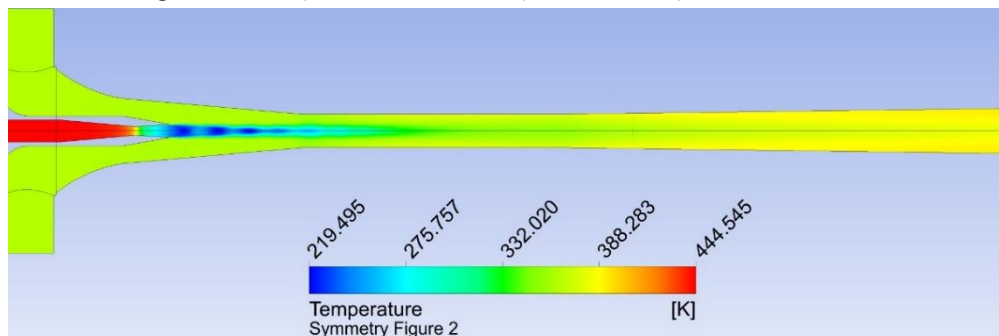


Figure C.3.3. Temperature distribution in ejector operating on Steam/Air at $t_{\text{gen}}=150^{\circ}\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^{\circ}\text{C}$, $P_{\text{eva}}=55\text{kPa}$.

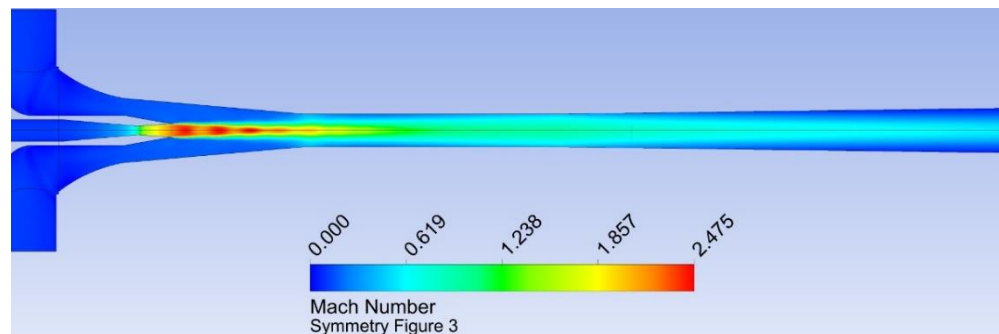


Figure C.3.4. Mach Number distribution in ejector operating on Steam/Air at $t_{\text{gen}}=150^{\circ}\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^{\circ}\text{C}$, $P_{\text{eva}}=55\text{kPa}$.

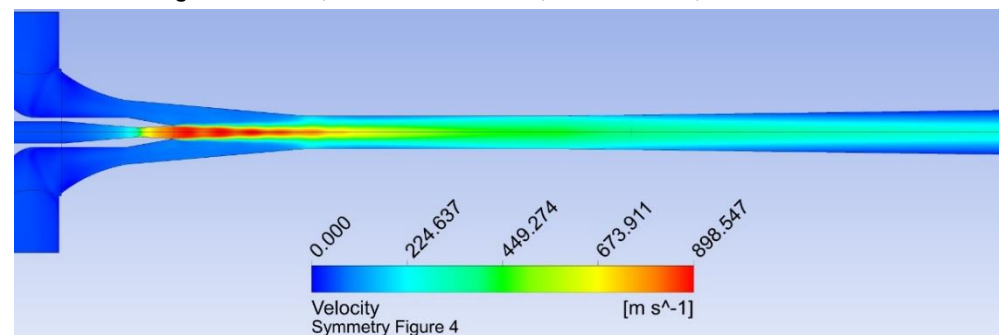


Figure C.3.5. Velocity distribution in ejector operating on Steam/Air at $t_{\text{gen}}=150^{\circ}\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^{\circ}\text{C}$, $P_{\text{eva}}=55\text{kPa}$.

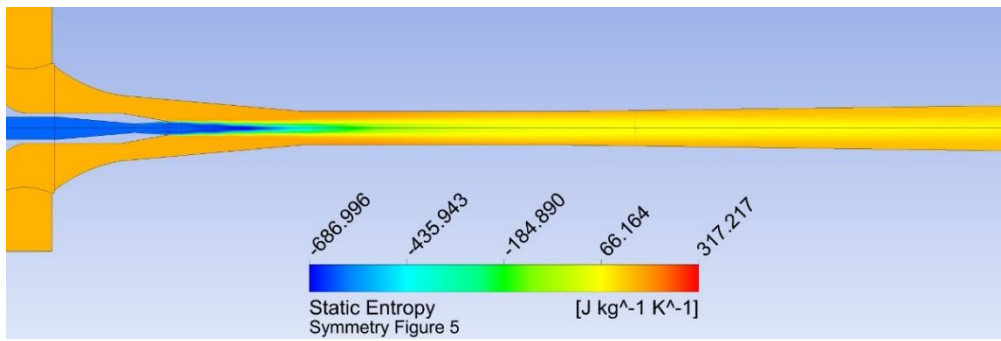


Figure C.3.6. Static Entropy distribution in ejector operating on Steam/Air at $t_{\text{gen}}=150^{\circ}\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^{\circ}\text{C}$, $P_{\text{eva}}=55\text{kPa}$.

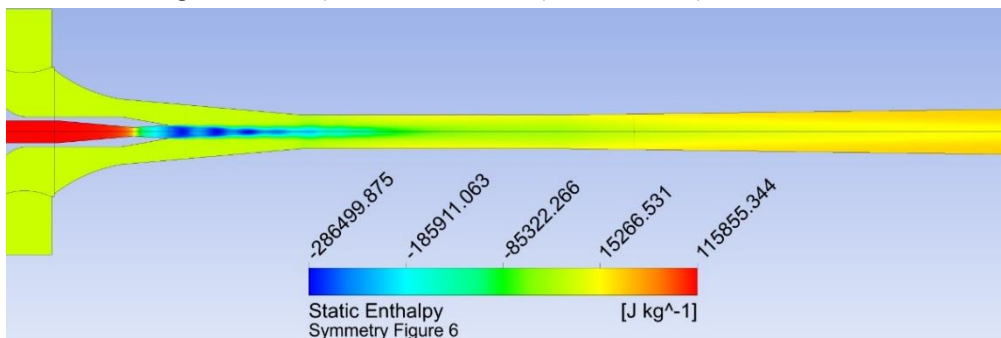


Figure C.3.7. Static Enthalpy distribution in ejector operating on Steam/Air at $t_{\text{gen}}=150^{\circ}\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^{\circ}\text{C}$, $P_{\text{eva}}=55\text{kPa}$.

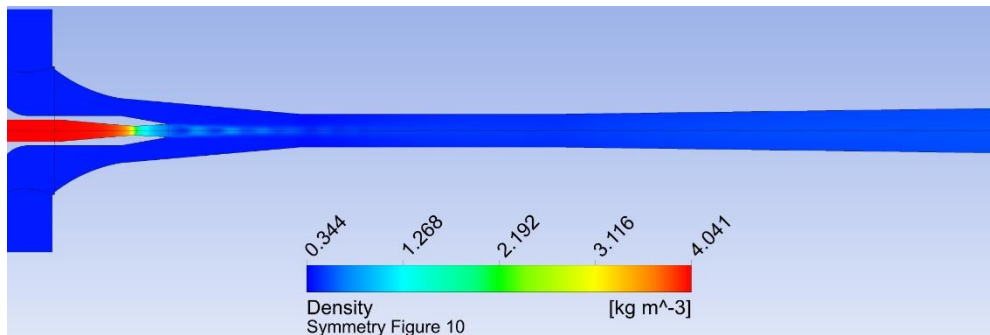


Figure C.3.8. Density distribution in ejector operating on Steam/Air at $t_{\text{gen}}=150^{\circ}\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^{\circ}\text{C}$, $P_{\text{eva}}=55\text{kPa}$.

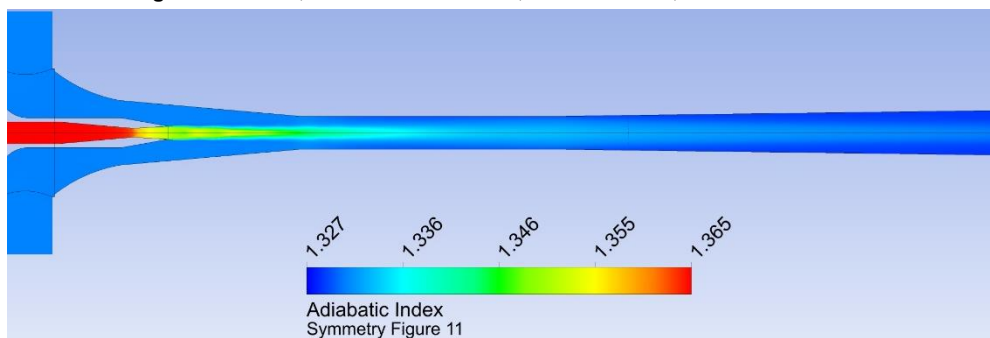


Figure C.3.9. Adiabatic Index distribution in ejector operating on Steam/Air at $t_{\text{gen}}=150^{\circ}\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^{\circ}\text{C}$, $P_{\text{eva}}=55\text{kPa}$.

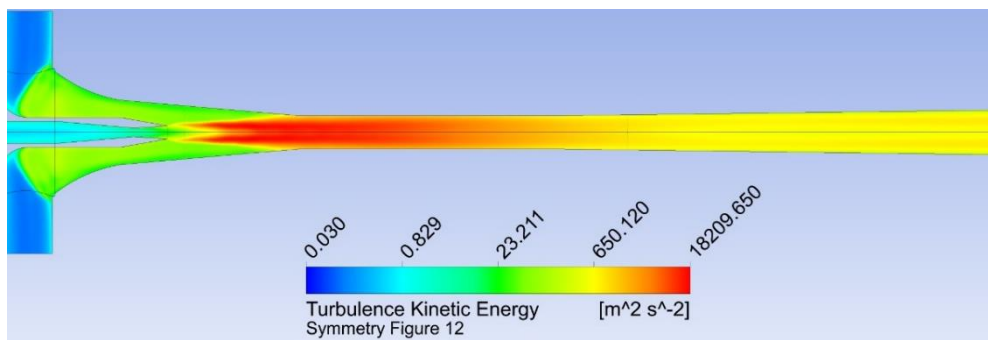


Figure C.3.10. Turbulence Kinetic Energy distribution in ejector operating on Steam/Air at $t_{\text{gen}}=150^{\circ}\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^{\circ}\text{C}$, $P_{\text{eva}}=55\text{kPa}$.

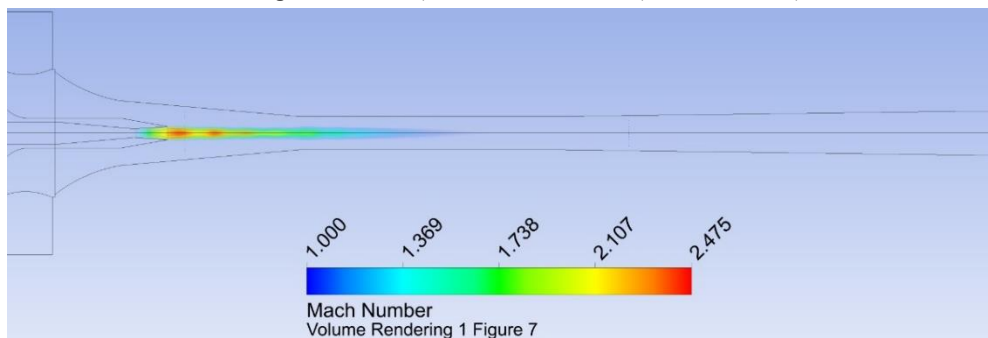


Figure C.3.11. Area of Mach Number ($M>1$) distribution in ejector operating on Steam/Air at $t_{\text{gen}}=150^{\circ}\text{C}$, $P_{\text{cond}}=101\text{kPa}$, $t_{\text{eva}}=40^{\circ}\text{C}$, $P_{\text{eva}}=55\text{kPa}$

APPENDIX D. Operating parameters and entrainment ratio results from CFX. R1233zd(E)/Butane

#	T _{gen} , K	P _{gen} , Pa	X _{gen}	T _{eva} , K	P _{eva} , Pa	X _{eva}	P _{cond} , Pa	U	U
DP 56	363.65	833559	1	288.65	169650	0.3	253894	0.62918	-0.629182
DP 57	363.65	833559	1	288.65	169650	0.3	270000	0.491767	-0.491759
DP 58	363.65	833559	1	288.65	169650	0.3	290000	0.124661	-0.124721
DP 59	363.65	833559	1	288.65	169650	0.3	310000	0	0.000142827
DP 60	363.65	833559	1	288.65	167451	0.35	250000	0.634067	-0.633835
DP 61	363.65	833559	1	288.65	167451	0.35	270000	0.461432	-0.46128
DP 62	363.65	833559	1	288.65	167451	0.35	290000	0.100059	-0.100036
DP 63	363.65	833559	1	288.65	167451	0.35	310000	0	0.000151423
DP 64	363.65	833559	1	288.65	164756	0.4	246955	0.630694	-0.630736
DP 65	363.65	833559	1	288.65	164756	0.4	270000	0.418895	-0.418575
DP 66	363.65	833559	1	288.65	164756	0.4	290000	0.0708381	-0.0708436
DP 67	363.65	833559	1	288.65	164756	0.4	310000	0	0.00021599
DP 68	363.65	833559	1	291.65	187731	0.3	258578	0.726856	-0.727085
DP 69	363.65	833559	1	291.65	187731	0.3	270000	0.686561	-0.686493
DP 70	363.65	833559	1	291.65	187731	0.3	290000	0.394624	-0.395174
DP 71	363.65	833559	1	291.65	187731	0.3	310000	0.0502772	-0.0502887
DP 74	363.65	993854	0.85	288.65	169650	0.3	253894	0.509835	-0.509818
DP 75	363.65	993850	0.85	288.65	169650	0.3	276717	0.509839	-0.509775
DP 84	363.65	993850	0.85	288.65	169650	0.3	260000	0.509837	-0.509819

#	T _{gen} , K	P _{gen} , Pa	X _{gen}	T _{eva} , K	P _{eva} , Pa	X _{eva}	P _{cond} , Pa	U	U
DP 85	363.65	993850	0.85	288.65	169650	0.3	270000	0.509845	-0.509859
DP 76	363.65	833559	1	288.65	169650	0.3	260000	0.604771	-0.604744
DP 77	363.65	833559	1	288.65	169650	0.3	280000	0.288757	-0.288858
DP 86	363.65	993850	0.85	288.65	169650	0.3	280000	0.509841	-0.509767
DP 87	363.65	993850	0.85	288.65	169650	0.3	290000	0.509838	-0.509853
DP 73	363.65	944344	0.9	288.65	169650	0.3	253.894	0.54555	-0.54556
DP 81	363.65	944340	0.9	288.65	169650	0.3	270000	0.545562	-0.545458
DP 82	363.65	944340	0.9	288.65	169650	0.3	290000	0.486437	-0.486454
DP 83	363.65	944340	0.9	288.65	169650	0.3	260000	0.545557	-0.54559
DP 72	363.65	890643	0.95	288.65	169650	0.3	253894	0.58663	-0.586675
DP 78	363.65	890640	0.95	288.65	169650	0.3	260000	0.586633	-0.586858
DP 79	363.65	890640	0.95	288.65	169650	0.3	270000	0.584898	-0.585006
DP 80	363.65	890640	0.95	288.65	169650	0.3	290000	0.29984	-0.299713

APPENDIX E. P&ID of Thermo-vacuum Drying System (Wilson Engineering Technologies Inc.)

