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Investigation and improvement of ejector refrigeration system using computational fluid dynamics technique

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Abstract

Ejector refrigeration systems are usually designed to utilize low grade energy for driving the cycle. They also have low maintenance cost because they operate without a compressor. Mainly, the ejector performance directly affects the refrigerating performance. Therefore, an investigation on the characteristics and an efficient design of the ejector are important to improve ejector refrigeration systems. In this study, the computational fluid dynamics (CFD) code, FLUENT, is employed to predict the flow phenomena and performance of CPM and CMA steam ejectors.

The ejector refrigeration system, using water as the working fluid, is operated at 120–140 °C boiler temperature and 5–15 °C evaporator temperature. CFD can predict ejector performance very well and reveal the effect of operating conditions on an effective area that is directly related to its performance. Besides, it is found that the flow pattern does not depend much on the suction zone because the results of axisymmetric and 3D simulation are similar. This investigation aids the understanding of ejector characteristics and provides information for designing the ejector to suit the optimum condition.
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Keywords: Ejector; Ejector refrigeration; Computational fluid dynamics (CFD)

1. Introduction

The ejector refrigeration system was firstly developed by Maurice Leblanc in 1910 [1]. This refrigeration system utilized low grade thermal energy or waste heat instead of using electricity. The main advantage of this system is its having fewer moving parts (no compressor). It is, therefore, very low in wear and significantly durable. It is also suitable to operate using water as a refrigerant. However, it usually has a very low coefficient of performance (COP), and this becomes the critical issue and disadvantage of this system.

Fig. 1 shows the operating cycle of the ejector refrigeration system. Comparing to the typical refrigeration cycle or vapor compression cycle, it can be seen that the ejector, the boiler (or steam generator) and the circulating pump are used to replace the compressor. The high pressure refrigerant, boiled in the boiler, is the primary gas feeding to the primary nozzle. It then expands through the nozzle throat at supersonic speed and causes a low pressure area where it connects to the evaporator. Therefore, the refrigerant in the evaporator can boil and evaporate easily. The heat absorbed at the evaporator is the refrigerating capacity. The evaporated refrigerant is called the secondary gas. The primary and secondary gases are mixed and flow through the ejector to the condenser. The liquid refrigerant is pumped back to the boiler partly, and some portion is fed through the expansion valve and evaporator to complete the cycle. It can be seen that the refrigeration performance of the system depends much on the performance of the ejector to induce the refrigerant flow rate through the evaporator.

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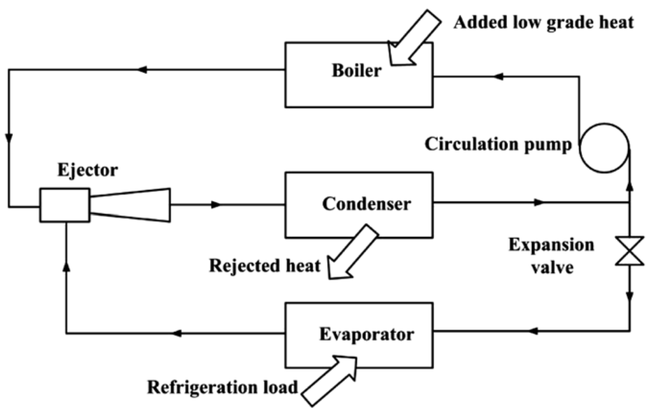

Fig. 1. Operating cycle of ejector refrigeration cycle.

Usually, the two parameters indicating ejector performance are entrainment ratio (Em) and critical back pressure (CBP). Em is defined as shown in Eq. (1), while CBP is the final pressure (condensing pressure) with the ejector working at its maximum capability.

$$\text{Entrainment ratio} \quad \text{Em} = \frac{\text{mass flow rate of secondary flow}}{\text{mass flow rate of primary flow}} \tag{1}$$

Typically, ejectors are categorized in two types based on the mixing concept at the primary nozzle exit. The first one is the constant mixing area (CMA) ejector in which the exit of the primary nozzle is placed at the constant area throat. The second type is the constant pressure mixing (CPM) ejector in which the exit of the primary nozzle is placed at the converging area throat. The setup of both the CMA and CPM ejector are shown in Fig. 2. These two ejectors are suitable to use in different situations. The CMA is capable of drawing more mass flow rate than the CPM, but the CPM is more flexible or suitable to operate in wider condensing pressure ranges.

In the past, the performances of the two ejectors have already been researched in ejector refrigeration systems. However, not much information on the ejector characteristics was reported in detail when there were many parameters involved. Also, they were mainly experimental works and were quite limited in the testing conditions. Therefore, this study aims to use the computational fluid dynamics (CFD) technique to simulate ejector performance in various conditions and to suggest the best possible solutions.

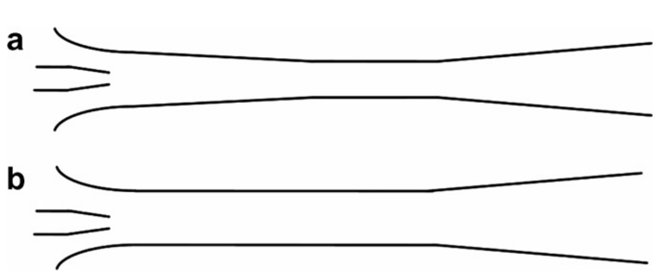

Fig. 2. Two typical ejector types: (a) constant pressure mixing ejector and (b) constant mixing area ejector.

Recently, a new concept of ejector design has been proposed by Eames [2]. It is called the constant rate of momentum change (CRMC) concept for which the ejector is expected to combine the benefits of CMA and CPM ejectors and perform better. In the CRMC concept, a new profile of the diffuser or diverging section of the ejector is proposed. It is claimed that the entrainment ratio of the CRMC is slightly higher than that of the CPM, and the CBP is significantly improved. Garris et al. [3] try to enhance the efficiency of the ejector by reducing the speed of the primary gas by allowing it to expand through a self rotating skew. By rotating the primary gas, the loss during mixing will be lower. From the Garris idea, Chang and Chen [4,5] have developed the petal nozzle and found that the Em and CBP of the ejector can be higher. However, Garris's and Chang's ejectors are quite complicated in structure compared to a typical ejector and are difficult to use in practice.

For many years, researchers have tried to investigate and describe the phenomena of ejector flow in order to develop a high performance ejector. Keenan and his team [6,7] were the first group who proposed an ejector theory. It is a one dimensional ejector flow theory and widely used to predict the properties of the fluid along the ejector axis based on mixing and gas dynamics theory. This theory was widely used, however, it could not describe the constant capacity effect when the exit pressure is decreased. Later, Munday and Bagster [8] successfully adopted the effective area concept (as shown in Fig. 3) within the calculation and showed good agreement with experiment. They studied this phenomenon and described that the primary and secondary fluids do not mix until their flow velocities reach the sonic condition. It is similar to the flow passing through the throat of the convergent–divergent nozzle and choking (of secondary fluid) phenomenon occurs. The flow area where the secondary fluid chokes is called ‘‘effective area’’ (as shown in Fig. 3).

Then, the ejector performance in the refrigeration system can be predicted. However, these concepts are not proved yet due to the limitation of experimentation and measurements, and there are still many factors involving the ejector performance.

Riffat et al. [9] simulated the flow behavior inside the ejector of the refrigeration system using methanol as refrigerant. He found that the CFD results agreed well with the experimental results and can be used to predict other various conditions. Rusly et al. [10,11] investigated the flow

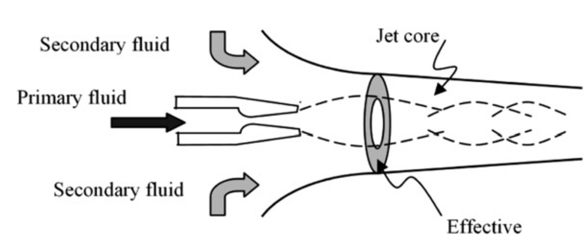

Fig. 3. Effective area occurring in the ejector throat.

characteristics of the ejector in a refrigeration system by using the real gas model in the commercial code, FLUENT, and found good agreement as well.

In this paper, the CFD code, FLUENT, is employed to investigate the flow phenomena and performance of two typical ejectors used in refrigeration systems, which are the CMA and CPM ejector. The results are validated with experiments and simulation in other various conditions. Thus, the most preferable conditions can be applied in system design.

2. CFD for flow simulations

2.1. Computational modeling for ejector flow

The ejector model used in this study is shown in Fig. 4. The model is composed of the primary nozzle and the ejector. The primary nozzle accelerates the high speed gas and induces the secondary flow through the ejector. The ejector consists of four parts, which are the secondary inlet, mixing chamber, throat, and diffuser. The primary nozzle is usually placed at the entrance of the secondary inlet. However, the position of the primary nozzle, called nozzle exit position or NXP, can be varied and affects the ejector performance.

The CFD code used in this study is FLUENT (version 6.0.12). The model ejector is the one used in Chunnanond's study [12]. The ejector geometry is set as axisymmetric. About 48 000 nodes of quadrilateral mesh are used. The dense meshes are preset at the mixing zone along the exit of the primary nozzle as shown in Fig. 4. This is to cope with the high gradient properties around that area. The solving method is couple implicit. The realizable $k-\varepsilon$ turbulence model is selected while the standard near wall function is used in the near wall treatment. Boundary conditions are the pressure inlet and outlet. The energy equation is included, while the fluid property is defined as an ideal gas. In addition, a three dimensional model (3D) is also investigated. This is to determine the effect of the third dimension, compared to the axisymmetric model (ASXM), on the ejector performance at the area around the suction pipe. The hexahedral cell, 5,000,000 nodes, is used in the 3D model as shown in Fig. 5.

2.2. Comparison of ASXM and 3D results

The 3D model of CPM ejector was simulated in order to check whether the suction pipe has any effects on the

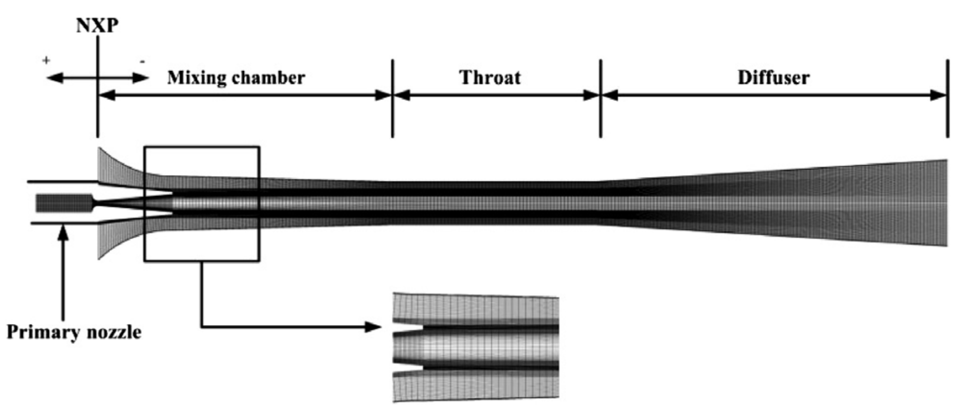

Fig. 4. Ejector geometry (2D) used in the CFD simulation.

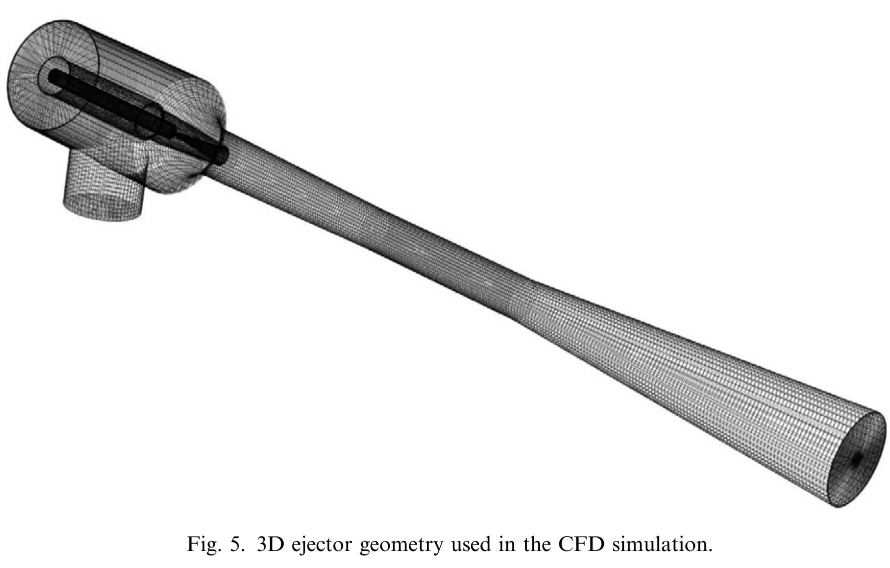

Fig. 5. 3D ejector geometry used in the CFD simulation.

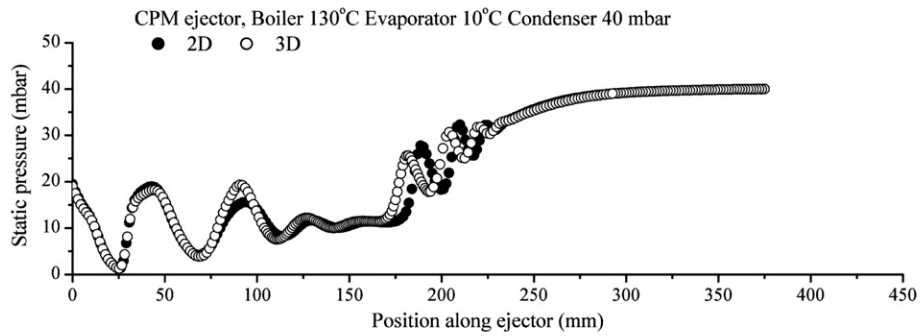

Fig. 6. Comparison of wall static pressure along the ejector from 2D (AXSM) and 3D model.

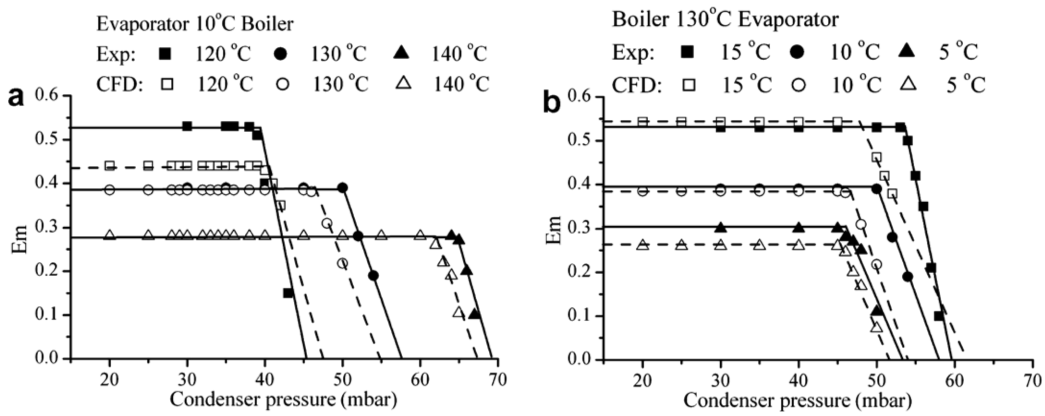

Fig. 7. Validation of the CFD and experimental results (a) at various boiler temperatures and (b) at various evaporator temperatures.

entrainment ratio of the ejector. The results of ASXM and 3D simulations are compared in Fig. 6. It shows very close values of static pressure along the ejector axis. Other properties are also very close. In both models, the pressures gradually increase and slightly fluctuate along the mixing chamber and ejector throat and then smoothly increase in the diffuser. From this comparison, it maybe summarized that the ASXM is good enough to give accurate results, and the 3D model is not necessary for further investigation.

2.3. Validation of CFD simulations

Validation of the CFD results using experimental work has been done in this study and also confirmed with the work of Chunnanond [13]. The effect of the condensing pressure at various boiler temperatures and evaporator temperatures has been investigated. For example, the results are shown in Fig. 7. The results of Em and CBP from CFD are slightly different from the results from the experiments, around 5%.

3. Results

It is well known that the ejector is the key equipment in the ejector refrigeration cycle because it determines the mass flow rate of the refrigerant in the evaporator (i.e. refrigerating capacity) and also the condensing pressure (i.e. heat rejecting capacity). This study, therefore, investigates the effects of various operating conditions and ejector geometries on the Em and CPB of the CPM and CMA ejectors.

3.1. Effect of operating conditions

The change of operating condition certainly affects Em and CBP. Detailed investigations are performed by CFD here. Fig. 8, shows that a higher boiler temperature gives a higher CBP but lower Em in both CPM and CMA ejectors. The evaporator temperature also affects the ejector performance by increasing Em and CBP when the evaporator temperature increases in both ejector types. However, at the same operating conditions, the CMA usually gives a higher mass flow rate or Em, but yields a lower CBP.

3.2. Effect of ejector geometry

3.2.1. Effect of NXP on ejector performance

In this investigation, the ejector performance is determined when the nozzle exit position (NXP) is varied at various operating conditions. The results are shown in Fig. 9. It is found that a higher entrainment ratio can be obtained when the NXP is moved further from the ejector inlet (negative direction). By doing this, the effective area in

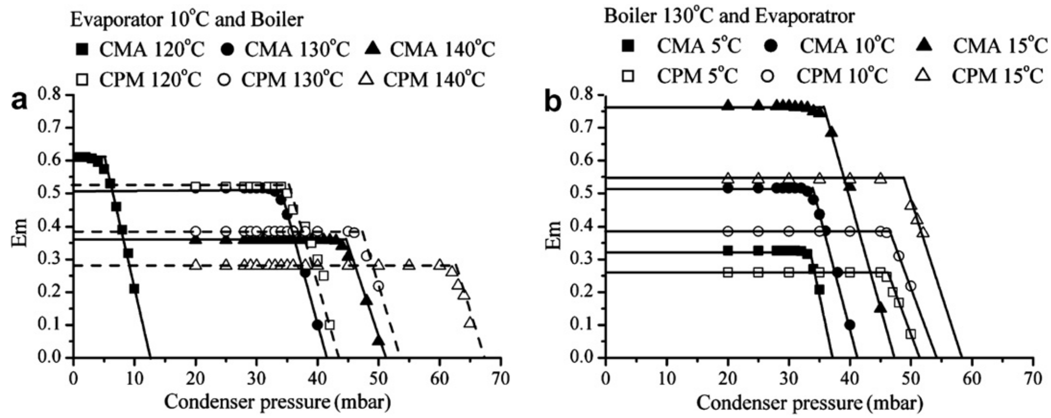

Fig. 8. Effect of operating conditions on ejector performance (a) at various boiler temperature and (b) at various evaporating temperature.

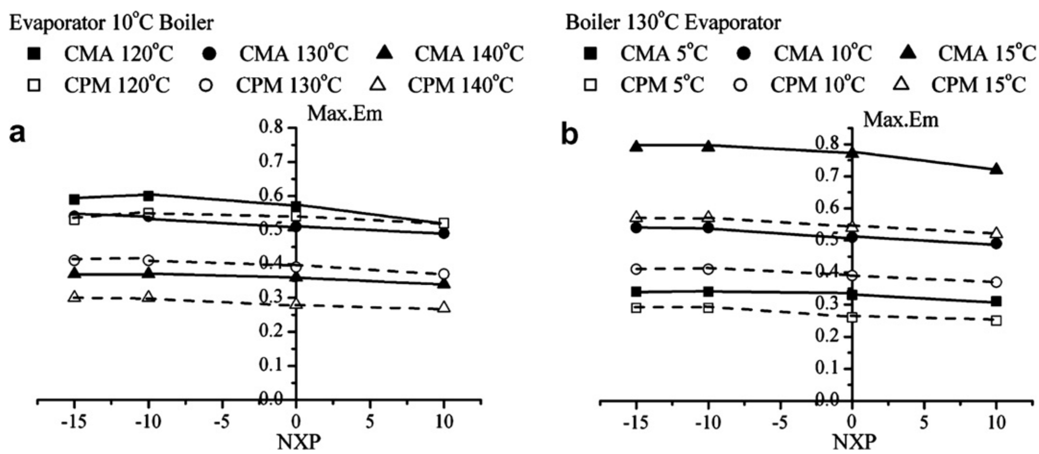

Fig. 9. Effect of NXP on ejector performances (a) at various boiler temperature and (b) at various evaporating temperature.

the ejector throat is getting bigger, and therefore, Em is higher. However, there is only one optimum position. If the NXP is moved too far, the momentum of the primary gas will be lower and cause a lower Em. Therefore, the CFD simulation is very useful in this case to decide the most suitable NXP in the actual system in particular operating conditions.

3.2.2. Effect of throat length on ejector performance

In this study, the throat length (TL) is the variable, while NXP = 0 and the mixing chamber is 125 mm (see Fig. 4). Both the CPM and CMA ejectors are investigated at various operating conditions as before.

Fig. 10 shows the effect of the throat length on the ejector performance at various operating conditions. It reveals

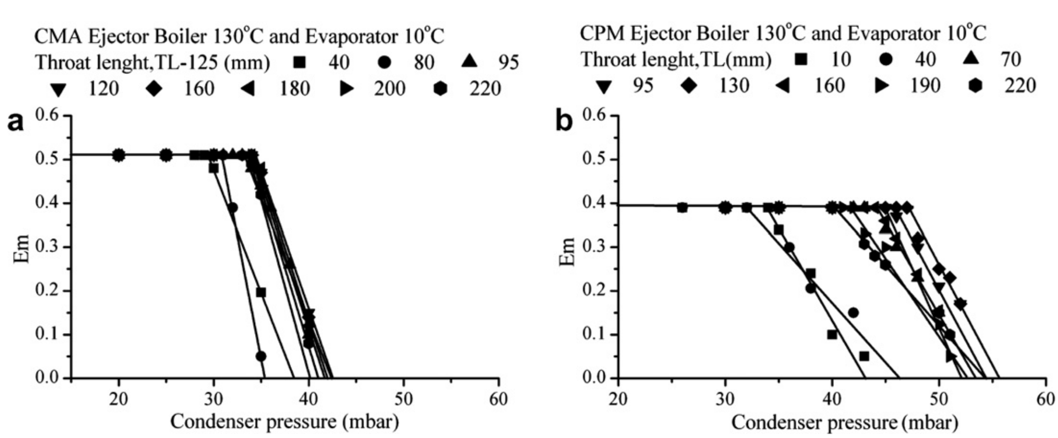

Fig. 10. Effect of throat length on ejector performances (a) CMA ejector and (b) CPM ejector.

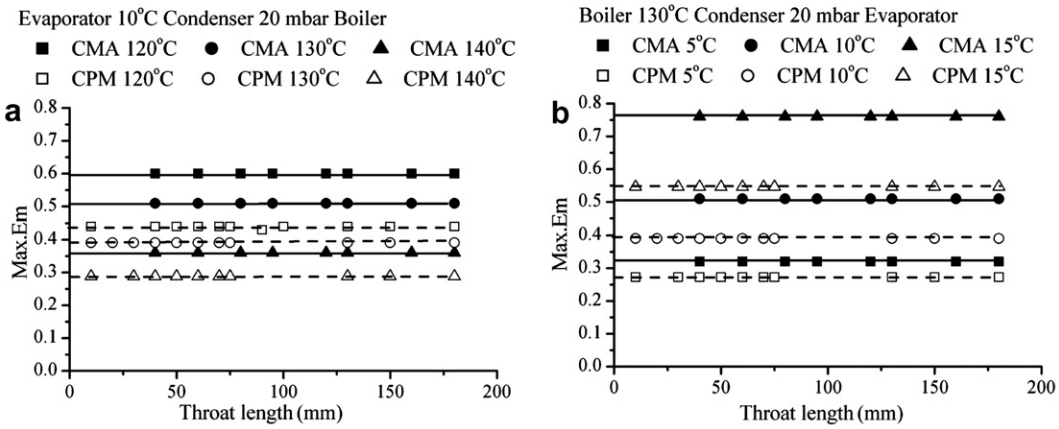

Fig. 11. Effect of throat length on entrainment ratio of an ejector (a) at various boiler temperature and (b) at various evaporator temperature.

that the longer throat ejector gives a higher CBP, but the Em is constant. In Fig. 10a, for the CMA ejector, the optimum TL should be between 95 and 120 cm. In Fig. 10b, for the CPM ejector, the optimum TL is 130 cm. If the TL is too long, the CBP of the ejector will decrease. However, the most suitable throat length also depends on the shape, other dimensions of the ejector and operating conditions. Fig. 11 shows the effect of the TL on the maximum Em at various operating conditions. They confirm that increasing TL does not affect Em. It only affects CBP.

From the above investigations, it can be summarized that the operating conditions and ejector shape or geometry directly affect the ejector performances. In practice, it is difficult to design one ejector to perform well at all conditions and experiment with many ejector geometries. CFD simulation, therefore, is very useful to provide basic understandings of the involved parameters. It also helps to determine the most suitable setup before building the real ejector refrigeration system. Furthermore, CFD simulation can reveal the flow phenomena inside the ejector relating to its performance very well as shown in the next section.

4. Discussions

4.1. Flow phenomena inside an ejector

One benefit of CFD investigation is the numerical visualization. The flow phenomena inside the ejector can be depicted from the post processing and used to support the quantitative results.

Fig. 12 describes the Mach number plot of the flow in the 3D ejector model. The operating conditions are 130 °C boiler temperature, 10 °C evaporator temperature and 40 mbar condensing pressure. From the plot, the flow near the nozzle exit along the center line of the ejector is very high and fluctuates because of the expansion shock waves. The secondary flow velocity at the ejector entrance is very low. However, after it mixes with the primary flow, it gains momentum, and they accelerate together. Then, the flow velocity reduces at the diffuser. At the mixing chamber, the velocity difference of the fluid at the ejector wall and at the primary fluid core is very high. This causes a separate flow layer. The high speed primary flow acts as another wall, so the choking condition of the secondary flow can occur. It can also be noticed that the flow velocity at the suction tube is very low compared to that in the mixing chamber or in the throat. This reduces the effect of the suction tube shape on Em. That is why the ASXM and 3D cases give significantly close results.

4.2. Reverse flow phenomena

In practice, only the on design operating conditions are of interest in the ejector flow. The on design operating condition means the condition that the ejector can still perform at its constant Em while the condensing pressure is decreased. However, understanding the off design phenomena is also helpful. In this study, it is found that there is always a reverse flow when the condensing pressure is increased higher than the critical or choking point of the ejector. In the CFD results, at reverse flow conditions, the secondary flow in the ejector throat re-circulates or even cycles. This makes the flow obstruct and reduce Em eventually as shown in Fig. 13.

Normally, the reverse flow occurs at the diffuser, not in the ejector. The flow velocity is decreased, and its kinetic energy or velocity pressure is converted to static pressure. The reverse flow at the throat or mixing chamber is undesirable and should be avoided. However, in the experiments, it is very difficult to detect or visualize the position of the reverse flow phenomena. Therefore, the CFD visualization is very useful in this case.

4.3. Effect of an effective area on ejector performance

The effective area concept, previously shown in Fig. 3, is clearly described by the CFD visualization. It is a very important phenomenon to indicate the Em capacity. Figs. 14–16 show that the effective area is dependent on the operation conditions. The core jet of the primary flow acts as

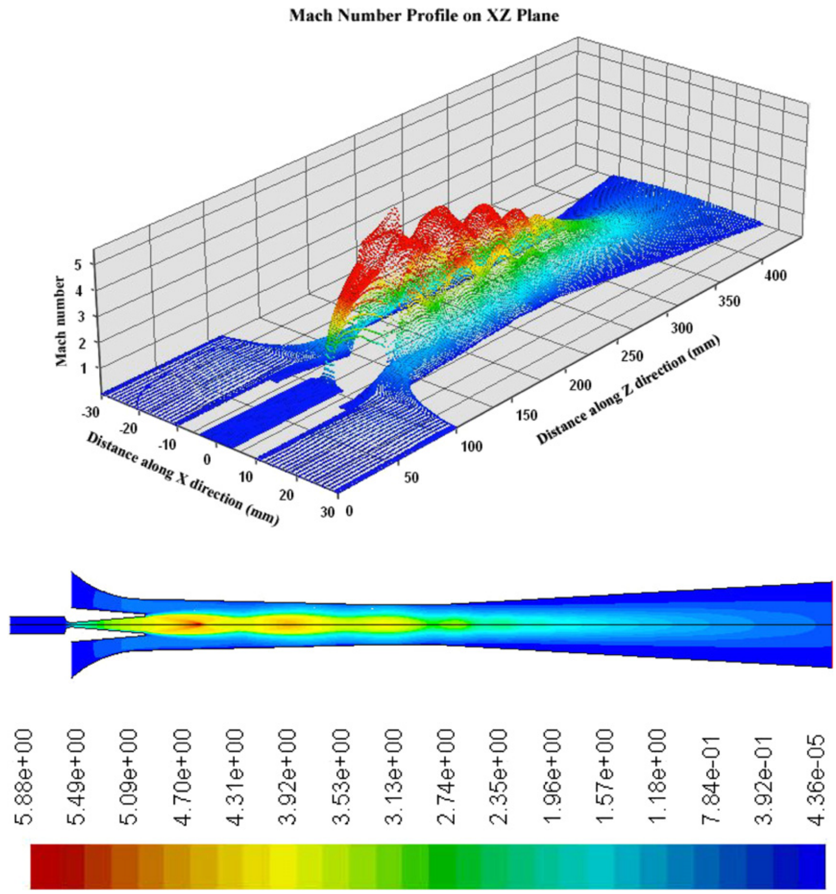

Fig. 12. Mach number profile in the 3D case.

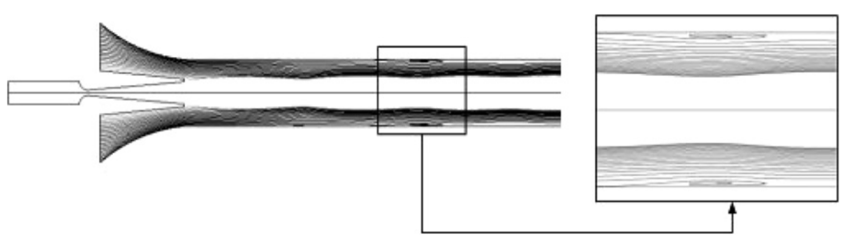

Fig. 13. Path lines in the reverse flow phenomena.

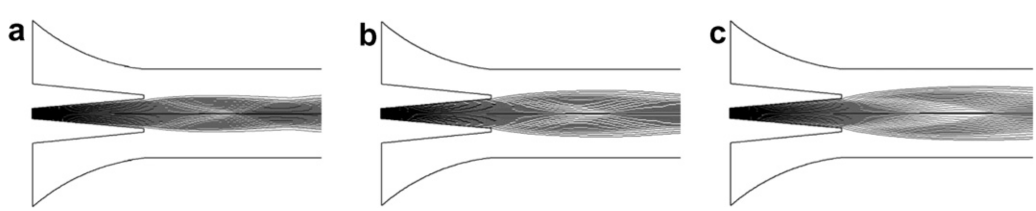

Fig. 14. Sizes of the jet cores at different boiler temperature. (a) 120 °C (b) 130 °C and (c) 140 °C.

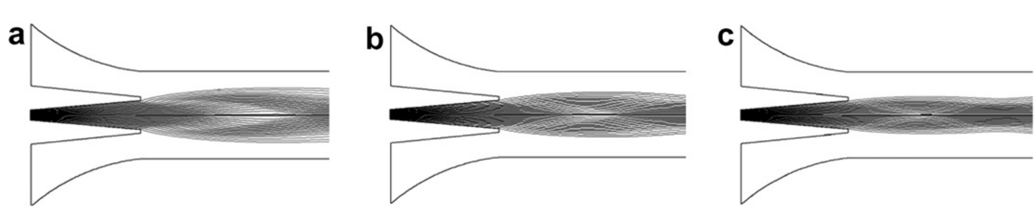

Fig. 15. Sizes of the jet cores at different evaporator temperature (a) 5 °C, (b) 10 °C and (c) 15 °C.

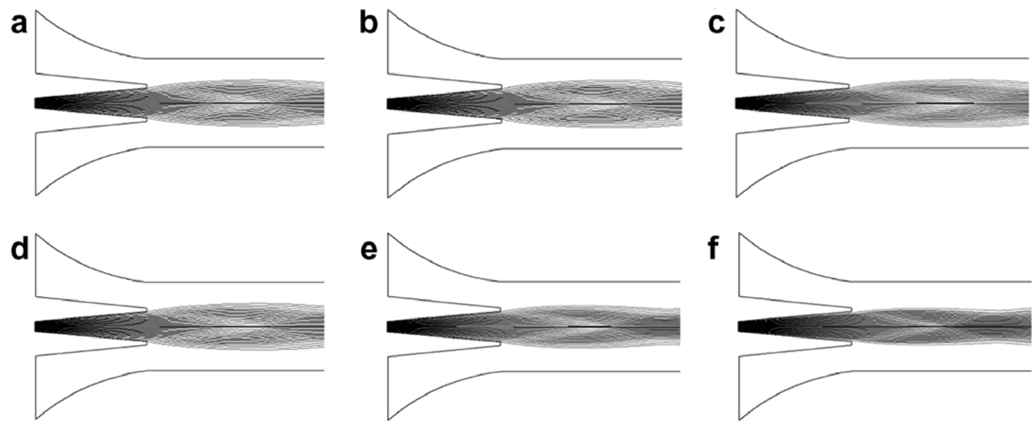

Fig. 16. Sizes of the jet cores at different condenser pressure (a) 20 mbar, (b) 25 mbar, (c) 30 mbar, (d) 35 mbar (CBP), (e) 38 mbar and (f) 40 mbar.

another wall. Therefore, the case that has the bigger effective area has the better Em (120 °C boiler temperature in this case). Fig. 15 shows the effective area when the evaporating temperature is varied. At higher evaporating temperature, the effective area is bigger, thereby giving the higher Em. These results correspond well with the performance investigation in the previous sections.

The ejector back pressure or condensing pressure also affects the ejector performance. If the shape of the jet core is examined carefully, while the condensing pressure is lower than the critical back pressure (CBP), the jet core size (i.e. effective area) does not change. Therefore, Em is constant. However, when the condensing pressure is higher than the critical back pressure (CBP), the jet core is smaller (i.e. bigger effective area), but Em does not increase. This can be summarized that the effective area concept is proved only at the on design operating conditions.

4.4. Effect of mixing process on build up of static pressure

In the mixing process of the primary and secondary fluid, the momentum of the two fluids is exchanged through the flow layer. The efficiency of the mixing can directly affect the regained pressure (build up of static pressure) of the ejector. The two factors involved are the size of the mixing chamber and the mixing period. If the size of the mixing chamber is small, the momentum transfer is quite complete. Therefore, the ejector can give high static pressure such as the CPM ejector. In addition, when the mixing chamber is small, the effective area will be small as well, and the ejector will give low Em. For the mixing period, if there is a long mixing period, the momentum transfer will be quite complete. That is why the reasonably long throat ejector can give higher static pressure or work well at higher CBP.

5. Concluding remarks

This study employs CFD techniques to investigate the flow characteristics of the ejector used in an ejector refrigeration system. The CFD results have been validated with experimental results. The axisymmetric and 3D cases have been compared in order to determine if the shape of the suction tube has affected the ejector performance. The results show very similar solutions because the flow velocity at the suction is very low and is not significant to the overall flow behavior. The effects of various operation conditions on the ejector performance have been investigated. Ejector shapes or geometries are also varied and the ejector performance simulated. The CFD visualization becomes a great benefit in the study because it can reveal the phenomena inside the ejector in detail. In summary, the overall view points on ejector performance related to its flow phenomena can be understood and become very useful tools to design an appropriate ejector for each particular case.

Acknowledgement

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References

[1] Chunnanond K, Aphornratana S. Ejectors: application in refrigeration technology. Renew Sust Eng Rev 2004;8:129–55.
[2] Eames IW. A new prescription for design of supersonic jet pumps: constant rate of momentum change method. Appl Therm Eng 2002;22:121–31.
[3] Hong WJ, Alhussan K, Zhang H, Garris Jr CA. A novel thermally driven rotor – vane/pressure-exchange ejector refrigeration system with environmental benefits and energy efficiency. Energy 2004;29: 2331–45.
[4] Chang YJ, Chen YM. Enhancement of a steam-jet refrigerator using a novel application of the petal nozzle. Exp Therm Fluid Sci 2000;22:203–11.
[5] Chang YJ, Chen YM. Enhancement of a steam-jet refrigerator using a novel application of the petal nozzle. J Chin Inst Eng 2000;23: 677–86.
[6] Keenan JH, Neumann EP. A simple air ejector. J Appl Mech-T ASME 1942;64:75–81.
[7] Keenan JH, Neumann EP, Lustwerk F. An investigation of ejector design by analysis and experiment. J Appl Mech-T ASME 1950;72: 299–309.

[8] Munday JT, Bagster DF. A new theory applied to steam jet refrigeration. Ind Eng Chem Proc DD 1997;16(4):442–9.
[9] Riffat SB, Omer SA. CFD modelling and experimental investigation of an ejector refrigeration system using methanol as the working fluid. Int J Eng Res 2001;25:115–28.
[10] Rusly E, Aye L, Charters WWS, Ooi A, Pianthong K. Ejector CFD modelling with real gas model. In: Proceedings of the 16th annual conference of mechanical engineering network Thailand; 2002. p. 150–5.
[11] Rusly E, Aye L, Charters WWS, Ooi A. CFD analysis of ejector in a combined ejector cooling system. Int J Refrig 2005;28:1092–101.
[12] Chunnanond K, Aphornratana S. An experimental investigation of a stream ejector refrigerator: the analysis of the pressure profile along the ejector. Appl Therm Eng 2004;24:311–22.
[13] Chunnanond K. A study of steam ejector refrigeration cycle, parameters affecting performance of ejector. Ph.D. Thesis of Sirindhorn International Institute of Technology University 1994.