Numerical investigation on the effect of second throat diameter on two-throat nozzle ejector

F T Jia¹, D Z Yang¹, J Xie¹,2,3*

¹College of Food Science and Technology, Shanghai Ocean University, Shanghai 201306, China
²Shanghai Professional Technology Service Platform on Cold Chain Equipment Performance and Energy Saving Evaluation, Shanghai 201306, China
³National Experimental Teaching Demonstration Center for Food Science and Engineering Shanghai 201306, China

E-mail: jxie@shou.edu.cn

Abstract. The employment of two-phase ejectors in the CO₂ refrigeration systems is widely developed recently. Due to the lack reports on the two-throat nozzle ejectors, the performance of CO₂ two-throat nozzle ejector varied with different second throat diameter (D₂) was numerically investigated under different primary pressures (P₁). The accuracy of established numerical simulation model was confirmed with the assistance of experimental data summarized in the literature. The simulated results show that the two-throat nozzle ejector performance corresponding to entrainment ratio (Er) is of better stability with relatively bigger D₂ under different working conditions. Next, the axial static pressure corresponding to bigger D₂ is lower than that of smaller one at pre-mixing chamber. And the secondary flow velocity of bigger D₂ is accelerated better as compared to that of smaller one.

1. Introduction

Ejector is a facility increasing the pressure of low pressure fluid (secondary flow) by the expansion of compressed fluid (primary flow) [1]. There exists the phenomenon of adiabatic expansion and phase change for high-pressure primary flow at converging-diverging section of nozzle, where the pressure decreases and velocity rises of primary flow at nozzle outlet as the function of transformation between pressure energy and kinetic energy. And then primary flow enters the pre-mixing chamber, forming a low pressure area to entrain the secondary flow. As a result, the pressure of hybrid flow is recovered as the kinetic energy is altered into pressure energy at the diffuser [2].

It is perceived that a lot of studies on the Laval nozzle ejectors have been widely developed. In recent years, research on ejectors is no longer limited to the traditional Laval nozzle, there exists preliminary studies implemented on various forms of ejectors. Yang et al. [3] numerically make a comparison on five structures of nozzle under the fixed working parameters. They summarized that the nozzle structure is of great influences on Er and back pressure, and the Er was increased by 9.1% with cross-shaped nozzle relative to standard circular nozzle. Xue et al. [4] made a numerical contrast among conical nozzle, petalage nozzle and crenation nozzle of ejectors, the outcomes revealed that crenation nozzle performed better in Er. Wang et al. [5] numerically developed a two-stage ejector to bring a 36.1% improvement in entrainment ratio and generate evaporating temperature as low as -25℃. Zhou et al. [6] developed a new type of ejector with double nozzle to optimize refrigeration system by
comparing the new cycle with the conventional ejector refrigeration system and the standard refrigeration system with a mathematical model, the consequence showed that the COP of the new cycle was 22.9-50.8% higher than that of the standard refrigeration system, and 10.5-30.8% greater than that of the conventional ejector refrigeration system. Rao et al. [7] developed Tip Ring Supersonic Nozzle and Elliptic Sharp Tipped Shallow Lobed Nozzle for supersonic ejectors to obtain a 30% improvement in $Er$. Wang et al. [8] made a design for auto-tuning AR and auto-tuning NXP ejectors to improve the performance of the ejector. Chen et al. [9] developed a novel ejector with a bypass installed in the low pressure region to optimize the ejector performance. In 2008, the two-throat nozzle ejector with two Laval nozzles in series was initially proposed by Japanese company DENSO. Next, Kang et al. [10] experimentally confirmed the superiority of two-nozzle ejector basing on the comparative test. Nevertheless, the research on new ejectors development has been confirmed promising, there still exists a lot of work to be done to achieve an in-depth study.

In view of the lack of reports on two-throat nozzle ejectors, in this paper performance of two-throat nozzle ejector varied with different second throat diameter ($D_2$) was investigated under different primary pressures ($P_p$) by computational fluid dynamics (CFD) method. The accuracy of ejector simulation mode was confirmed with the aid of experimental data reported in the literature [11]. The discussed results were analyzed to provide guidance for the design of nozzle structures of ejectors.

2. Numerical calculation model

2.1. Physical model

The calculation work on preliminary structural dimension parameters of ejector is based on gas dynamic function method under the design conditions ($P_p$ of 9.3 MPa, $T_p$ of 35 °C, $P_s$ of 4.1 MPa, $T_s$ of 12 °C, $P_c$ of 4.7 MPa). The two-throat nozzle ejector are presented in Figure 1 and Table 1.

![Figure 1. Structural representation of the two-throat nozzle ejector.](image)

| Geometric objects                  | Value/mm | Geometric objects                  | Value/mm |
|-----------------------------------|----------|-----------------------------------|----------|
| Nozzle inlet diameter/$D_p$       | 17       | Expansion chamber outlet diameter/$D_d$ | 28       |
| Suction nozzle diameter/$D_s$     | 34       | Mixing chamber diameter/$D_m$     | 8.5      |
| First throat outlet diameter/$D_1$| 3.6      | Mixing chamber length/$L_m$       | 64       |
| Diameter of the second throat/$D_2$| 3.6     | Length of diffuser/$L_d$           | 72       |

2.2. Simulation settings

FLUENT 19.0 was employed as the solver for the ejector model, and the pressure-based solver was used to discretize the nonlinear control equations. The boundary condition of pressure inlet was chosen for the primary and secondary flow as well as the ejector outlet adapted pressure outlet, and the wall
was no-slip solid wall. The details for simulation settings are recorded in Table 2.

| Table 2. Settlement parameters. |
|--------------------------------|
| Primary flow inlet | Pressure inlet |
| Secondary flow inlet | Pressure inlet |
| Outlet | Pressure outlet |
| Wall | No-slip |
| Solver | Pressure-based |
| Time | Steady |
| Turbulence model | Standard k-ε |
| Near-wall treatment | Standard wall functions |
| Materials | CO₂ |
| Pressure-velocity coupling | SIMPLE |
| Convective terms | First-order upwind |
| Convergence criteria | 1x10⁻⁶ |

2.3. Controlling equations
The simulation calculating method for ejector was established. To simplify the calculation, some assumptions were propose as follows:
1) The expansion process of flow kept isentropic.
2) The ejector inner wall surface was smooth and adiabatic.
3) The flow eventually kept steady.
4) The state of primary and secondary inlet flow was saturated.

On the basis of above assumptions, the conservation equations including mass, momentum and energy utilized to a control cell in the computational domain are listed as follows:

Mass equation:
\[ \frac{\partial \rho}{\partial t} + \text{div}(\rho \mathbf{u}) = 0 \]  (1)

Momentum equation:
\[ \frac{\partial (\rho \mathbf{u})}{\partial t} + \text{div}(\rho \mathbf{u} \mathbf{u}) = \text{div}(\rho \mathbf{u} \mathbf{u}) - \frac{\partial p}{\partial x} + s_u \]  (2)
\[ \frac{\partial (\rho \mathbf{v})}{\partial t} + \text{div}(\rho \mathbf{v} \mathbf{u}) = \text{div}(\rho \mathbf{u} \mathbf{v}) - \frac{\partial p}{\partial y} + s_v \]  (3)
\[ \frac{\partial (\rho \mathbf{w})}{\partial t} + \text{div}(\rho \mathbf{v} \mathbf{u}) = \text{div}(\rho \mathbf{w} \mathbf{u}) - \frac{\partial p}{\partial z} + s_w \]  (4)

Energy equation:
\[ \frac{\partial (\rho T)}{\partial t} + \text{div}(\rho \mathbf{u} T) = \text{div}\left( \frac{k}{c_p} \text{grad}T \right) + s_T \]  (5)

Where \( \rho \) is density, \( P \) is static pressure, \( u \) and \( v \) represent vector velocity, \( S_u, S_v, \) and \( S_w \) represent external body force, \( C_p \) is specific heat, \( T \) is static temperature, \( k \) is heat conductivity, \( S_T \) is viscosity dissipation.

The turbulent flows were restrained by standard k-ε double-equation model, the transfer equations for turbulent kinetic energy \( k \) and dissipation rate \( \varepsilon \) are listed as follows:
\[ \frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left( \mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right) + \mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \rho \varepsilon \]  (6)
\[ \frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho \varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_j} \right) + C_{\varepsilon_k} \frac{\varepsilon}{k} (G_k + C_{\varepsilon G_k} \rho - C_{\varepsilon w} \frac{\varepsilon}{k}) \]  (7)
The grid of the calculation model is divided into five different quantities, and the result was calculated with same structural ejector model and solution. The result of grid independence test is illustrated in Figure 2. From the figure we observe that the $Er$ increase with the enlargement of grid quantities firstly, as the grid quantities are more than 250 thousand, $Er$ keep steady with the increase of grid quantities. Thus, the grid division quantity of the calculation model is confirmed around 250 thousand.

![Figure 2. Variation of $Er$ with the number of grids.](image)

In order to validate the simulation method, the ejector calculation model having the same size with that of literature[11] was established, the inlet and outlet mass flow rates of the ejector were simulated under different $P_s$ and compared with its experimental data. Table 3 presents the outcomes of simulation mass flow rate under different $P_s$. There existed a small error of mass flow rates between the simulated values by the established mathematical model and the experimental values for different $P_s$ conditions, with a maximum error of 1.86% for the primary flow, 10.39% for the secondary flow and 8.9% for the entrainment coefficient. As a whole, the simulation method can predict the ejector internal flow with good accuracy.

| $P_s$/Mpa | $m_p_{exp}$/kg·s$^{-1}$ | $m_p_{CFD}$/kg·s$^{-1}$ | $\delta m_p$/% | $m_o_{exp}$/kg·s$^{-1}$ | $m_o_{CFD}$/kg·s$^{-1}$ | $\delta m_o$/% | $Er_{exp}$ | $Er_{CFD}$ | $\delta Er$/% |
|-----------|-----------------|-----------------|-------------|-----------------|-----------------|-------------|-----------|-----------|-------------|
| 3.45      | 0.043           | 0.0422          | 1.86        | 0.008           | 0.0086          | 6.88        | 0.1860    | 0.2026    | 8.9         |
| 3.65      | 0.043           | 0.0422          | 1.86        | 0.018           | 0.0161          | 10.39       | 0.4186    | 0.3822    | 8.69        |
| 3.76      | 0.043           | 0.0422          | 1.86        | 0.022           | 0.0217          | 1.32        | 0.5116    | 0.5145    | 0.55        |
| 3.85      | 0.043           | 0.0422          | 1.86        | 0.027           | 0.0248          | 8.33        | 0.6279    | 0.5865    | 6.60        |
| 4.3       | 0.043           | 0.0422          | 1.86        | 0.043           | 0.0401          | 6.74        | 1.0000    | 0.9502    | 4.98        |

Table 3. Comparison of simulated $m$ with published in the literature[11] under different $P_s$. 

\[ G_k = \mu_l \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \cdot \frac{\partial u_i}{\partial x_j} \]  
\[ \mu_l = \rho C_{\mu} \frac{k^2}{\varepsilon} \]
4. Discussion and results

In this study, the effects of the $D_t$ on the ejector performance are numerically investigated by varying $D_t$ from 2.8 mm to 4.0 mm under 4 sets of $P_p$ (8.9 MPa, 9.1 MPa, 9.3 MPa, 9.5 MPa). $Er$ reflects the entrainment capacity as a significant indicator for performance evaluation of the ejector. The $Er$ is defined as the ratio of the secondary flow mass flow rate ($m_s$) to the primary flow mass flow rate ($m_p$).

$$Er = \frac{m_s}{m_p}$$  \hspace{1cm} (10)

![Figure 3. Variations of $Er$ with different second throat diameters and primary pressures.](image)

The variations of $Er$ as a function of second throat diameter at different $P_p$ are presented in Figure 3. It is evident from the figure that the $Er$ over the entire range of $P_p$ increases sharply along with the rises of $D_t$ initially, and attains their peak value at the $D_t$ from 3.4 mm to 3.6 mm, after that the $Er$ falls slightly. For the variety of $D_t$ analysed from 2.8 mm to 4.0 mm at $P_p$ of 9.3 MPa, the least $Er$ is recorded for 2.8 mm, the maximum $Er$ of 0.942 is recorded for 3.6 mm. After 3.6 mm, the decrease for $Er$ is not sharp up to 4.0 mm. In addition, the trend of broken line emphasizes that the optimal $D_t$ of each curve representing $Er$ is getting small for a increase in $P_p$. It is also observed that the $Er$ changes dramatically for different $P_p$ when $D_t$ is no more than 3.4 mm, while the $Er$ changes little for the variations of different $P_p$ as the $D_t$ varies from 3.4 mm to 4.0 mm. That is to say, the two-throat nozzle ejector performance corresponding to $Er$ is of better stability with relatively bigger $D_t$ under different working conditions. As a result, it is necessary to choose a suitable $D_t$ for a new two-throat nozzle ejector.

![Figure 4. Distributions of axial static pressure for different second throat diameters at $P_p$ of 9.3 MPa.](image)
The phenomenon of Er variations with Dt can be further explained with the assistance of axial static pressure distribution at Pp of 9.3 MPa illustrated in Figure 4. It is perceived that the variance for axial static pressures occurs from the first nozzle throat, where the axial static pressure goes down in response to the enlargement of Dt between the first and second nozzle throat. However, the relationship between axial static pressure and Dt comes to be the opposite of former. The possible reason behind this would be the delay of transition between pressure energy and kinetic energy for the increase in Dt. Next, an inversion for the relationship between them is observed again from the figure enlarged part at pre-mixing chamber. This implies that the various degrees of shock waves come into being due to the huge difference in velocity between primary and secondary flows at pre-mixing chamber. Obviously, the axial static pressure corresponding to bigger Dt is lower than that of smaller one at pre-mixing chamber, which confirms the trend of Er summarized in Figure 3.

![Figure 5. Fields of velocity for different second throat diameters at Pp of 9.3 MPa.](image)

Figure 5 depicts the partial fields of velocity for simulated ejectores as Dt varies from 2.8 mm to 4.0 mm at Pp of 9.3 MPa. It can be noticed from the figure that velocity decreases for the enlargement in Dt inside the nozzle. The reason to this difference in velocity among the entire ejectors could be the primary flows undergo different levels of expansion and compression in response to the variations of Dt between first nozzle throat and nozzle outlet. Next, a reverse relationship between primary flow velocity and Dt is observed at pre-mixing chamber. Namely, the primary flow velocity at pre-mixing chamber increases as the enlargement in Dt, where the higher velocity leads to the formulation of a relatively lower pressure region to entrain more secondary flow. In addition, it is obvious can be seen that the secondary flow velocity of bigger Dt is accelerated better as compared to that of smaller one and primary flow of bigger Dt maintains high velocity for longer distances at mixing chamber. As a result, the figure reveals the reason why the two-throat nozzle ejector with relatively bigger Dt owns high Er.

5. Conclusions
In this research, the performance of two-throat nozzle ejector varied with different second nozzle
throat diameters are numerically investigated under different operating conditions. The simulated method verified by experimental data from relevant literature possessed satisfactory accuracy. Based on above simulated results, it can be drawn the conclusions that:

1. The two-throat nozzle ejector performance corresponding to Er is of better stability with relatively bigger Dt under different working conditions.
2. The axial static pressure corresponding to bigger Dt is lower than that of smaller one at pre-mixing chamber.
3. The secondary flow velocity of bigger Dt is accelerated better as compared to that of smaller one.

Acknowledgement
This research was supported by Science and Technology Innovation Action Plan of Shanghai Science and Technology Commission (19DZ1207503), Public Service Platform Project of Shanghai Science and Technology Commission (20DZ2292200).

Nomenclature

| Symbol | Definition |
|--------|------------|
| P      | Static pressure, MPa |
| T      | Static temperature, °C |
| m      | Mass flow rate, kg s⁻¹ |
| D      | Diameter, mm |
| L      | Length, mm |
| δ      | Error, % |
| ρ      | Density, kg m⁻³ |
| u, v   | Vector velocity, m s⁻¹ |
| Sₚ, Sₛ, Sₜ | External body force, N m⁻³ |
| Cₚ     | Specific heat, J kg⁻¹ K⁻¹ |
| k      | Heat conductivity, W m⁻¹ K⁻¹ |

Abbreviations

- CFD: Computational fluid dynamics
- Er: Entrainment ratio

Subscripts

- p: Primary flow
- s: Secondary flow
- c: Back flow
- t: Second throat

References

[1] Elhub B, Mat S, Sopian K, Elbreki AM, Raslan MH, Ammar AA 2018 Performance evaluation and parametric studies on variable nozzle ejector using R134A Case Stud Therm Eng 12 258-70
[2] Haida M, Smolka J, Hafner A, Palacz M, Ostrowski Z, Bodys J, et al. 2020 Performance operation of liquid ejectors for a R744 integrated multi-ejector supermarket system using a hybrid ROM International Journal of Refrigeration 110 58-74
[3] Yang X, Long X, Yao X 2012 Numerical investigation on the mixing process in a steam ejector with different nozzle structures International Journal of Thermal Sciences 56 95-106
[4] Xue K, Li K, Chen W, Chong D, Yan J 2017 Numerical Investigation on the Performance of Different Primary Nozzle Structures in the Supersonic Ejector Energy Procedia 105 4997-5004
[5] Wang X, Wang L, Song Y, Deng J, Zhan Y 2021 Optimal design of two-stage ejector for subzero refrigeration system on fishing vessel Applied Thermal Engineering 187 116565
[6] Zhou M, Wang X, Yu J 2013 Theoretical study on a novel dual-nozzle ejector enhanced
refrigeration cycle for household refrigerator-freezers *Energy Conversion and Management* 73 278-84

[7] Rao S M V, Jagadeesh G 2014 Novel supersonic nozzles for mixing enhancement in supersonic ejectors *Applied Thermal Engineering* 71 62-71

[8] Wang L, Liu J, Zou T, Du J, Jia F 2018 Auto-tuning ejector for refrigeration system *Energy* 161 536-43

[9] Chen W, Chen H, Shi C, Xue K, Chong D, Yan J 2016 A novel ejector with a bypass to enhance the performance *Applied Thermal Engineering* 93 939-46

[10] Huang K, Guo X M, Zhang P L. Influence of Structural Parameters of Two-Throat Nozzle Ejector on Performance of Two-phase Flow Ejector Refrigeration System. In: Yan J, Sun F, Chou SK, Desideri U, Li H, Campana P, et al., editors. 8th International Conference on Applied Energy. Amsterdam: Elsevier Science Bv; 2017. p. 5091-7

[11] Guangming C, Xiaoxiao X, Shuang L, Lixia L, Liming T 2010 An experimental and theoretical study of a CO2 ejector *International Journal of Refrigeration* 33 915-21