Numerical Analysis of Hybrid Heat Driven Ejector System Based on the Ejector Performance Map Approach

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ABSTRACT

Recently, the utilization of low-grade thermal energy has gained increased attention as an attractive opportunity to save energy. Heat driven ejector refrigeration systems are promising solutions for utilizing thermal energy from waste heat. However, the main drawbacks of this system are low efficiency under high ambient temperature conditions and difficult controllability. The hybrid ejector system is a promising solution for overcoming these drawbacks. In this system, a booster compressor is installed in order to improve efficiency under a wide range of conditions. In addition, three valves are used to switch between the three operation modes: ejector mode, hybrid mode, and compression mode. In this study, a numerical analysis of the hybrid ejector system is performed and compared with experimental results. The investigations focus on chilled water supply conditions (evaporation temperature is 5 °C) under various condensation temperature conditions ranging from 15 °C to 35 °C. Low-grade waste heat at 70 °C is used to drive the ejector. R290 (propane), which is a promising natural refrigerant, is used in the system. The Newton-Raphson method implemented in MATLAB is used for iterative calculations. Regarding the ejector model, a novel ejector map approach is proposed in order to make an empirical ejector model. In this modeling, two pressure ratios are used to calculate the ejector entrainment ratio. The numerical analysis results and ejector modeling results show good agreement with the experimental results and these methods are validated.

Keywords:

heat driven ejector system, hybrid ejector system, waste heat utilization, ejector performance map, numerical simulation.

1. INTRODUCTION

With the increasing need for low-carbon technologies, the utilization of low-grade thermal energy is considered an attractive opportunity for saving energy. The heat driven ejector refrigeration system is a promising solution for utilizing thermal energy from waste heat, which is a free energy source in many fields (*e.g.* from industrial processes or solar heat). In this system, an ejector driven by thermal energy and a refrigerant pump are used instead of a mechanical compressor. Since a pump consumes less energy than a compressor, the heat driven ejector system requires much lower electric energy than the conventional vapor compression system. The main applications of this system are seen in cooling of industrial and commercial buildings, such as chilled water supply, air conditioning, and cooling processes.

However, the main drawbacks of this system are the limited range of operation conditions and difficult controllability. Especially under high condensation temperature conditions such as in summer, the performance of the ejector drops dramatically, because the pressure lift (pressure difference between evaporator and condenser) is higher. Therefore, this system is not applicable in summer. Although this system shows high performance under low temperature conditions such as in winter, spring or fall, these drawbacks are severe obstacles that currently prevent widespread application of this technology.

The hybrid ejector refrigeration system is a promising solution for solving these drawbacks. In this system, a booster compressor is installed in order to improve efficiency under a wide range of conditions and to substantially improve controllability. Because of their potential, hybrid ejector systems are receiving increased attention these days. Wang *et al.* (2016) performed an experimental system comparison with a conventional vapor compression system for an air-conditioning application using R134a and found that performance was 34% higher. Yoshida *et al.* (2022) studied its seasonal performance in an experiment and revealed that the hybrid system shows an annual performance that is 66% higher than a vapor compression system. However, the number of studies is still very limited. In this study, a numerical analysis of the hybrid ejector system based on novel ejector modeling is performed and compared with experimental results.

2. HYBRID EJECTOR REFRIGERATION SYSTEM

2.1 System configuration

The hybrid driven ejector refrigeration system described by Wang *et al.* (2016) is shown in Figure 1. A booster compressor is installed at the suction of the ejector. Three valves are installed around the ejector and compressor to change the flow path. Figure 2 (a)-(c) shows the three possible operation modes for this system. The first mode is vapor compression mode, which is a conventional vapor compression system. The second mode is ejector mode, which is a conventional heat driven ejector system and it is used under low condensation temperature conditions. The third mode is hybrid mode, which is a combination of the compression and ejector systems. One of the biggest advantages of the hybrid ejector refrigeration system is that the appropriate mode for maximizing the performance under various conditions can be selected.

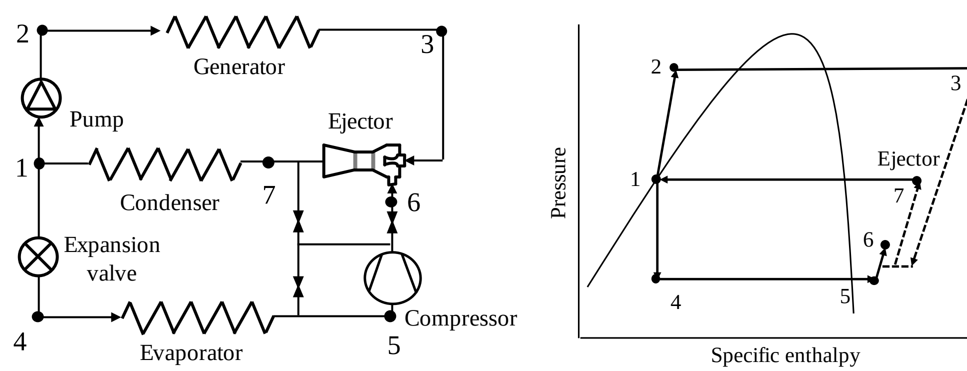

Figure 1: Hybrid ejector refrigeration system

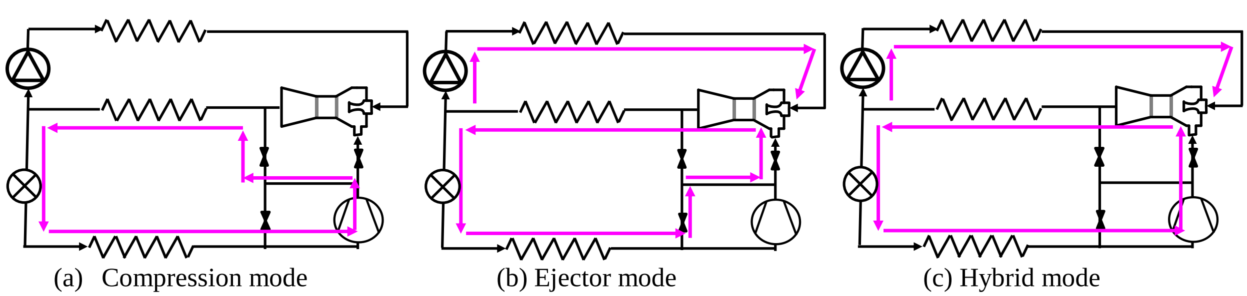

Figure 2: Three operation modes of the hybrid ejector refrigeration system

3. EJECTOR MODELING

3.1 Novel ejector performance map approach

Ejector modeling is one of the most important factors in simulation and design, and a number of researchers have been working on ejector modeling. He *et al*., (2009) summarized the progress of ejector modeling and showed several mathematical models by previous authors. Of those models, Huang *et al*.'s (1999) model has been widely used and cited in many previous works. Figure 3 shows the principle schematic of the heat driven ejector proposed by Huang *et al*. In this model, several critical phenomena are assumed such as double choking (both motive flow and suction flow are choked), and a normal shock wave. Yoshida *et al.* (2021b) applied this model to the calculation of the hybrid ejector system, and revealed promising results for the hybrid ejector system.

However, these assumptions are applicable under limited conditions and model accuracy is insufficient for use. For example, double choking doesn't occur under high pressure lift conditions, and an oblique shock wave is usually observed instead of a normal shock wave. Figure 4 shows the validation results of Huang's model under a wide range of operation conditions. Huang's model clearly doesn't agree with the experimental results under single choking conditions, and doesn't agree even under double choking conditions (within ±30%). This result demonstrates the difficulty of creating an accurate ejector model.

Therefore, empirical modeling is one of the practical ways for predicting ejector behavior accurately. However, an ejector has five-parameters (P_g, T_g, P_s, T_s, P_c) for identifying the working condition and it is a time-consuming experiment to create a model. In order to make both a simple and accurate empirical model, the authors propose a new ejector performance map approach that uses non-dimensional parameters. In this method, two non-dimensional pressures P_g/P_c, P_c/P_s are chosen as potential parameters to characterize the ejector mass flow entrainment ratio m_s/m_g (Equation (1)).

$$\frac{m_s}{m_g} = f\left(\frac{P_g}{P_c}, \frac{P_c}{P_s}\right) \tag{1}$$

The physical meaning of P_g/P_c is motive flow expansion work, and P_c/P_s is suction flow compression work. m_s/m_g is the non-dimensional mass flow rate. In this method, temperature effects are assumed to be negligible. Temperature difference causes a density difference in the flow, but under practical operation conditions, this temperature difference is not significant and the effect is much lower than the pressure effect (*e.g.* for R290, density is 11.5 kg/m^3 at 550 kPa, 12 °C, and 10.5 kg/m^3 at 32 °C). This method is similar to the water pump performance map which is summarized by pressure lift and flow rate. An ejector has both an expansion function and compression function, and the performance map will be 3-dimentional. This point is also beneficial for researchers or engineers to understand the ejector characteristics visually.

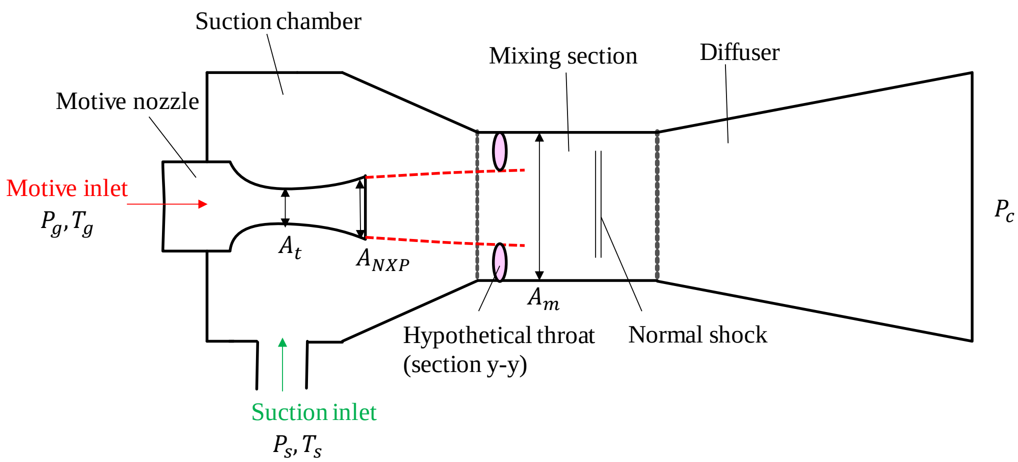

Figure 3: Principle schematic of heat driven ejector

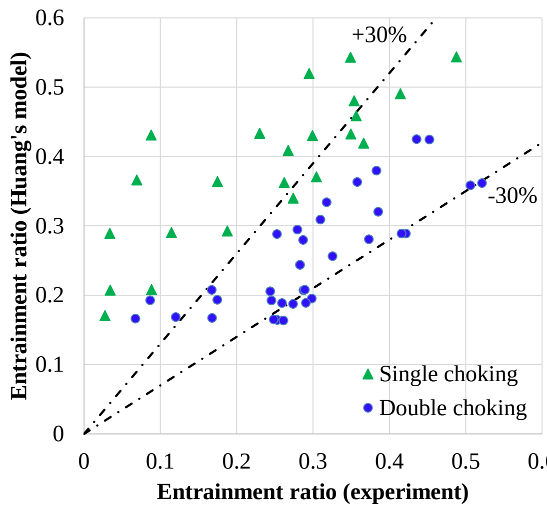

Figure 4: Validation results of Huang's conventional ejector model

3.2 Validation conditions

In order to validate this performance map method, experimental evaluation was conducted. Figure 5 shows the ejector geometry. The ejector is designed based on the one-dimensional ejector model described by Yoshida et al. (2021a). Based on this model, the nozzle throat diameter is designed at 1.48 mm, and the mixing section diameter is designed at 1.96 mm. Table 1 shows the experimental conditions for validation. A wide range of conditions assuming a chilled water supply are tested and 115 data are obtained. Since this performance method is a non-dimensional approach, "theoretically", it is possible to apply it to the different refrigerants. Therefore, three refrigerants (R290, R134a, R1234yf) are also evaluated in this section. Based on these experimental data, an ejector performance map numerical equation is obtained. A third dimensional polynomial function is applied and each coefficient is fitted by multiple regression analysis (Equation (2)).

$$\frac{m_s}{m_g} = a_{00} + a_{10}\left(\frac{P_g}{P_c}\right) + a_{01}\left(\frac{P_c}{P_s}\right) + a_{20}\left(\frac{P_g}{P_c}\right)^2 + a_{11}\left(\frac{P_g}{P_c}\right)\left(\frac{P_c}{P_s}\right) + a_{02}\left(\frac{P_c}{P_s}\right)^2 + \cdots + a_{03}\left(\frac{P_c}{P_s}\right)^3 \tag{2}$$

Table 1: Experimental conditions for validation of ejector performance map method

Evaporation temperature T_e	5-15 ℃
Condensation temperature T_c	15-35 ℃
Generation temperature T_g	45-65 ℃
Evaporator inlet water temperature $T_{w,e,in}$	12-20 ℃
Waste heat temperature $T_{w,g,in}$	60-70 ℃
Refrigerant	R290, R134a, R1234yf

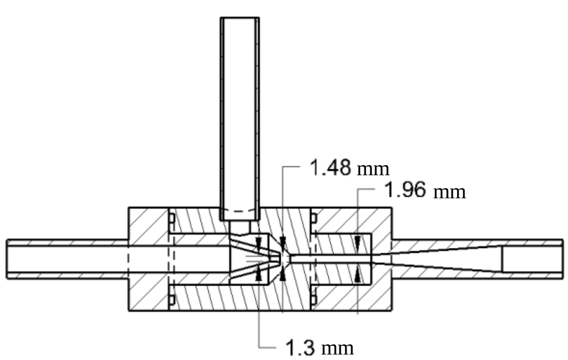

Figure 5: Ejector geometry

3.3 Validation results

Figure 6 shows the ejector performance map based on this method and a smooth surface is observed. The comparison results between the ejector performance map approach and the experiment are described in Figure 7. The performance map approach agrees with the experimental results well (most data within ±10%), and this method is validated for a wide range of operations and for different refrigerants. These results show the effectiveness of the proposed performance map approach for creating a simple and accurate ejector empirical model.

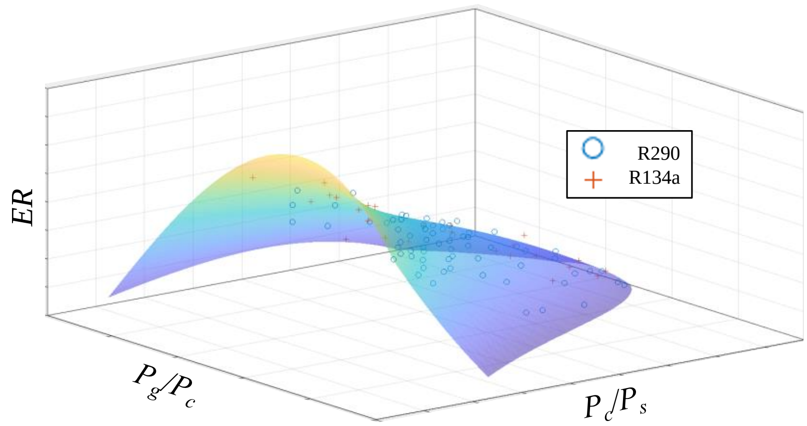

Figure 6: Ejector performance map

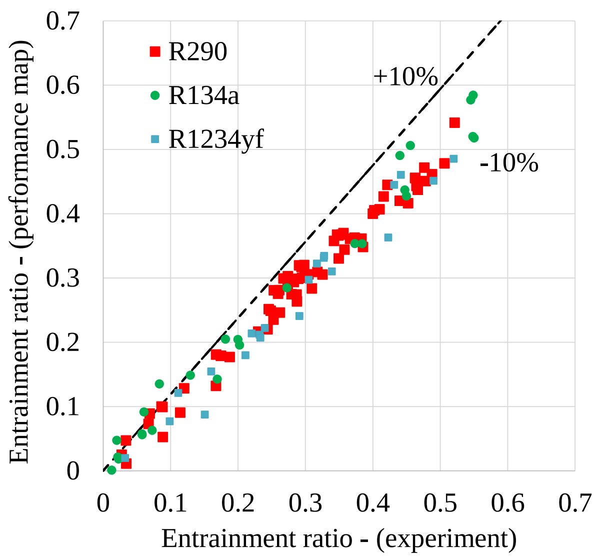

Figure 7: Validation results of the ejector performance map

4. NUMERICAL ANALYSIS

4.1 Compressor, pump, heat exchangers models

In this calculation, the compressor is assumed to be a rotary compressor (positive displacement type). In order to calculate the system performance accurately, compressor isentropic efficiency is critical because it changes dramatically under different pressure ratio conditions. Therefore empirical values obtained by Yoshida *et al.* (2022) are used for isentropic efficiency. For the pump, a diaphragm pump (positive displacement type) is used and isentropic efficiency is assumed to be constant. For heat exchangers, the discretized counter flow model is fabricated. To calculate the two-phase region heat transfer coefficient, Chen's (1966) correlation is used for boiling (generator and evaporator), and Nusselt (1916) theory is used for condensation. Detailed equations are described in Figure 8.

4.2 System calculation

In this calculation, six iterative variables are used and solved by the Newton Raphson parallel method. The simulation is implemented in MATLAB, which is a powerful matrix calculation environment. Figure 8 shows the detailed calculation procedure. First, initial values are set for the six iterative variables. In the system calculation, a six equations array (***F***) is calculated based on iterative variables, input values, each component's equations, and component connection description. The calculation is iterated until the equations errors are small enough to converge.

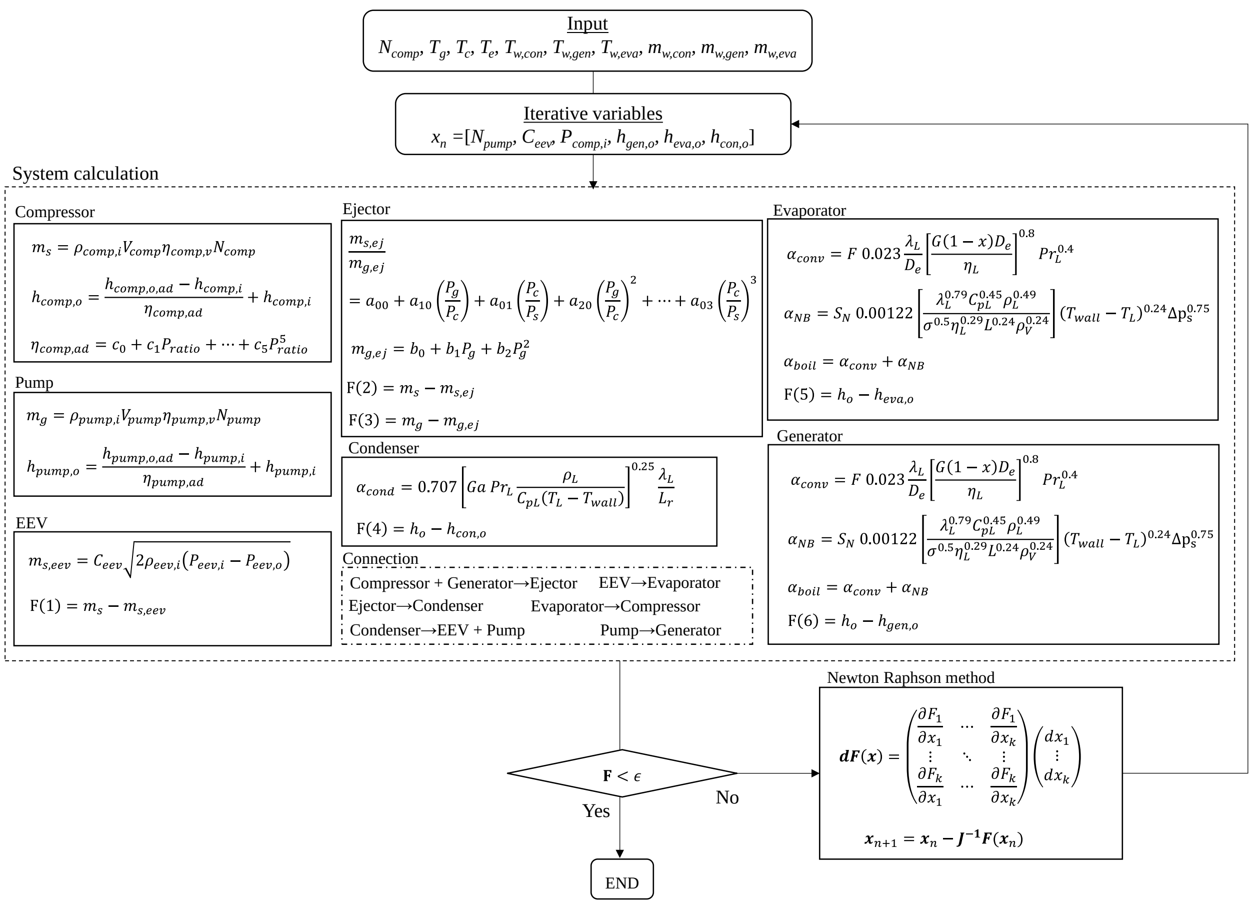

Figure 8: Calculation procedure

4.3 Simulation conditions

Table 2 shows the calculation conditions. The calculation assumes a chilled water supply condition (T_e=5 ℃) at various condensation temperatures T_c ranging from 15 ℃ to 35 ℃ (corresponding to cooling water temperatures of 10 ℃ to 30 ℃). R290 (propane), which is a promising natural refrigerant, is used in the system.

Table 2: Simulation conditions

Evaporation temperature T_e	5 °C (551 kPa)
Condensation temperature T_c	15-35 °C (732-1218 kPa)
Generation temperature T_g	55 °C (1907 kPa)
Evaporator inlet water temperature $T_{w,e,in}$	12 °C
Waste heat temperature $T_{w,g,in}$	70 °C

Since the ejector system is a heat driven system, two COP definitions exist and are used to evaluate system performance. One COP is electric COP (COP_e) which is defined by the ratio of the cooling capacity at the evaporator and the electrical work at the refrigerant pump and compressor. This definition is the same as the conventional vapor compression system. The other COP is thermal COP (COP_{th}) which is defined by the ratio of the cooling capacity at the evaporator and the required heat input at the generator. This COP_{th} has been widely used in heat driven systems such as absorption and adsorption cooling systems.

$$COP_e = \frac{Q_e}{W_{pump} + W_{comp}} \tag{3}$$

$$COP_{th} = \frac{Q_e}{Q_g} \tag{4}$$

4.4 Simulation results

Figure 9 (a), (b) shows a comparison between the simulation results for the three modes and the experimental results obtained by Yoshida *et al.* (2022). In the results, the combination of ejector and hybrid mode shows a significant improvement of COP_e and comparing the conventional vapor compression system the simulation results agree with the experimental results well. For the ejector modes, a 31% maximum difference was observed, but this might be because of the uncertainty of the experiment. In the experiment, the pump electric consumption is calculated based on the small-temperature difference across the pump. This causes a high uncertainty for the COP_e of the ejector mode.

The hybrid system shows a higher COP_{th} than the conventional ejector system and it clarifies the advantage of the hybrid system in terms of thermal efficiency. Regarding the accuracy of the simulation, the simulation results show a 13% maximum difference. This is because of the accuracy of the ejector model and heat exchanger model. Considering the accuracy of the ejector model (±10%), these results are about what can be expected. Heat exchanger models are also a key factor for improving accuracy. In any case, this simulation demonstrates the potential of the hybrid system well, and it will be beneficial to discuss the optimal control and system design.

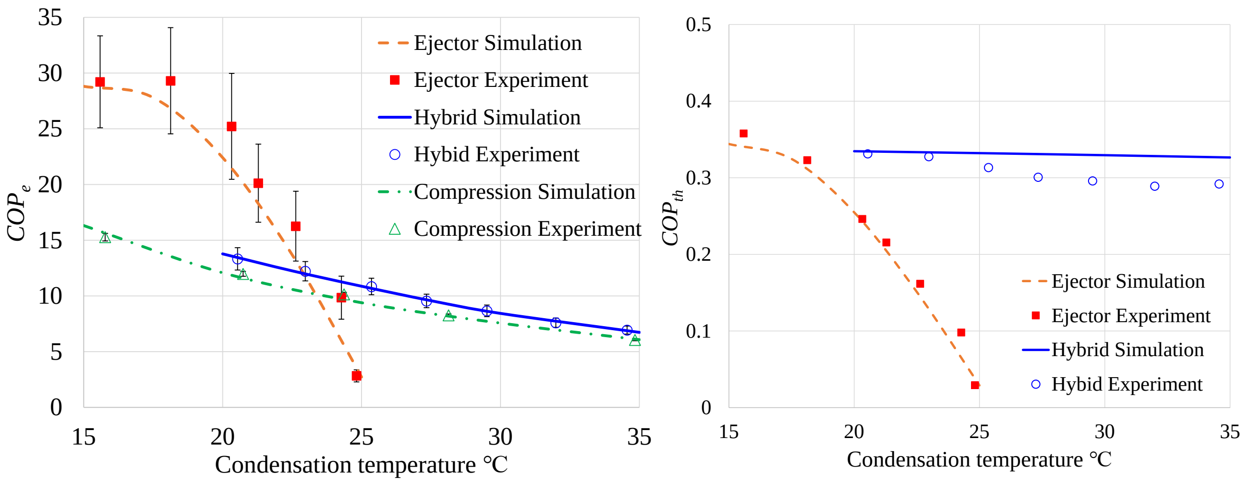

Figure 9: Comparison between the simulation and the experimental results (obtained by Yoshida *et al.* (2022)) (a) electric COP, (b) thermal COP

5. CONCLUSIONS

In this study, a numerical analysis of the hybrid ejector system is performed and compared with experimental results. A novel ejector map approach is proposed in order to make an empirical ejector model. In this modeling, two pressure ratios (P_g/P_c, P_c/P_s) are used to calculate the ejector entrainment ratio. The ejector modeling results show good agreement (±10%) with the experimental results, validating this modeling method. A system simulation is also performed based on the ejector map modeling method. The results show a significant improvement compared to the conventional ejector system, and the simulation results also agreed with the experimental results well. As for work going forward, optimal system control will be discussed based on this system simulation method.

NOMENCLATURE

C	Coefficient of discharge	(-)
C_p	Specific heat	(J/kg/K)
COP_e	Electric coefficient of performance	(-)
COP_{th}	Thermal coefficient of performance	(-)
D	Diameter	(m)
ER	Entrainment ratio (=m_s/m_g)	(-)
Ga	Galilei number	(-)
h	Specific enthalpy	(J/kg)
L	Length	(m)
m	Mass flow rate	(kg/s)
P	Pressure	(Pa)
Pr	Prandtl number	(-)
Q	Heat	(W)
T	Temperature	(°C)
V	Displacement volume	(m^3)
W	Mechanical work	(W)
x	Quality	(-)
α	Heat transfer coefficient	(W/kg/K)
λ	Thermal conductivity	(W/m/K)
η	Efficiency	(-)
ρ	Density	(kg/m^3)
σ	Surface tension	(Pa·s)

Subscript

c	Condenser
$comp$	Compressor
e	Evaporator
g	Generator
i	Inlet
o	Outlet
p	Primary flow
$pump$	Pump
s	Suction flow
sh	Superheat
w	Water

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