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Modeling of Isentropic Coefficients Used in One Dimensional Model to Predict Ejector Performance at Critical Mode

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ABSTRACT

A number of isentropic coefficients are used in the one-dimensional models which predict ejector performance at critical mode. Some of these coefficients have considerable effects on accuracy of the model. These coefficients depend on geometry, working fluid and operating conditions; but, they are usually taken constants or are presented as functions of geometry and working condition based on a specific experiment. In this work, the idea of using the flow parameters to determine these coefficients is introduced and has been analyzed. For this purpose, four models with different formulations are employed. The fluid has been considered as a real gas; hence, the models which are based on the ideal gas assumption are modified. The experimental data related to some ejectors with different geometries, working fluids and working conditions have been used. Using the empirical data, correlations between some of the isentropic coefficients and the flow parameters are developed for some models. Using these correlations, entrainment ratios are calculated with the maximum relative error of 35%, while in most cases the maximum relative error is about 10%. However, errors are acceptable since the empirical data are extracted from a vast range of different geometrical and operational conditions.

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NOMENCLATURE

A	Area of cross-section (m^2)	*exp*	Experimental
A_r	Ejector area ratio (A_3/A_t)	g	Generator
h	Specific enthalpy (J/kg)	*is*	Isentropic process
M	Mach number	m, $2'$	Mixed flow
$\dot{m}$	Mass flow rate (kg/s)	p	Primary flow, Primary nozzle
P_r	Ejector pressure ratio (P_e/P_g)	pA	Primary flow at the entrance of the constant area section
u	Velocity (m/s)	py, $p2$	Primary flow at the hypothetical throat
Greek Symbols		ps	Entrained flow at the location of choking for the entrained flow
η	Isentropic efficiency	$p1$	Primary nozzle exit
ρ	Density	s	Secondary flow
ϕ	Efficiency coefficient	sA	Secondary flow at the entrance of the constant area section
ψ	Expansion coefficient	sy, $s2$	Secondary flow at the hypothetical throat
ω	Entrainment ratio	t	Throat
Subscripts		$2h$	Primary flow at the pressure equal to the secondary inlet pressure
c	Condenser	3	Exit of the constant area section

INTRODUCTION

Nowadays, attentions have been paid to ejector cooling systems. These systems may use low grade waste energy, solar energy or geothermal energy and hence fossil fuel consumption and greenhouse gas emissions are

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decreased. Ejector cooling systems compared to other heat-driven cooling systems have benefits such as using low temperature resources, simple design and operation, high reliability and low cost. On the other hand, due to the low coefficient of performance (COP) of these systems, they are not very popular [1].

Ahmadi and Taheri [2] evaluated the optimum parameters for an ejector-absorption refrigeration system using multi objective optimization. The results showed an increase in thermodynamic and exergetic COPs and decrease in product cost per exergy unit.

Figure 1 shows a schematic of a typical ejector. As the primary flow passes through the primary nozzle, it accelerated to supersonic velocity. At the nozzle exit, a jet is formed, which entrains the secondary flow. The primary and secondary flows mix within the constant area section. Finally, the kinetic energy of this mixed flow is converted into pressure energy in the diffuser. Ejector is the key element in ejector cooling system. Performance of ejector cooling systems is highly dependent on the ejector efficiency which should be evaluated by precise models. Ejector performance is either computed by multidimensional CFD models or calculated by one-dimensional models. CFD models are complicated and time-consuming and require appropriate mesh and turbulence model. In contrast, one-dimensional models are computationally cheap and may quickly determine the ejector performance in a vast range of working conditions. Moreover, one-dimensional models may easily be combined with system models to predict the overall system performance [3].

Conservation equations of mass, momentum and energy are employed in one-dimensional models to evaluate the ejector performance. Some simplifying assumptions such as adiabatic, steady and one-dimensional flow are made. A number of isentropic efficiencies are defined in order to consider friction losses in different sections, including motive and suction nozzles, mixing section and diffuser. Aidoun et al. [4] have reviewed ejector analytical modeling methods.

Liu et al. [5] have recently presented an improved one-dimensional thermodynamic model for performance predicting and design of two-phase ejector. They applied this model to analyze the effects of mixing area ratio, operating conditions and ejector components efficiencies on ejector performance and its geometric parameters. The results showed an increase in entrainment ratio from 0.47 to 1.14 by increasing the mixing area ratio from 1.0 to 1.2.

Haghparast et al. [6] showed that inappropriate choice of isentropic efficiencies to estimate sizing of monophasic ejector components is one of the main source of error in their design. Therefore, they proposed the polytropic efficiencies to be used in one-dimensional ejector models.

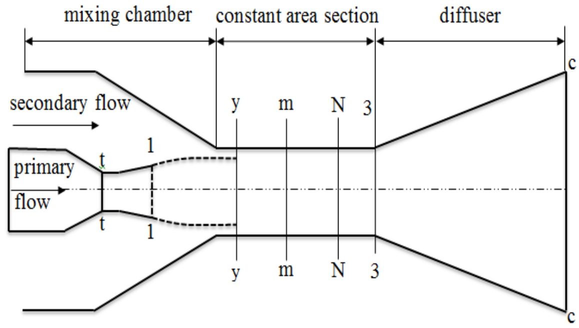

Figure 1. Schematic diagram of ejector

COP of an ejector cycle is highly dependent on the ejector efficiency. In most of the studies, efficiency of each ejector element is either assumed to be a constant or is presented by an equation in terms of only geometrical characteristics or operational conditions. Some of these studies are presented in Table 1.

Liu and Groll [7] determined the efficiencies of ejector components using the two-phase-flow ejector model and experimental data. They showed that for CO_2 refrigeration cycles, the efficiencies of motive nozzle, suction nozzle and mixing section, under different operating conditions change between 0.50 and 0.93, 0.37 and 0.97 and 0.5 and 1.00 respectively. They found an experimental correlation between motive nozzle efficiency, ejector pressure ratio and the ratio of nozzle throat diameter to the constant area section diameter. They also found a correlation between suction nozzle efficiency, ejector pressure ratio and entrainment ratio and another correlation between mixing section efficiency, nozzle throat diameter to constant area section diameter and entrainment ratio.

Kumar and Ooi [8] presented a model in which a normal shock has been fixed at the diffuser entry and the Fanno flow concept has been applied to capture frictional compressible flow within the mixing chamber. In this way, friction coefficient rather than isentropic efficiency has been used in the mixing process. Friction coefficient is calculated using the Schlichting equation after calculation of the averaged Reynolds number in mixing chamber.

Chen et al. [9] used Huang et al. [10] experimental data along with a constant nozzle efficiency of 0.95 and found the mixing and diffuser efficiencies in a way that the error of estimating entrainment ratio and area ratio was minimized. Then based on the obtained mixing and diffuser efficiencies, they presented a correlation for mixing efficiency in terms of pressure ratio, area ratio and entrainment ratio and a correlation for diffuser efficiency in terms of pressure ratio and entrainment ratio.

Table 1. Some of the ejector component efficiencies used in the literature

Reference	Fluid	Efficiencies
Ahmadi and Taheri [2]	Ammonia	$\eta_p=1$, $\eta_s=1$, $\phi_p=1$, $\eta_m=1$
Huang et al. [10]	R141b	$\eta_p=0.95$, $\eta_s=0.85$, $\phi_p=0.8$-0.84 (based on ejector area ratio)
Aly et al. [12]	Steam	$\eta_n=0.90$, $\eta_m=0.95$, $\eta_d=0.9$
Zhu et al. [13]	R141b	$\psi_p=0.95$, $\psi_s=0.85$, $\psi_{exp}=0.8075$
	R11	$\psi_p=0.90$, $\psi_s=0.85$, $\psi_{exp}=0.765$
Zhu and Li [14]	R141b	$\psi_p=0.90$, $\psi_{exp}=0.87+0.6(P_s/P_p)$
	R11	$\psi_p=0.72$, $\psi_{exp}=0.67+0.9(P_s/P_p)$
Cardemil and Colle [15]	R141b	$\eta_t=0.95$, $\eta_m=0.95$, $\eta_d=0.95$, $\psi=0.046/(P_rA_r)+0.764$, $\phi_m=0.9788-0.0073A_r$
	Steam	$\eta_t=0.85$, $\eta_m=0.95$, $\eta_d=0.95$, $\psi=0.0265/(P_rA_r)+0.847$, $\phi_m=0.77$
	CO_2	$\eta_t=0.95$, $\eta_m=0.95$, $\eta_d=0.95$, $\psi=0.374/(P_rA_r)+0.5209$, $\phi_m=1$
Chen et al. [3]	R141b, Air	$\eta_p=0.95$, $\eta_{py}=0.88$, $\eta_s=0.88$, $\psi_m=0.8$-0.84
	Propane	$\eta_p=0.98$, $\eta_{py}=0.95$, $\eta_s=0.85$, $\psi_m=0.84$
Chen et al. [16]	R123	$\eta_n=0.90$, $\eta_d=0.90$, $\eta_m=0.85$
	R141b	$\eta_n=0.95$, $\eta_d=0.84$-0.88, $\eta_m=0.80$-0.84
Chen et al. [17]	R141b	$\eta_p=0.95$, $\eta_{py}=0.80$, $\eta_s=0.85$, $\psi_m=0.95$
	R11	
Li et al. [18]	Air	$\eta_p=0.95$, $\eta_s=0.85$
	R141b	$\phi_p=0.9927-0.1236A_r/P_r-0.0036P_r/A_r$, $\phi_m=0.8574-0.1236A_r/P_r+0.0446\,P_r/A_r$
	R245fa	$\phi_p=0.8091+0.05354P_r/A_r$, $\phi_m=0.8811-0.04322\,A_r/P_r+0.05053\,P_r/A_r$
		$\phi_p=0.6896+0.0414A_r/P_r+0.1064P_r/A_r$, $\phi_m=0.9212-0.0596\,A_r/P_r$
Chen et al. [19]	R141b	$\eta_p=0.95$, $\eta_s=0.90$, $\eta_d=0.81$, $\eta_m=0.975$
	R245fa	
	R134a	
	R600a	

Besagni et al. [11] developed some maps for primary nozzle, suction chamber and mixing efficiencies in a vast range of operating conditions using CFD results and then applied these maps in the lumped parameter model. They showed that ejector component efficiencies are very effective on the model accuracy and they cannot be taken as constants.

Zheng and Deng [20] determined the efficiencies of CO_2 ejector component using an ejector model and experimental data. They utilized the distributed-parameter method for ejector modeling. They established empirical correlations between ejector component efficiencies, ejector pressure ratio, area ratio and entrainment ratio. They showed that the established correlations can be used to predict variable ejector component efficiencies during transient simulation.

It can be seen that in all the previous research, the isentropic coefficients used in one-dimensional models were either considered as constants or were calculated by empirical correlations in terms of operating conditions and ejector geometry only. Furthermore, for each model and each coefficient there is a specific correlation which only fits a specific empirical data. In order to solve this problem and derive generalized correlations, the idea of using flow parameters to determine the isentropic coefficients used in one-dimensional models is presented. For this purpose, empirical correlations were derived for the isentropic coefficients used in different models in terms of the flow parameters and based on the experimental data for different ejectors, working fluids and operating conditions.

MODELS

Four models with different formulations are used in this study: Huang et al. [10], Zhu et al. [13], Cardemil and Colle [15] and Chen et al. [16]. These models can be considered as the advanced forms of the previous models. They have been broadly used in the literature as reference or for comparison [21]. In the models Huang et al. [10], Zhu et al. [12], and Chen et al. ideal gas assumption is used which may be unacceptable depending on the working fluid characteristics. Therefore, these models are used with real gas assumption in this work and the required modifications for each model are presented in the next sections. Thermodynamic properties are extracted from the thermo-physical property library, Cool Prop [22]. The thermodynamic and thermoeconomic analysis was investigated by a system to recognize effective design parameters on thermodynamic COP, exergetic COP and product cost per exergy unit of the whole system. The multi objective optimization model of an ejector-flash tank-absorption refrigeration system fuelled by solar energy was evaluated through a genetic algorithm [23]. To find correlations for the coefficients used in the above-mentioned models, empirical data from the literature have been used, as mentioned below:

1. Huang et al. [10] tested 11 different ejectors with area ratios between 6.44 and 10.64 and presented 39 data series. The working fluid was R141b. They used 2 nozzles with area ratios of 3.271 and 2.905 and 8 mixing chambers with diameters of 6.7 to 9.2 millimeters. In these experiments, vapor temperature in degrees Celsius was between 78 and 95 at nozzle inlet, between 8 and 12 at ejector suction port and between 28 and 36 at condenser.
2. Ablwaifa [23] tested an ejector with an area ratio of 11.46, used R245fa as working fluid and presented 20 data series. In these experiments, vapor temperature in degrees Celsius was between 100 and 120 at nozzle inlet and between 8 and 15 at ejector suction port and between 34 and 46 at condenser. He also tested an ejector with an area ratio of 7.44 using R236fa as working fluid and presented 9 other data series. In these experiments, vapor temperature in degrees Celsius was between 82 and 88 at nozzle inlet and between 4 and 12 at ejector suction port and between 27 and 33 at condenser.
3. Yapici et al. [24] tested 6 ejectors with area ratios of 6.56 to 11.45 using R123 as working fluid. For condenser pressure of 125 kPa and evaporator temperature of 10ºC, they found the optimum vapor generator temperature for each case. They used 3 nozzles with area ratios of 3.11, 3.93 and 4.48 and 2 mixing chambers with diameters of 8.22 and 9.0 millimeters and lengths of 61.65 and 67.5 millimeters and 2 diffusers with diameters of 19.73 and 21.60 millimeters in order to have different combinations.
4. Hakkaki-Fard et al. [25] applied R134a in 3 ejectors with area ratios of 4.17, 5.35 and 8.48 and vapor temperatures of 86, 100 and 106 degrees Celsius at nozzle inlet and vapor temperatures of 30, 25 and 50 degrees Celsius at ejector suction port. They plotted entrainment ratio versus ejector exit pressure for all cases. In this work, three data related to the critical points of these experiments have been used.

EMPIRICAL DATA

Huang et al. model

Huang et al. [10] developed a model to predict ejector performance at critical mode. In this model it is assumed that there is a constant pressure mixing in constant area section and gas flow is at choking condition. If nozzle geometry and working condition are known, entrainment ratio and the required constant area section will be calculated. The aim of this work is to predict ejector performance at different working conditions for a known geometry. Therefore, the experimental values of cross sectional area at constant area section have been employed. Furthermore, the ideal gas assumption is not used. As a result, isentropic coefficients of primary and secondary nozzles and diffuser are defined as below [7, 15]:

$$\eta_p = \frac{h_g - h_t}{h_g - h_{t,is}}, \quad \eta_s = \frac{h_e - h_{sy}}{h_e - h_{sy,is}}, \quad \eta_d = \frac{h_{c,is} - h_3}{h_c - h_3} \tag{1}$$

Real gas behavior assumption makes changes in calculation of the primary and secondary flow mass flow rates and in shock wave calculations. Using this assumption, the primary flow mass flow rate at choked condition, is calculated by the following equations:

$$\dot{m}_p = \rho_t A_t u_t \tag{2}$$

$$u_t = a_t = \sqrt{\left(\frac{\partial p}{\partial \rho}\right)_s} \tag{3}$$

$$\rho_t = \rho\left(P_t, h_t\right) \tag{4}$$

$$h_g = h_t + \frac{u_t^2}{2} \tag{5}$$

$$\eta_p = \frac{h_g - h_t}{h_g - h_{t,is}} \tag{6}$$

Equations (2) to (6) are implicit in terms of P_t and should be solved iteratively. Similar equations are used to calculate the mass flow rate of the secondary flow.

Assuming isentropic expansion, the primary nozzle outlet conditions are calculated by the following equations:

$$h_{p1} = h\left(P_{p1}, s_t\right) \tag{7}$$

$$\rho_{p1} = \rho\left(P_{p1}, s_t\right) \tag{8}$$

$$u_{p1} = \frac{\dot{m}_p}{\rho_{p1} A_{p1}} \tag{9}$$

$$h_g = h_{p1} + \frac{u_{p1}^2}{2} \tag{10}$$

Equations (7) to (10) are implicit in term of P_{pl} and should be solved iteratively.

Due to the constant area between the sections m-m and 3-3 in Figure 1, continuity, momentum and energy equations for a control volume between these two sections are written as follows:

$$\rho_m u_m = \rho_3 u_3 \tag{11}$$

$$\rho_m u_m^2 + P_m = \rho_3 u_3^2 + P_3 \tag{12}$$

$$h_m + \frac{u_m^2}{2} = h_3 + \frac{u_3^2}{2} \tag{13}$$

Using Equations (11) and (12):

$$\frac{P_3 - P_m}{\rho_3 - \rho_m}\left(\frac{\rho_3}{\rho_m}\right) = u_m^2 \tag{14}$$

Equations (11), (13) and (14) are implicit in terms of P_3 and should be solved iteratively.

To assess the variability of the estimations due to changes in the assumed parameters, a sensitivity analysis has been performed. To do this, effects of ±5% changes of the isentropic efficiencies on entrainment ratio (ω) and condenser critical pressure (P_c^*) is invesigated and presented in Figures 2 and 3. The largest sensitivity of entrainment ratio is for the primary flow isentropic efficiency when expanded between the nozzle outlet and hypotetical throat (ϕ_p). For condeser critical pressure, the largest sensitivity is for mixing efficiency (ϕ_m).

Considering the values of η_p=0.95, η_s=0.85 and η_d=0.95 and using the experimental data for entrainment ratio (ω) and condenser critical pressure (P_c^*) isentropic efficiencies ϕ_p and ϕ_m can be determined. As shown in Figure 4, there is a correlation between ϕ_p and the ratio of throat area and ideal area of the primary flow at hypothetical throat (A_3/A'_{py}). The only exception is for the Yapici et al. data in which constant area ejectors are used. For this case, ϕ_p which is calculated by Huang model is mostly more than one. This shows that Huang model is not appropriate for constant area ejectors.

In order to show the importance of consideration of the variation of ϕ_p, the entrainment ratios were calculated and compared for two cases of constant ϕ_p and variable ϕ_p. This has been shown in Figure 5 for Huang et al. data [10] and in Figure 6 for Ablwaifa (R245fa) data [23]. Based on the empirical data and employing regression method, the values of 0.77 and 0.75 were derived for ϕ_p using Huang and Ablwaifa data respectively. As shown in Figures 5 and 6, accuracy of ω is 7% for Huang data and 10% for Ablwaifa data. However, the maximum error when a constant ϕ_p is used is -20% for the former and -30% for the latter.

A correlation between ϕ_m and Mach numbers ratio *Mm/Mpy* is shown in Figure 7. To show the accuracy of the ϕ_m predicted by this correlation, these results are compared with the Huang et al. [10]. The comparison of the entrainment ratios is shown in Figure 8. For these comparisons, similar to the Huang work, a known critical pressure and an unknown ejector area ratio have been

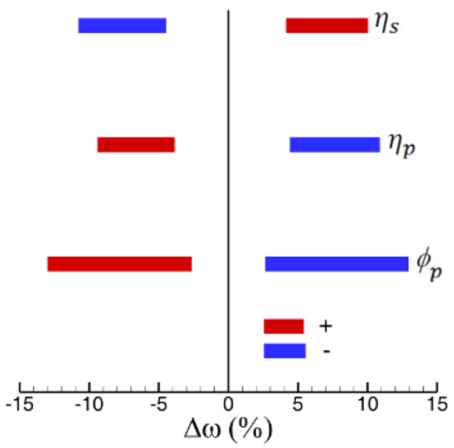

Figure 2. Sensivity of entrainment ratio to ejector components efficiency (Huang model [10]

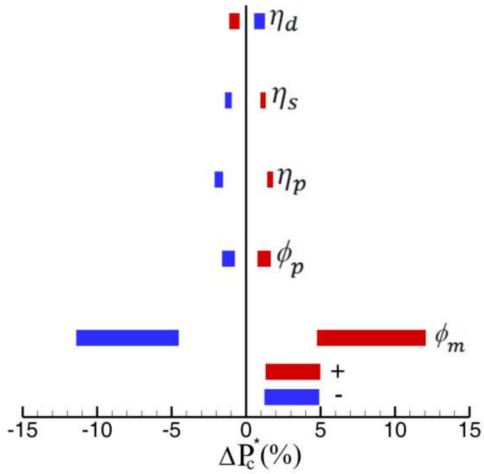

Figure 3. Sensivity of condenser critical pressure to ejector components efficiency (Huang model [10])

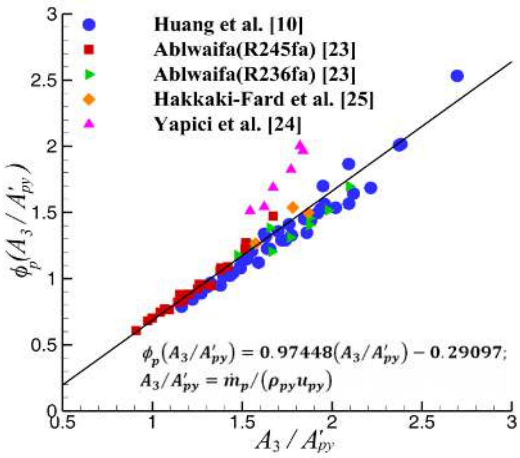

Figure 4. Variation of isentropic efficiency coefficient (ϕ_p) with area ratio (A_3/A'_{py}) (Huang et al. model [10])

used. The error for estimation of the isentropic coefficients by flow parameters is only 8% for entrainment ratios and area ratios. This is while Huang et al. calculated the entrainment ratios and area ratios with 22% and 10% errors respectively when they estimated the mixing coefficient by the ejector area ratio.

Deriving the correlations between isentropic efficiencies and flow parameters and having theejector geometry and working conditions, the entrainment ratio and condenser critical pressure are determined. These two parameters have been drawn against the experimental data in Figures 9 and 10. As shown in Figure 9, the maximum relative error is about 30% which is for Yapici et al. [24] experimental data. This result was predictable, because as mentioned before, Huang model is not appropriate for constant area ejectors. However, for

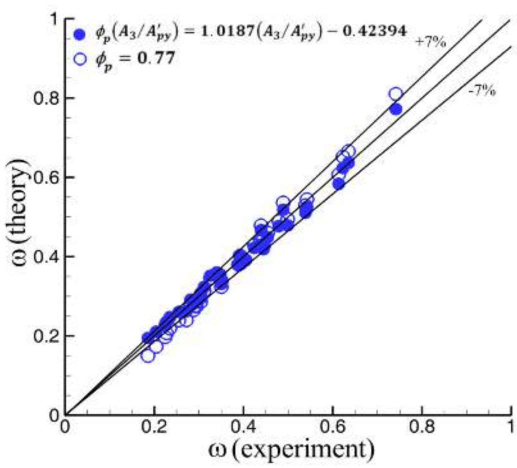

Figure 5. Comparison of the entrainment ratios for constant and variable isentropic coefficients for Huang et al. data [10]

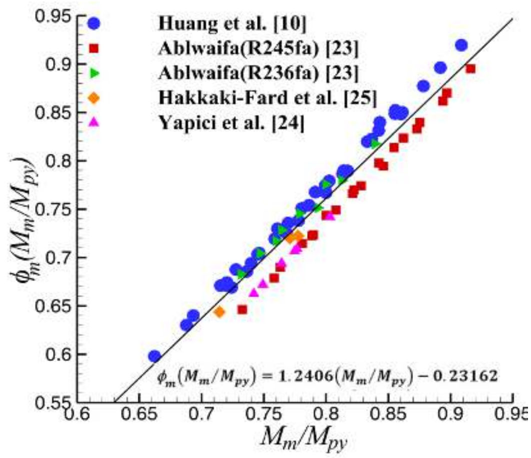

Figure 7. Variation of mixing efficiency (ϕ_m) with Mach numbers ratio (Huang et al. model [10])

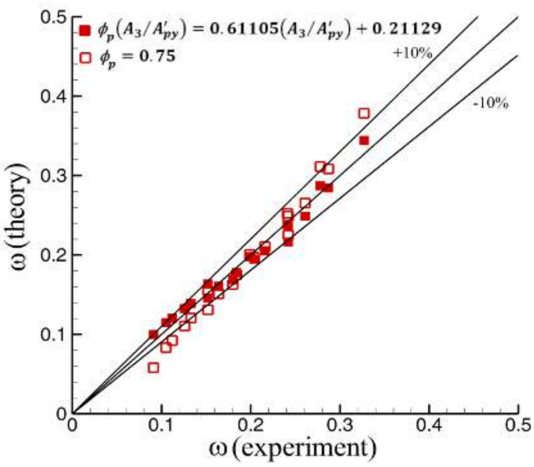

Figure 6. Comparison of the entrainment ratios for constant and variable isentropic coefficients for Ablwaifa (R245fa) data [23]

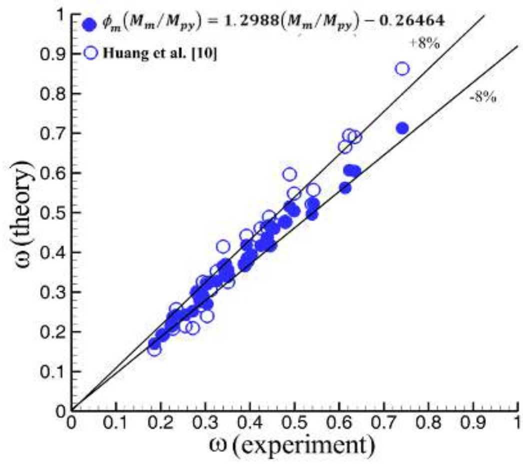

Figure 8. Comparison of the entrainment ratios calculated by the present method with those calculated by the Huang et al. method [10]

all other cases, relative error is less than 10%, which shows the effectiveness of Huang et al. model. For critical pressure, as shown in Figure 10 the maximum error is about 10%.

Zhu et al. model

Zhu et al. [13] proposed the shock circle model to predict ejector performance at critical operating mode. In this model, a shock circle has been assumed at the entrance of the constant area chamber to take into account the non-uniform velocity distribution in radial direction. Assuming that a zero-thickness shock circle separates the two flows and assuming sonic flow at shock circle, an exponential velocity distribution in radial direction is obtained which is an estimate for the near-wall viscous flow. In Zhu et al. model, an ideal gas with constant properties (specific heat ratio) has been assumed as the working fluid. However, in this work, real gas equations have been employed and the isentropic coefficients of the primary and secondary nozzles are similar to Huang et al. model mentioned in the previous section.

The $\pm 5\%$ parameter changes effects on entrainment ratio is investigated and presented in Figure 11. Sensitivity of entrainment ratio to the primary nozzle isentropic efficiency and efficiency of the primary flow expansion in suction chamber (ψ_{exp}) is high, but its sensitivity to the secondary flow isentropic efficiency is negligible.

For isentropic efficiency of the primary flow expansion in suction chamber (ψ_{exp}), Zhu et al. used the product of the two isentropic coefficients η_p and η_s. Considering the values of η_p=0.95, η_s=0.85 and using the experimental data for entrainment ratio (ω) isentropic

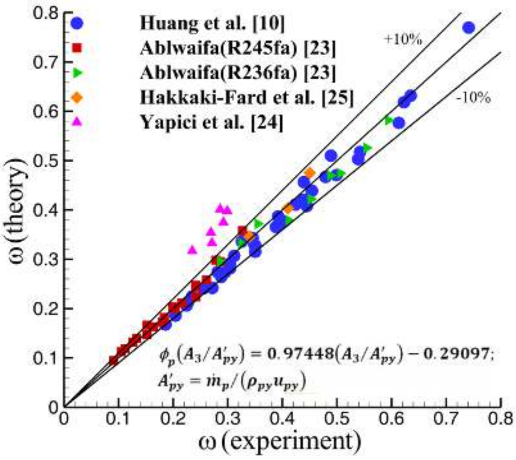

Figure 9. Entrainment ratio found from the present method, compared with the experimental data (Huang et al. model [10])

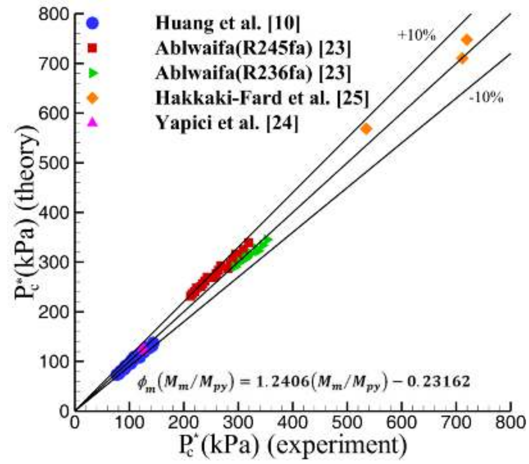

Figure 10. Critical back pressure found using the present method, compared to the experimental data (Huang et al. model [10]).

efficiency of the primary flow expansion in suction chamber (ψ_{exp}) can be determined.

As shown in Figure 12, there is a correlation between ψ_{exp} and the ratio of ideal effective expanded area of primary flow (A'_{pA}) and throat area (A_3). It was found that constant area ejectors used by Yapici et al. behave differently in this model as well. The amount of ψ_{exp} for these ejectors is about 0.5, but for the other ejectors it is about 0.7.

Knowing ψ_{exp} as a function of $\frac{A'_{pA}}{A_3}$ for a known operating condition and geometry, entrainment ratio (ω) can be determined. The calculated entrainment ratios are compared with the experimental data in Figure 13. As seen in this figure, the maximum relative error is about 35% which is related to the experimental data of Yapici et al., but the errors for the rest are less than 16%.

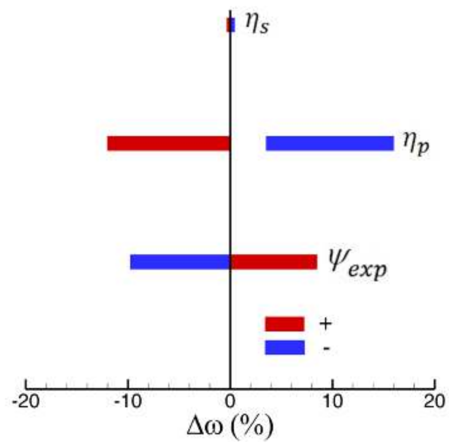

Figure 11. Sensivity of entrainment ratio to ejector components efficiency (Zhu model [13]).

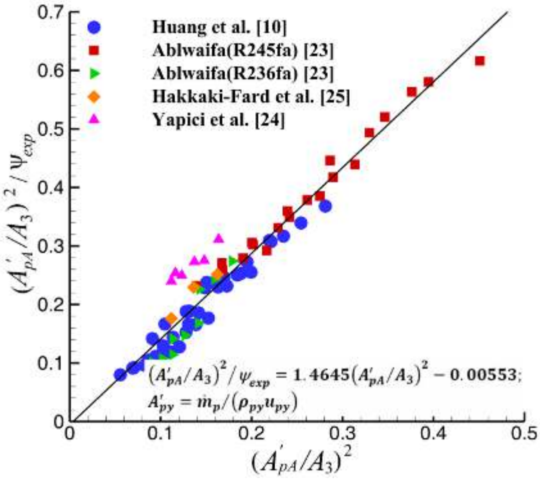

Figure 12. Variation of isentropic efficiency (ψ_{exp}) with area ratio $(\acute{A}_{pA}/A_3)$ (Zhu et al. model [13])

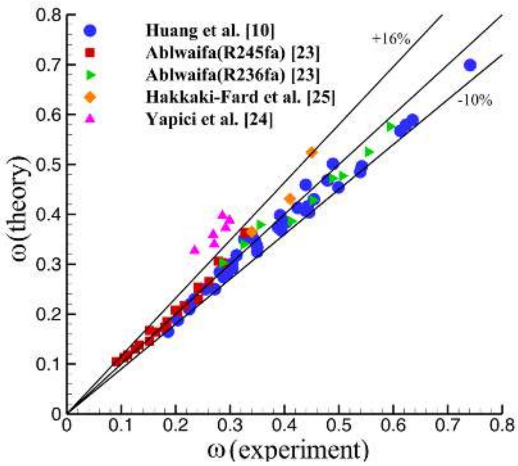

Figure 13. Comparison of the entrainment ratio found by the present method and the experimental data (Zhu et al. model [13])

Cardemil and Colle model

Cardemil and Colle [15] used real gas equations in their model, thus there is no need for modification in this case. In addition to the nozzle isentropic efficiency (η_p), they used another coefficient for the primary flow expansion in suction chamber (η_m). They also used a coefficient ψ to calculate the actual area occupied by the primary flow at hypothetical pressure (A_{p2}) in terms of the effective area occupied by primary flow (A_{2h}) using equation $A_{p2} = A_{2h}/\psi^2$. For the mixing process they used the mixing loss factor (ϕ_m) to take into account the losses due to wall friction in mixing chamber and non-uniformity of the secondary flow due to blockage by the primary nozzle assembly.

The results of sensitivity analysis for this model for $\pm 5\%$ change of parameters are shown in Figures 14 and 15. These figures show that the highest sensitivity of entrainment ratio and condenser critical pressure are for ψ^2 and ϕ_m respectively.

Using the values $\eta_p = 0.95$ and $\eta_m = 0.95$ and the experimental results for entrainment ratio (ω) and condenser critical pressure (P_c^*), ψ^2 and ϕ_m can be determined. According to Figure 16, except for the constant area ejectors used by Yapici et al., there is acorrelation between ψ^2 and the ratio of effective area of the primary flow (A_{2h}) and throat area (A_3). Figure 17 shows a correlation between ϕ_m and the Mach numbers ratio M_m/M_{2h}. M_m is the mixing flow Mach number before the normal shock wave and M_{2h} is the primary flow Mach number at a pressure equal to the secondary

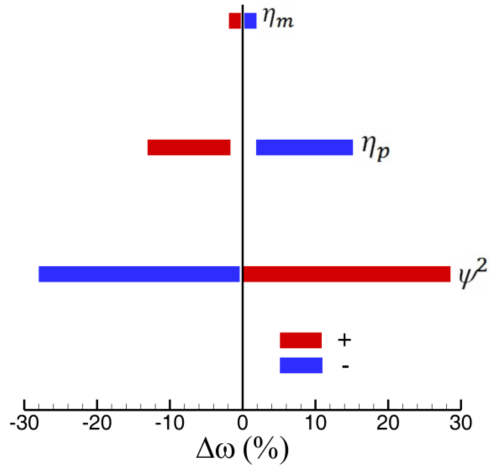

Figure 14. Sensivity of entrainment ratio to ejector components efficiency (Cardemil and Colle model [15])

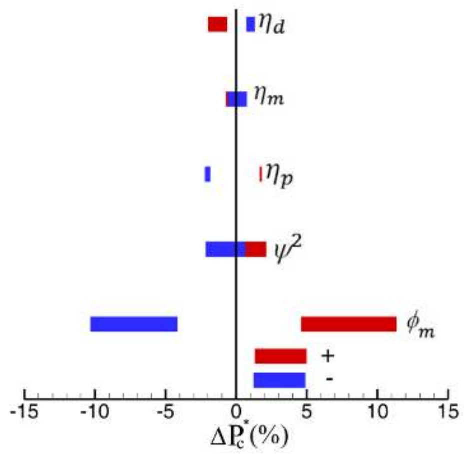

Figure 15. Sensivity of condenser critical pressure to ejector components efficiency (Cardemil and Colle model [15])

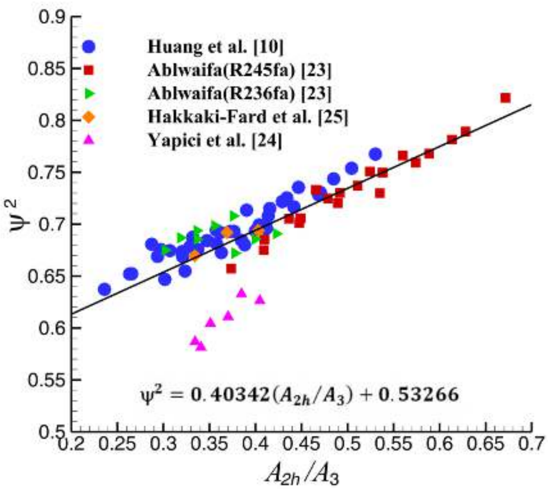

Figure 16. Variation of expansion coefficient (ψ^2) with area ratios (A_{2h}/A_3) (Cardemil and Colle model [15])

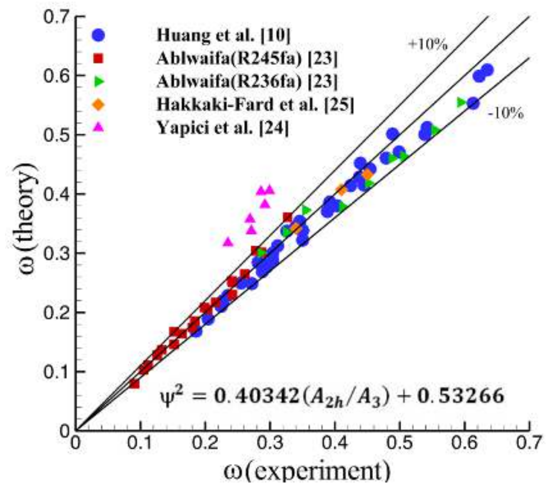

Figure 18. Comparison of the entrainment ratio found by the present method and the experimental data (Cardmil and Colle model [15])

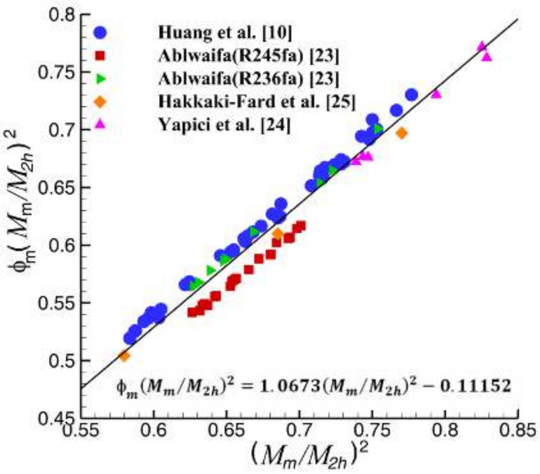

Figure 17. Variation of mixing loss factor (ϕ_m) with Mach number ratios (Cardemil and Colle model [15])

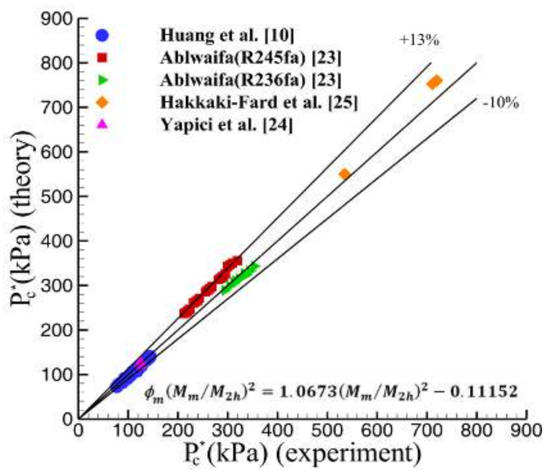

Figure 19. Comparison of the critical back pressure found by the present method and the experimental data (Cardmil and Colle model [15])

inlet pressure. Having the correlations for ψ^2 and ϕ_m, entrainment ratio and condenser critical pressure can be determined for a specific operating condition and geometry. The resulting entrainment ratio and condenser critical pressure are shown in Figures 18 and 19 respectively. Figure 18 shows that also Cardemil and Colle model is incapable of prediction of constant area ejectors, but for other ejectors, it can predict their entrainment ratios with a maximum error of 10%. As shown in Figure 19, critical back pressure is calculated with a maximum error of 13% by this model.

Chen et al. model

Chen et al. [16] presented a model to determine the optimum performance and area ratio using the ideal gas assumption in some processes. In this work however, real gas equations were used for all processes. To model the normal shock, the Equations (11-14) are employed. Chen et al. used an equation based on the ideal gas assumption to calculate area ratio in terms of different flow parameters. Therefore, an equation based on the real gas assumption is required which is shown below:

$$\frac{A_3}{A_t} = \frac{\dfrac{\dot{m}_p}{A_t}}{\dfrac{\dot{m}_p + \dot{m}_s}{A_3}\dfrac{1}{1+\dot{m}_s/\dot{m}_p}} \tag{15}$$

Total pressure after the normal shock equals the condenser pressure and total enthalpy before and after the shock are equal. Hence, knowing the pressure downstream of the shock and assuming isentropic flow downstream of the shock, flux of the mixed flow $\left(\frac{\dot{m}_p+\dot{m}_s}{A_3}\right)$ can be calculated. Also, since the primary nozzle is in choking condition, the isentropic flow assumption helps calculating the primary flow flux $\left(\frac{\dot{m}_p}{A_t}\right)$.

The nozzle isentropic efficiency (η_p), mixing efficiency (η_m) and diffuser efficiency (η_d) are used in Chen et al. model. Diffuser efficiency includes the losses due to the normal shock wave and pressure recovery at diffuser.

Sensitivity analysis for this model showed a high sensitivity of entrainment ratio to changes of some parameters. Therefore, only $\pm 2\%$ change was applied on parameters. Figures 20 and 21 show that the highest sensitivity of entrainment ratio is for η_d and sensitivity of area ratio is almost equal for all parameters.

Assuming η_p=0.95 and using the empirical results for entrainment ratio and condenser critical pressure, mixing and diffuser efficiencies can be calculated. The correlations derived from Chen et al. model for diffuser and mixing efficiencies are shown in Table 2 for five series of data. To validate their model, Chen et al. used the Huang et al. data [10] and calculated entrainment ratio and area ratio. They used 11 diffusers and mixing efficiencies for 11 experimental ejectors, while in the current work, only one correlation is developed for all the ejectors. In Figure 22, the results from the correlations presented in Table 2 for Huang et al. data and the results of Chen et al. [16] have been compared. As seen in Figure 22, the maximum error for entrainment ratio is about 9% for the current method, while the maximum error for the Chen et al. results is about 29%. Moreover, the maximum error for area ratio in the current work is 8% and in Chen et al. work is around 6%.

Calculations were done for the four other correlations in Table 2 and the results are shown in Figure 23. As shown in this figure, the maximum error for entrainment ratio is about 15%. The results also showed that the error for area ratio is about 8%.

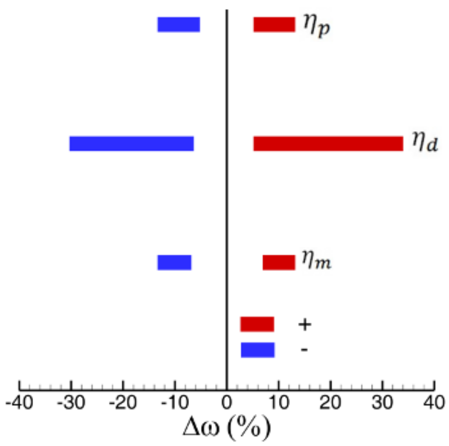

Figure 20. Sensivity of entrainment ratio to ejector components efficiency (Chen model [16])

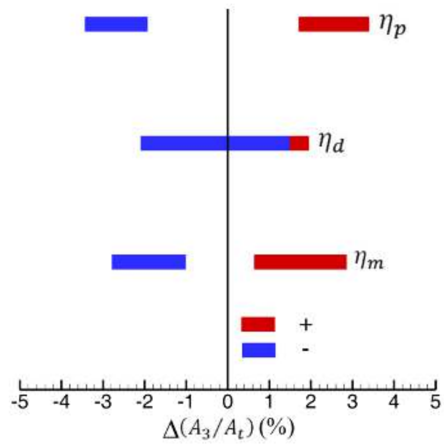

Figure 21. Sensivity of area ratio to ejector components efficiency (Chen model [16])

Table 2. Correlations between diffuser and mixing efficiencies (η_m and η_d) and pressure ratio $\left(\frac{P_g}{P_e}\right)$ and enthalpy ratio $\left(\frac{h_{2'}}{h_e}\right)$ derived for different data series by Chen et al. model.

Experiment	$\left(\frac{P_g}{P_e}\right)\left(\frac{h_{2'}}{h_e}\right)\frac{\ }{\eta_m} = p_1\left(\frac{P_g}{P_e}\right)\left(\frac{h_{2'}}{h_e}\right)+p2$		$\left(\frac{P_g}{P_e}\right)\left(\frac{h_{2'}}{h_e}\right)\frac{\ }{\eta_d} = p_1\left(\frac{P_g}{P_e}\right)\left(\frac{h_{2'}}{h_e}\right)+p2$	
	p_1	p_2	p_1	p_2
Huang et al. [10]	1.0671	0.7418	1.4588	-2.6593
Ablwaifa(R245fa) [23]	1.5976	-2.6462	1.2281	-1.4520
Ablwaifa(R236fa) [23]	0.9323	2.0909	1.5779	-3.8259
Hakkaki-Fard et al. [25]	1.5654	-1.5402	1.0758	-0.0992
Yapici et al. [24]	1.1593	-0.0312	1.4595	-1.9640

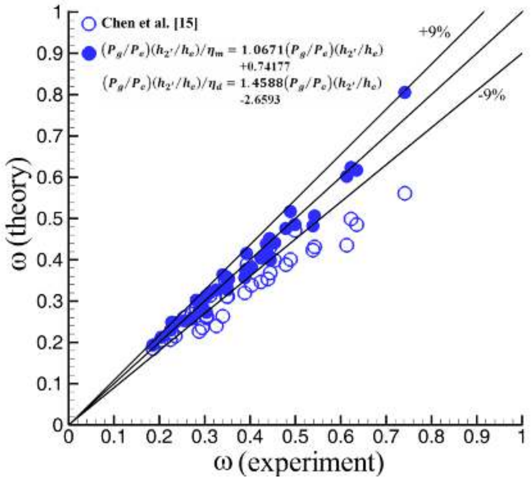

Figure 22. Comparison of the entrainment ratios found by the present model and Chen et al. model [16] for Huang et al. data [10]

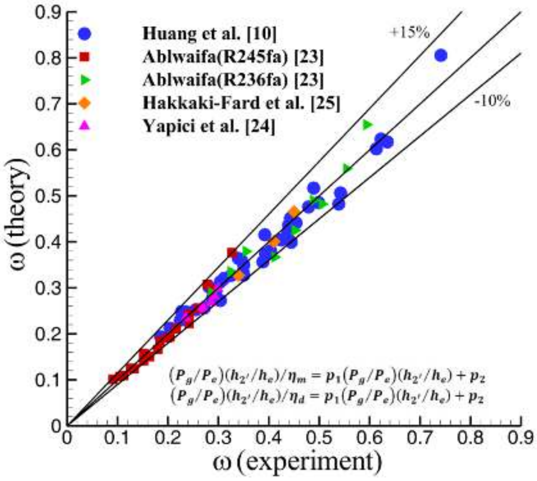

Figure 23. Comparison of the entrainment ratios found by the present method and the experimental data for 5 different data series

CONCLUSION

In this work, the possibility of using flow parameters to determine the isentropic coefficients used in one-dimensional models is investigated. For this purpose, four models of Huang et al. [10] , Zhu et al. [13], Crademil and Colle [15] and Chen et al. [16] and the results from studies performed by Huang et al. [10], Ablwaifa [23], Yapici et al. [24] and Hakkaki-fard et al. [25] are employed using the working fluids R141b, R245a, R236fa, R123 and R134a. All the calculations are made using the real gas equations and the models which are based on the ideal gas assumption have been modified. In Huang et al. model, a correlation between the primary flow isentropic efficiency during expansion from the nozzle exit to hypothetical throat $\left(\phi_p\right)$ and the ratio of throat area and ideal area of the primary flow at hypothetical throat $\left(A_3/A_{py}'\right)$ and a correlation between mixing efficiency $\left(\phi_m\right)$ and the ratio of Mach number of the mixing flow before normal shock and Mach number of the primary flow at hypothetical throat $\left(M_m/M_{py}\right)$ have been derived. The results show that Huang et al. model is not able to predict the performance of constant area ejectors. However, for other ejectors, these correlations can predict entrainment ratio and condenser critical pressure with a maximum error of 10%. In Zhu et al. model, a correlation between isentropic efficiency of primary flow expansion in suction chamber $\left(\psi_{exp}\right)$ and the ratio of ideal effective expanded area of primary flow and throat area $\left(\frac{A_{pA}'}{A_3}\right)$ has been derived. Similarly, in this model, except for the constant area ejectors, entrainment ratios were calculated with the maximum error of 16%. In Cardemil and Colle model, a correlation between the ratio of the primary flow actual and effective areas at hypothetical pressure (ψ^2) and the ratio of effective area occupied by primary flow and throat area $\left(\frac{A_{2h}}{A_3}\right)$ and a correlation between mixing efficiency $\left(\phi_m\right)$ and the ratio of the mixing flow Mach number before the normal shock and the primary flow Mach number at a pressure equal to the secondary flow inlet pressure (M_m/M_{2h}) have been derived. This model is also inappropriate for the constant area ejectors, but for the other ejectors it predicts entrainment ratio and condenser critical pressure with maximum errors of 10% and 13% respectively. In Chen et al. model, for each data series, correlations were derived for diffuser and mixing efficiencies in term of the ratio of primary and secondary pressures $\left(\frac{P_g}{P_e}\right)$ and the ratio of the primary flow enthalpy at mixing pressure and the secondary flow enthalpy at evaporator $(\frac{h_{2'}}{h_e})$. With the aid of these correlations, entrainment ratios and area ratios were calculated with the maximum errors of 15% and 8% respectively. As a result, in three models of Huang et al., Zhu et al. and Cardemil and Colle, some general equations can be derived for isentropic coefficients in terms of flow parameters. These equations can be used to calculate entrainment ratio and critical pressure with good accuracy.

Considering successful prediction of isentropic coefficients using the flow parameters, this idea can be used to predict the isentropic coefficients at subcritical region and for two-phase ejectors in future research. Furthermore, use of machine learning methods to find the relation between isentropic coefficients and flow parameters may be investigated.

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Persian Abstract

چکیده

در مدل‌های یک بعدی تحلیل عملکرد اجکتور در حالت بحرانی ضرایب ایزونتروپیک به کار برده می‌شوند. تاثیر برخی از این ضرایب بر دقت مدل قابل ملاحظه است. مقادیر این ضرائب بستگی به هندسه، سیال عامل و شرایط کاری دارد اما معمولا آنها را ثابت در نظر می‌گیرند یا بر اساس داده‌های تجربی مربوط به یک آزمایش خاص برحسب پارامترهای هندسی و شرایط کاری ارائه می‌کنند. در این مطالعه ایده استفاده از پارامترهای جریان جهت تعیین ضرایب ایزونتروپیک موجود در مدل های یک بعدی ارائه و بررسی شده است. برای این منظور چهار مدل با فرمولاسیون‌های متفاوت به کار رفته است. در این مطالعه فرض گاز واقعی بکار برده شده و بنابراین در مدل‌هایی با فرض گاز ایده‌آل اصلاحات مورد نیاز اعمال شده است. داده‌های تجربی مربوط به اجکتورهایی با هندسه‌ها، شرایط کاری و سیال‌های مختلف مورد استفاده قرار گرفته است. به کمک داده‌های تجربی همبستگی برخی ضرایب ایزونتروپیک موجود در مدل‌ها با پارامترهای جریان نشان داده شده است. با استفاده از روابط به دست آمده برای ضرایب ایزونتروپیک برحسب پارامترهای جریان، نسبت دبی‌های جرمی با حداکثر خطای نسبی کمتر از ۳۵٪ تعیین شد، در حالی که خطای نسبی تعیین نسبت دبی‌های جرمی در اکثر موارد کمتر از ۱۰٪ است. با توجه به دامنه نسبتا وسیع داده‌های تجربی به کار رفته از نظر اندازه اجکتورها و شرایط کاری خطاها قابل قبول می‌باشند.