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Review on Ejector Efficiencies in Various Ejector Systems

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ABSTRACT

The efficiency of the ejector cycle is very sensitive to the ejector efficiency. This paper provides a literature review on ejector efficiencies in various ejector systems, such as vapor compression systems, solar driven ejector systems. The definitions of overall ejector efficiency and ejector component efficiencies in literature are summarized. The assumed constant ejector component efficiencies used in the ejector modeling, and the empirical correlations of the ejector efficiencies developed based on the external measured parameters are summarized and compared; the methods of determining energy efficiencies are summarized. The effects of ejector geometries, operating conditions and working fluid characteristics on ejector efficiencies are discussed. This review will be useful for further research on ejector efficiency, optimal design and control of ejectors and ejector systems.

1. INTRODUCTION

Ejector has been widely used in different cycles for refrigeration purposes, such as recovery of expansion work, utilization of low-grade energy (solar energy, geothermal energy and waste heat). Ejector expansion device is attractive and has great market potential, because it is simple to construct and provide robust operation without moving parts while still yielding significant performance improvements. Ejector expansion device has long service life and low maintenance cost. Figure 1 shows a schematic of an ejector, which consists of motive nozzle, suction nozzle, mixing section and diffuser. Figure 2 shows the ejector working process in a CO_2 pressure enthalpy diagram. The high pressure motive stream expands in the motive nozzle and entrains the low pressure suction stream into the mixing section; then the two streams are mixed in the mixing section, and the mixed stream flows through the diffuser increasing its pressure along the way.

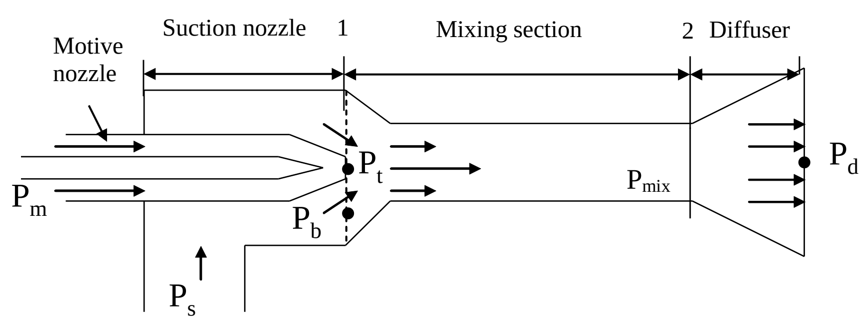

Figure 1: Schematic of ejector working processes

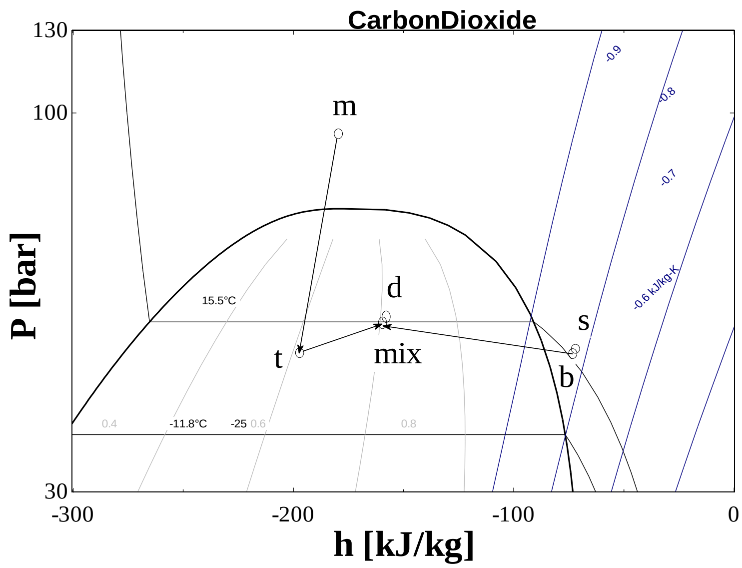

Figure 2: Ejector working processes in a CO_2 pressure-enthalpy diagram

Some review papers have been published to summarize the research efforts and achievements focused on ejector and ejector systems. The review by Sun and Eames (1995) outlined the developments in mathematical modeling and design of jet ejectors. However, this review considered only the thermodynamic model based on two basic approaches, the mixing of the primary fluid and the entrained fluid either at constant pressure or at constant-area. In order to enhance the efficiencies and reduce the cost of ejector cooling systems, efforts made by several researchers have been summarized by Zhang and Shen (2002). Chunnanond and Aphornratana (2004) summarized the ejectors and their applications in refrigeration, and concluded that the understanding of the ejector theory had not been completely cleared. He *et al*. (2009) examined the progress made in the area of mathematical modeling on the ejector, and summarized comprehensively the numerous significant works that has been done on modeling the ejector. Sumeru (2012) provided a comprehensive review of two-phase ejector as an expansion device in vapor compression refrigeration cycle over the past two decades. Sarkar (2012) provided a review of existing literatures on two-phase ejectors and their applications in vapor compression system. In this review paper, geometry, operation and modeling of ejector, and effects of various operating and geometric parameters, and refrigerant varieties on the ejector performances as well as performance characteristics of both subcritical and transcritical vapor compression systems with various cycle configurations are well-summarized. Chen *et al*. (2013) provided a literature review on the recent development in ejectors, applications of ejector refrigeration systems and system performance enhancement. A number of studies are reported and categorized in several topics including refrigerant selections, mathematical modeling and numerical simulation of ejector system, geometric optimizations, operating conditions optimizations and combinations with other refrigeration systems. The efficiency of the ejector cycle is very sensitive to the ejector efficiency (Domanski, 1995). However, there is very limited research on ejector efficiency. Up to now, no review paper focused on ejector efficiencies has been found in the literature. This paper aims to providing a literature review on recent research works that has been done on ejector efficiencies, including overall ejector efficiency, ejector component efficiencies and their determination methods, in order to optimize ejectors and ejector systems.

2. OVERALL EJECTOR EFFICIENCY

2.1 Definitions and Empirical Equations

For the description of an ejector, Köhler *et al*. (2007) first introduced an ejector efficiency. The main advantage of this efficiency definition is the fact that only external parameters of the ejector are used, which can be easily measured. The ejector efficiency is the product of these two isentropic efficiencies, which is calculated by Equation (1).

$$\eta_{ejector} = \varphi \frac{(h'_{s,isen} - h_s)}{(h_m - h'_{m,isen})} \tag{1}$$

In Equation (1), $h'_{s,isen}$ is the specific enthalpy for an assumed isentropic change of state from the suction nozzle inlet condition to the ejector outlet pressure. Therefore, $h'_{s,isen}$ depends on the suction specific entropy s_s and the ejector outlet pressure p_e. The enthalpy $h'_{m,isen}$ is the specific enthalpy for an assumed isentropic change of state from the motive nozzle inlet condition to the ejector outlet pressure, thus it is defined similar to $h'_{s,isen}$. Entrainment ratio is defined as

$$\varphi = \frac{\dot{m}_s}{\dot{m}_m} = \frac{\dot{m}_e}{\dot{m}_g} \tag{2}$$

Elbel and Hrnjak (2008) defined an ejector efficiency based on standard pressure, temperature, and mass flow rate measurements. It compares the amount of expansion work rate recovered by the ejector with the maximum possible expansion work rate recovery potential.

$$\eta_{ejector} = \frac{\dot{W_{rec}}}{\dot{W_{rec,max}}} \tag{3}$$

Elbel and Hrnjak (2008) used a different derivation approach from Köhler *et al*. (2007), but they get the same expression for the ejector efficiency. Ejector efficiency increases when mass entrainment ratio and/or pressure lifting ratio increase.

Dvorak and Vit (2005) defined the ejector efficiency as Equation (4). Butrymowicz *et al*. (2014) calculated the efficiency of the ejector using relation proposed by Dvorak and Vit (2005) in their study.

$$\eta_{ejector} = \frac{\dot{m}_e}{\dot{m}_g} \frac{1-\left(\frac{P_e}{P_c}\right)^{\frac{k-1}{k}}}{\left(\frac{P_g}{P_c}\right)^{\frac{k-1}{k}}-1} \tag{4}$$

McGovern *et al*. (2012) developed ejector performance metrics to evaluate the thermodynamic ideality of a process by comparing useful work done in a real process to that in a defined reference process. McGovern *et al*. (2012) presented four efficiencies based on the comparison of real and reversible processes (Reversible entrainment ratio efficiency, Reversible discharge pressure efficiency, Turbine-compressor efficiency, Compression efficiency), and an exergetic efficiency. Exergetic analysis is a means of evaluating ejector performance from a Second Law point of view. The premise of exergetic efficiency is to compare the useful exergetic output of a system or component to the exergetic input (McGovern *et al*., 2012). An analytical expression for the exergetic efficiency may be obtained when the inlet fluids are ideal gases of identical and constant specific heats. If the further restriction of having inlet fluids at the same temperature is imposed, the discharge enthalpy becomes independent of the entrainment ratio. The discharge temperature equals the inlet temperature, and thus outlet and inlet specific enthalpies are also equal. Consequently, the exergetic ejector efficiency takes the following form.

$$\eta_X = \varphi \frac{\ln(P_d/P_s)}{\ln(P_m/P_d)} \tag{5}$$

Lucas *et al*. (2013) used the equation (1) combined with a correlation of the ejector efficiency by Fiorenzano (2011), derived the ejector efficiency as Equation (6).

$$\eta_{ejector} = 0.43630 \left[\underbrace{\left(\frac{P_s \ln\frac{P_d}{P_s}}{P_m - P_d}\right)^{0.87843}}_{1} \underbrace{\left(\frac{P_s}{P_m}\right)^{0.10313}}_{2} \underbrace{\left(\frac{Oh_m}{Oh_s}\right)^{1.33917}}_{3} \left(\frac{d_{mix}\rho_s}{d_t \rho_m}\right)^{-0.71533} \right] - 0.01770 \tag{6}$$

The correlation coefficients are determined using the experimental data from a CO_2 ejector with a fixed geometry. For the presented correlation, the work of Fiorenzano (2011) is used as a starting point. Fiorenzano (2011) uses dimensions numbers to describe the ejector efficiency. The first term in the correlation published by Fiorenzano (2011) is the ratio of the volumetric work need to isothermally compress an ideal gas from the suction pressure to the ejector exit pressure to the volumetric dissipated energy in an isenthalpic expansion process from the motive pressure to the ejector outlet pressure. The dissipated energy is the energy used by the ejector. Consequently, the first term is an efficiency description of an ideal gas ejector. The second term correlation is the pressure ratio between the motive inlet and the suction inlet of the ejector. The pressure fraction thereby represents the energy within the motive mass flow rate. With increasing pressure fraction, the kinetic energy within the motive mass flow rate increases. The last term in the correlation published by Fiorenzano (2011) is the ratio of Ohnesorge number of the motive and the suction nozzle. The Ohnesorge number is used to describe free jet flows. As shown by Tischendorf *et al*. (2010), the opening angle of the motive mass flow exiting the motive nozzle increases with increasing Ohnesorge number. Thus, it is expected to have an impact on the ejector efficiency.

2.2 Variation of Overall Ejector Efficiencies

Elbel and Hrnjak (2008) investigated the performance of ejector in transcritical R744 system operation experimentally, and found that the ejector performed with a higher efficiency when the high-side pressure was relatively low. It was also found experimentally that despite lower ejector efficiencies, the COP increased as the high-side pressure increased as a result of using the integrated needle to reduce the motive nozzle throat area in the ejector. Ejector efficiency was affected by motive nozzle throat area and diffuser angle. The highest ejector efficiencies were achieved when the smallest diffuser angle of 5° was used to recover the static pressure of the high-speed two-phase flow entering the diffuser. Ejector efficiencies between 3.5% and 14.5% were achieved for different diffuser angles (5°, 10°, 15°) and outdoor air temperatures (35 °C, 45 °C) at T_{id} = 27 °C, RH_{id} =30%, and ε_{IHX} = 90%.

Lucas and Koehler (2012) investigated experimentally the relationship between ejector efficiency and the high-side pressure. The results show that the ejector efficiency has a maximum. The high-side pressure at which the ejector efficiency is maximal is decreasing with decreasing evaporation pressure. Furthermore, it can be seen that the ejector efficiency is decreasing with decreasing evaporation pressures as well as increasing gas cooler outlet temperatures. The ejector efficiency dependency on the gas cooler outlet temperature is small at an evaporation pressure of 3.4 MPa, where the maximum ejector efficiency is decreasing by less than 1% between a gas cooler outlet temperature of 30 °C and 40 °C. However, this dependency is stronger at the evaporation pressure of 2.6 MPa where a 20% decrease of the maximum ejector efficiency is shown.

Nakagawa *et al*. (2011a) experimentally investigated the influence of an internal heat exchanger and an ejector on system COP, as well as the influence of an internal heat exchanger on ejector efficiency. Their research results show that the maximum ejector efficiency for the cycle without internal heat exchanger is about 14%, while that the ejector efficiency is increasing with increasing internal heat exchanger size. The comparison of the data with and without internal heat exchanger show that the COP improvement increases with increasing internal heat exchanger size compared to the baseline cycle with the same internal heat exchanger. Nakagawa *et al*. (2010) presented an experimental investigation of ejector geometry, which shows that the ejector efficiency has maximum with respect to high pressure and ejector efficiencies were reached up to 23%. The maximum of ejector efficiency depends on the ejector geometry, the evaporation temperature and the gas cooler outlet temperature. Nakagawa *et al*. (2011b) investigated the effect of the mixing tube on the ejector efficiency. They determined the ejector efficiency and the COP of the baseline and the ejector cycle with and without an internal heat exchanger. Their results show that there is an optimal mixing length at the maximal ejector efficiency. The maximal measured ejector efficiency of the cycle without internal heat exchanger is 11% while ejector efficiencies of up to 17% with internal heat exchanger are shown.

Banasiak and Hafner (2010) presented an extensive study of the influence of the ejector geometry on the ejector efficiency. The mixing tube diameter, the mixing tube length and the diffuser angle were varied. The ejector efficiency reveals a maximum with respect to high pressure. The results regarding mixing tube length and mixing tube diameter are similar to the data provided by Nakagawa *et al*. (2010, 2011b). The data show an optimum mixing tube length and mixing tube diameter. The variation of the diffuser angle shows maximal ejector efficiencies at a diffuser angle of 5°. The data agrees with the results of Elbel and Hrnjak (2008). Maximal ejector efficiencies of 34% are shown.

Butrymowicz *et al*. (2014) investigated ejection air-conditioning system driven by low temperature heat source, with isobutene as a working fluid. Their research results show that the ejector efficiency is affected by operating condition and refrigerant characteristics k. Similarly like in the case of the variation of the mass entrainment ratio with change of the motive vapor temperature, the efficiency of the ejector significantly increases from η_{min} = 0.08 to the η_{max} = 0.165 in range of 50 < tg < 55 °C and then decreases to the minimum value η_{min}. The motive vapor temperature does not influence on the ejector efficiency at the on-design operating conditions. Butrymowicz *et al*. (2014) found that there is no particular influence of the nozzle position in analyzed range of the operation parameters on the ejector efficiency.

McGovern *et al*. (2012) analyzed the efficiencies of an air-air and a steam-steam ejector and found that the properties of working fluid affect the ejector efficiencies significantly; within the compression ratio from 1 to 1.5, the exergetic efficiency of steam-steam ejector varies from 5% to 40%, while the exergetic efficiency of air-air ejector varies from 0.8% to 2%. For general air-air and steam-steam ejectors, the exergetic efficiency η_X is very close in numerical value to the reversible entrainment ratio efficiency η_{RER}. The exergetic efficiency η_X for an ideal gas ejector with inlet fluids at the same temperature is identical to the reversible entrainment ratio efficiency η_{RER}.

3. EJECTOR COMPONENT EFFICIENCIES

3.1 Definitions

Ejector consists of motive nozzle, suction nozzle, mixing section and diffuser.

3.1.1 Motive nozzle efficiency:

The isentropic efficiency of the motive nozzle is defined as (Liu and Groll, 2013, Yu and Li, 2007):

$$\eta_m = \frac{h_m - h_t}{h_m - h_{t,is}} \tag{7}$$

3.1.2 Suction nozzle efficiency:

The isentropic efficiency of the suction nozzle is defined as (Liu and Groll, 2013, Yu and Li, 2007):

$$\eta_s = \frac{h_s - h_b}{h_s - h_{b,is}} \tag{8}$$

3.1.3 Mixing efficiency:

The mixing section efficiency η_{mix} is assumed to account for the friction losses in the mixing chamber in Huang (1999) and Liu and Groll (2013).

$$p_t A_t + \eta_{mix}\rho_t A_t V_t^2 + p_b\left(A_{mix} - A_t\right) + \eta_{mix}\rho_b\left(A_{mix} - A_t\right)V_b^2 = p_{mix}A_{mix} + \rho_{mix}A_{mix}V_{mix}^2 \tag{9}$$

The mixing efficiency is given as Equation 10 (Manjili and Yavari 2012, Yu *et al*. 2007):

$$\eta_{mix} = \frac{u_{mix'}^2}{u_{mix}^2} \tag{10}$$

Where u' is the corrected form of u, in order to account for mixing section losses.

3.1.4 Diffuser

The diffuser efficiency is defined as Equation 11 in Ksayer (2007) and Li and Groll (2005).

$$\eta_d = \frac{h_{d,out,is} - h_{ms,out}}{h_{d,out} - h_{ms,out}} \tag{11}$$

Elbel and Hrnjak (2008) defined diffuser efficiency as follows.

$$\eta_d = \frac{(h_{d,out,is} - h_{ms,out})}{\frac{1}{2}u_{ms,out}^2} \tag{12}$$

Liu and Groll (2013) used the pressure recovery coefficient in their simulation study. The pressure recovery coefficient, *Ct*, is defined as:

$$Ct = \frac{p_d - p_{mix}}{\frac{1}{2}\rho_{mix}V_{mix}^2} \tag{13}$$

Owen *et al*. (1992) proposed a correlation to calculate the pressure recovery coefficient (Liu and Groll, 2013):

$$Ct = 0.85\rho_{mix}\left[1-\left(\frac{A_{mix}}{A_d}\right)^2\right]\left[\frac{x_{mix}^2}{\rho_{g,mix}} + \frac{(1-x_{mix})^2}{\rho_{f,mix}}\right] \tag{14}$$

3.2 Assumption Values

In most of the literature studies, values of 0.7 to 1.0 were assumed for the individual ejector component efficiencies as listed in Table 1.

Table 1: Summary of literature - assumed ejector component efficiencies in modeling studies

Authors	System Type	Fluid	η_m	η_s	η_{mix}	η_d
Keenan *et al*. (1950)	ejector	Air	1.0	1.0		
Alexis and Rogdakis (2003)	steam ejector refrigerator system	Water	0.7			0.8
Sun (1996)	powered by low-grade thermal energy	LiBr-H_2O/H_2O-NH_3	0.85	0.85		0.85
Vereda *et al*. (2012)	ejector-absorption refrigeration cycle	Ammonia/lithium nitrate	0.85	0.85	0.9	0.8
Domanski (1995)	compression cycle	-	0.85 - 0.9	0.85-0.9		0.7
Yapici and Ersoy (2005)	low grade waste heat in the vapor generator	R123	0.85	0.85		0.85
Yu and Li (2007)	conventional ejector refrigeration system	R141b	0.9		0.85	0.85
Yu *et al*. (2007)	regenerative ejector refrigeration cycle	R142b	0.85		0.95	0.85
Elbel and Hrnjak (2004)	vapor compression cycle	CO_2	0.9	0.9		0.9
Li and Groll (2005)	vapor compression cycle	CO_2	0.9	0.9		0.8
Ksayer and Clodic (2006)	vapor compression cycle	CO_2	0.85	0.85		0.75
Ksayer (2007)	vapor/liquid compression cycle	R141b	0.95	1.0	0.9-0.98	1
Deng *et al*. (2007)	vapor compression cycle	CO_2	0.7	0.7		0.8
Sarkar (2008)	vapor compression cycle	CO_2	0.8	0.8		0.75
Elbel and Hrnjak (2008)	vapor compression cycle	CO_2	0.8	0.8		0.8
Sun and Ma (2011)	vapor compression cycle	CO_2	0.9	0.9		0.8
Manjili and Yavari (2012)	vapor compression cycle	CO_2	0.7	0.7	0.95	0.8

3.3 Determination Methods

Varga *et al*. (2009) firstly determined ejector efficiencies for the primary nozzle, suction, mixing and diffuser according to their definitions, using an axi-symmetric computational fluid dynamics (CFD) model. They used water as working fluid and selected the operating conditions in a range that would be suitable for an air-conditioner powered by solar thermal energy. Their research results show that nozzle efficiency can be considered as constant, the efficiencies related to the suction, mixing and diffuser sections of the ejector depend on operating conditions.

Ksayer (2007) found that the mixing efficiency varies between 0.9 and 0.98 and depends on the diameter ratio of the nozzle throat and the constant area diameter. A correlation of η_{mix} is elaborated:

$$\eta_{mix} = -0.0113(D_{mix}/D_t)^2 + 1.0501 \tag{15}$$

Liu and Groll (2013) determined ejector component efficiencies in refrigeration cycles based on an ejector model and the measured performance data. A two-phase flow ejector model, consisting of sub-models for motive nozzle flow, suction nozzle flow, mixing section flow and diffuser flow, was developed. Experimental data in conjunction with the ejector simulation model were used to determine the isentropic efficiencies of the motive and suction nozzles, and the efficiency of mixing section. The application of this method is illustrated with a case study of a controllable ejector in transcritical CO_2 air conditioning systems at outdoor air temperatures of 27.8 °C, 35 °C and 37.8 °C. Study results show that ejector geometries and operating conditions affect ejector component efficiencies significantly: 1) the motive nozzle efficiency is very sensitive to ejector throat diameter and it ranges from 0.50 to 0.93; 2) the suction nozzle efficiency somewhat is affected by motive nozzle throat diameter, motive nozzle exit position and outdoor air temperature, and it ranges from 0.37 to 0.90; 3) the efficiency of mixing section is affected by the motive nozzle exit position, and it ranges from 0.50 to 1.00. Small motive nozzle throat diameter leads to low motive nozzle efficiency and high suction nozzle efficiency. The motive nozzle placed at a distance of 1.5 times D_{mix} from the mixing section inlet resulted in a little bit higher mixing section efficiency than at a distance of 6 times D_{mix}. The ranges of the determined ejector component efficiencies in this study are larger than those of the assumed constant ejector component efficiencies in literatures.

Based on the determined efficiencies of ejector components at various ejector geometric parameters and various operating conditions, three empirical correlations of ejector efficiencies were developed. Equations (16)-(18) show the correlations for the motive nozzle isentropic efficiency η_m, the suction nozzle isentropic efficiency η_s and the mixing section efficiency η_{mix}, respectively, as the functions of ejector geometry, pressure ratio, and ejection ratio (Liu and Groll, 2013).

$$\eta_m = -36.137 - 4.160\left(\frac{P_m}{P_s}\right) + 1.161\left(\frac{P_m}{P_s}\right)^2 - 0.106\left(\frac{P_m}{P_s}\right)^3 + 212.320\left(\frac{D_t}{D_{mix}}\right) - 355.359\left(\frac{D_t}{D_{mix}}\right)^2 + 196.035\left(\frac{D_t}{D_{mix}}\right)^3 \tag{16}$$

$$\begin{aligned}\eta_s = & -3173.171 + 934.102\left(\frac{P_m}{P_s}\right) - 314.471\left(\frac{P_m}{P_s}\right)^2 + 79.521\left(\frac{P_m}{P_s}\right)^3 - 12.222\left(\frac{P_m}{P_s}\right)^4 \\ & + 0.814\left(\frac{P_m}{P_s}\right)^5 + 694222.1\varphi - 2956145\varphi^2 + 7950453\varphi^3 - 114327270\varphi^4 + 6689155\varphi^5 \\ & - 649905.1Z + 2647000Z^2 - 6885025Z^3 + 9627161Z^4 - 5490126Z^5 \left(Z = \varphi\left(\frac{P_m}{P_s}\right)^{0.02}\right)\end{aligned} \tag{17}$$

$$\eta_{mix} = -6869.077 + 19308.18Z' - 18089.31Z'^2 + 5649.417Z'^3 \left(Z' = \left(\frac{D_t}{D_{mix}}\right)^{0.1}(1+\varphi)^{0.35}\right) \tag{18}$$

These correlations should be used within the following boundaries:

8.0 MPa < P_m <14.0 MPa, 2.5 MPa < P_s <5.0 MPa,

40 °C < T_m < 60 °C, 15 °C < T_s < 26 °C,
0.1 g/s < $\dot{m}_m$ < 0.25 g/s, 0.05 g/s < $\dot{m}_s$ < 0.07 g/s, 1.8 mm < D_t <2.7 mm, D_{mix} = 4 mm.

4. CONCLUSIONS

This paper describes a number of research studies on overall ejector efficiency and ejector component efficiencies. The different definitions of overall ejector efficiencies were reviewed and discussed. The investigations about the effects of ejector geometries, operating conditions and working fluid characteristics on overall ejector efficiencies in literature are reviewed. The assumed constant ejector component efficiencies are summarized. The methods of determining ejector component efficiencies were reviewed, such as CFD simulation method, a method combining experimental data and simulation modeling. It is hoped that this contribution will be useful for the future research on the optimal design and control of ejectors and ejector systems.

Though a large amount of works have been conducted on ejector efficiencies, further efforts are still needed:

1) To improve the methods of determining actual ejector component efficiencies.
2) To make a comprehensive study about the effects of operating conditions, ejector geometries and working fluid characteristics on ejector component efficiencies.

NOMENCLATURE

COP	coefficient of performance	(–)
h	specific enthalpy	(kJ/kg)
$\dot{m}$	mass flow rate	(kg/s)
Oh	ohnesorge number	(–)
P	pressure	MPa, bar
s	specific entropy	(kJ/kg/K)
η	efficiency	(–)
ε	effectiveness	(–)
φ	ejection ratio	(–)
k	adiabatic exponent	(–)

Subscript

b	suction nozzle exit
c	condenser
d	diffuser
e	evaporator; secondary fluid
g	generator; primary fluid
m	motive nozzle
mix	mixing section
id	indoor
IHX	internal heat exchanger
isen	isentropic
RER	reversible entrainment ratio
s	suction nozzle
t	motive nozzle throat
X	exergetic

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