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Performance Predictions of Dry and Wet Vapors Ejectors Over Entire Operational Range

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Abstract: If a traditional ideal-gas ejector model is used to evaluate the performance of a wet vapor ejector, large deviations from the experimental results will be unavoidable. Moreover, the model usually fails to assess the ejector performance at subcritical mode. In this paper, we proposed a novel model to evaluate the performance of both dry and wet vapors ejectors over the entire operational range at critical or subcritical modes. The model was obtained by integrating the linear characteristic equations of ejector with critical and breakdown points models, which were developed based on the assumptions of constant-pressure mixing and constant-pressure disturbing. In the models, the equations of the two-phase speed of sound and the property of real gas were introduced and ejector component efficiencies were optimized to improve the accuracy of evaluation. It was validated that the proposed model for the entire operational range can achieve a better performance than those existing for R134a, R141b and R245fa. The critical and breakdown points models were further used to investigate the effect of operational parameters on the performance of an ejector refrigeration system (ERS). The theoretical results indicated that decreasing the saturated generating temperature when the actual condensing temperature decreases, and/or increasing the saturated evaporating temperature can improve the performance of ERS significantly. Moreover, superheating the primary flow before it enters the ejector can further improve the performance of an ERS using R134a as a working fluid.

Keywords: ejector performance; wet vapor; dry vapor; critical mode; subcritical mode; superheating

1. Introduction

Electrical energy consumed by compressor-based refrigeration systems (CRSs), which provide cooling capacities for air conditioning systems, rapidly grows with people's increasing demand for thermal comfort [1,2]. Ejector refrigeration technology is an energy-efficient strategy [1,3] and an attractive alternative to a conventional compressor-based refrigeration. Thus, researches on the ejector refrigeration systems (ERSs) have attracted great attentions, in particular theoretical models of ejectors since the ejector is the heart of an ejector refrigeration system.

An ejector refrigeration system (ERS) basically includes an ejector, an evaporator, a generator, a condenser, a pump and an expansion valve, as illustrated in Figure 1. The corresponding thermodynamic cycle is shown in Figure 2. In the generator, the liquid refrigerant is evaporated at high temperature and constant pressure and changed into high saturated vapor (called the primary flow) at point p_0 , then it enters the ejector and entrains the saturated vapor (called the secondary flow) at point s_0 , which have provided cooling by evaporating at constant pressure and low temperature in the evaporator. The two streams then mix and the mixed flow experiences a pressure lift in the

ejector. At point c, it leaves the ejector as a medium-pressure, superheated vapor, and then enters the condenser to be condensed at constant pressure. At point c1, the liquid refrigerant from the condenser is divided into two parts, one being pumped into the generator at point g, and the other expanding through the expansion valve and entering the evaporator at point e. Then the two parts of refrigerant evaporate in the generator and evaporator, respectively.

Figure 1. Equipments diagram for basic ejector refrigeration system.

Figure 2. Pressure-enthalpy charts for ejector refrigeration cycles with dry and wet vapors.

Many ideal-gas ejector models for critical mode have been developed as they are simple yet effective in the ejector performance evaluation. Keenan and Newman [4] proposed a 1-D ejector model, which is probably the first constant-area mixing model. Then, Keenan et al. [5] developed a model for a constant-pressure mixing ejector. Munday and Bagster [6] postulated that the secondary flow is choked at a fictive throat and then it mixes with the primary flow under a uniform pressure. Based on Keenan's model [5], Eames et al. [7] took three isentropic efficiencies to consider losses inside an ejector. Their experimental results were within 85% of theoretical predictions. Huang et al. [8] introduced four ejector component efficiencies to account for losses inside the ejector in their ideal-gas ejector model. Compared with their experiment data of R141b ejectors, the maximum evaluation error is 22.99%. Besagni et al. [9] proposed a model for sonic and sub-sonic ejectors based on the ideal-gas assumption. In the model, the correlation equations of ejector component efficiencies were determined by CFD method. Khennich et al. [10] carried out an exergy analysis of a one-phase ejector taking R141b as the refrigerant and found that large exergy losses are created during the shock. If an ideal-gas model is used to predict the performance of wet-vapor ejectors, large deviations from the experimental results will be unavoidable, since the wet vapor refrigerants, such as R11, R22, and etc., usually deviate far from the ideal gas law [11]. To address this issue, real gas treatment was introduced in ejector models

for critical mode. Lear et al. [12] presented a 1-D ejector model, which can handle the two-phase flow in an ejector with Fabri choking. Cardemil and Colle [13] proposed a real-gas ejector model. The evaluation results show good agreement with the experimental data for the R141b [8], water [14] and carbon dioxide [15] ejectors operating at critical modes.

All the ejector models mentioned above are only capable of evaluating the ejector performance at critical mode. To date, a few attempts were made to develop the ejector model for the entire operational range, including critical and subcritical modes. Chen et al. [16] extended the ideal-gas ejector model of Huang et al. [8] to the entire operational range. The prediction errors at subcritical mode were about 20%. Chen et al. [11] also proposed a real-gas ejector model for the entire operational range. For the dry vapor refrigerant R141b, the prediction errors of critical condensing pressure and entrainment ratio were within $\pm 9\%$ and $\pm 17\%$, respectively. For the wet vapor refrigerant R11, the maximum error of COP was -31.58% .

From the literature review above, it is found that there are few ejector models, which can predict the ejector performance over the entire operational range with high accuracy, especially for wet vapors ejectors. The main objective of this paper is hence to propose a novel model for both dry and wet vapor ejectors over the entire operational range. To obtain high accuracy, the following strategies were adopted: (1) developing the critical and breakdown points models based on constant-pressure mixing and constant-pressure disturbing, and integrating them with linear characteristic equations of ejector to form the model; (2) introducing real gas property and the equations of sound speed with high evaluation accuracy into the models; (3) adopting the analysis method of the effect of change (EOC) of the efficiencies [17] and the sparsity-enhanced optimization method [18] to identify and determine the correlation equations of ejector components efficiencies. Then, the model was employed to analyze the influences on the ejector performance induced by superheating the primary flow before it enters the ejector.

2. Preliminaries

Ejector is a compression component of an ERS and the entrainment ratio u , which is the ratio between the mass flow rate of the secondary flow and the primary flow, is a key parameter of it. Schematic diagram of an ejector is shown in Figure 3. The working fluids of ERSs are classified into two categories: wet vapor (R11, R12, R134a or Steam) and dry vapor (R113, R123 or R141b) [19]. The thermodynamic cycle for the two working fluids are shown in Figure 2. It can be observed that during the expansion of the primary flow in the nozzle, the isentropic line for wet working fluid goes into the wet vapor region. While for dry working fluid, it goes into the superheated region. It can be deduced that the property of wet vapor is far from that of ideal gas.

Figure 3. Schematic diagram of ejector.

2.1. Operation Modes

Figure 4 illustrates the three operational modes of ejector. The modes are divided by the two key points, i.e., critical point, at which entrainment ratio u equals critical entrainment ratio u_c and

condensing pressure P_c equals critical condensing pressure P_{cc} , and breakdown point, at which $u = 0$ and P_c is equal to breakdown condensing pressure P_{cb} . The three operational modes are:

1. Critical mode: the primary flow and the secondary flow are choked at the nozzle throat and section y-y, respectively, and u is constant, as $P_c \leq P_{cc}$;
2. Subcritical mode: only the primary flow is choked at the nozzle throat and u decreases with P_c as $P_{cc} < P_c \leq P_{cb}$;
3. Back-flow mode: no choking phenomenon exists and $u < 0$ as $P_c < P_{cb}$.

For critical mode, the performance characteristic equation of ejector and its boundary are:

$$u = u_c \quad (1)$$

$$P_c \leq P_{cc} \quad (2)$$

For subcritical mode, the performance characteristic equation of ejector and its boundary are:

$$u = u_c (P_{cb} - P_c) / (P_{cb} - P_{cc}) \quad (3)$$

$$P_{cc} < P_c \leq P_{cb} \quad (4)$$

Figure 4. Operational modes of ejector.

2.2. Assumptions

It can be found that the ejector loses its function at backflow mode as indicated in Figure 4. Thus, the entire operational range only includes the critical and subcritical modes. The prediction performance at critical mode can be obtained by P_{cc} , u_c , Equations (1) and (2). Similarly, the model at subcritical mode can be obtained by P_{cc} , u_c , P_{cb} , Equations (3) and (4). Therefore, the ejector model over the entire operational range can be developed based on the models at the critical and breakdown points. To develop the ejector models at the critical and breakdown points, we require the following assumptions:

1. The inner wall of the ejector is adiabatic and the flow is one dimensional and steady inside the ejector;
2. The kinetic energy of the primary flow at the nozzle inlet, the secondary flow at the suction inlet and the mixing flow at the diffuser outlet are negligible;
3. Constant-pressure mixing exists at critical point. After fanning out of the nozzle, the primary flow entrains but does not mix with the secondary flow before section y-y. The secondary flow is choked at section y-y and starts to mix with the primary flow with a uniform pressure, i.e., $P_m = P_{py} = P_{sy}$ before the shock;
4. Constant-pressure disturbing occurs at breakdown point. The primary and secondary flows do not mix before section x-x. Then, before the shock, the secondary flow disturbs the the primary flow by mixing in and out of the primary flow with a uniform pressure, i.e., $P_m = P_{px} = P_{sx}$.

3. Ejector Performance Modelling

3.1. Model for the Critical Point

3.1.1. Primary Flow from Inlet to Nozzle Throat

According to assumption (2), the inlet enthalpy h_{p0} is thought as stagnant enthalpy. Then the energy conservation equation is:

$$h_{p0} - h_t = v_t^2 / 2 \quad (5)$$

where h_t and v_t are enthalpy and velocity at nozzle throat, respectively. The velocity v_t equals the sound of speed since the primary flow is choked at throat, namely

$$v_t = a_t \quad (6)$$

where, a_t is the speed of sound at the nozzle throat. For single phase state, it can be obtained directly from NIST [20]; for wet vapor, the calculation method of it will be described in Section 3.3. In general, it can be expressed as the following relation:

$$a_t = \text{Fun}(P_t, \rho_t) \quad (7)$$

where,

$$\rho_t = \rho(P_t, h_t) \quad (8)$$

In this model, the isentropic efficiency η_t is introduced to account for the losses during the expansion process, and it should be determined by experimental data. It is defined as:

$$\eta_t = (h_{p0} - h_t) / (h_{p0} - h_{ts}) \quad (9)$$

where h_{ts} is the enthalpy obtained after a hypothetical isentropic expansion from the inlet to the nozzle throat. It can be expressed as:

$$h_{ts} = h(s_{ts}, P_t) \quad (10)$$

where s_{ts} is the entropy of the primary flow corresponding to the hypothetical enthalpy h_{ts} at throat, which equals the entropy s_{p0} at the inlet of nozzle. P_t is the pressure at throat, which is a parameter used to determine the speed of sound and can be derived by an iterative process using Equations (5)–(10).

The primary flow rate is obtained by the following relation:

$$m_p = \rho_t A_t v_t \quad (11)$$

3.1.2. Primary Flow from Throat to Exit of Nozzle

The energy conservation equation during this process is:

$$h_t + v_t^2 / 2 = h_{p1} + v_{p1}^2 / 2 \quad (12)$$

where h_{p1} and v_{p1} are enthalpy and velocity at nozzle exit, respectively.

The velocity v_{p1} meets the following mass conservation equation:

$$m_p = \rho_{p1} A_{p1} v_{p1} \quad (13)$$

where,

$$\rho_{p1} = \rho(P_{p1}, h_{p1}) \quad (14)$$

This process is a simple expansion process since the back pressure, namely, the pressure of the secondary flow are much less than that of the primary flow in nozzle throat and thus oblique shocks will occur out of the nozzle [13] rather than between the throat and the exit of nozzle. Thus, it is assumed that the expansion process is approximately isentropic as other ejector models. Then, the entropy of the primary flow at the exit of nozzle, s_{p1} , follows:

$$s_{p1} \approx s_t = s(h_t, P_t) \quad (15)$$

Then,

$$h_{p1} = h(s_{p1}, P_{p1}) \quad (16)$$

3.1.3. Primary Flow from Nozzle Exit to Section y-y

The energy conservation equation between the nozzle exit and section y-y is:

$$h_{p1} + v_{p1}^2/2 = h_{py} + v_{py}^2/2 \quad (17)$$

where h_{py} and v_{py} are enthalpy and velocity at section y-y, respectively.

The velocity v_{py} meets the following mass conservation equation:

$$m_p = \rho_{py} A_{py} v_{py} \quad (18)$$

where,

$$\rho_{py} = \rho(P_{py}, h_{py}) \quad (19)$$

The isentropic efficiency η_m is used to considering the losses cause by the oblique shocks during the expansion process, and it is defined as:

$$\eta_m = (h_{p1} - h_{py}) / (h_{p1} - h_{pys}) \quad (20)$$

where h_{pys} is the enthalpy of the secondary at section y-y after an isentropic expansion process, it follows:

$$h_{pys} = h(P_{py}, s_{p1}) \quad (21)$$

where, $P_{py} = P_{sy}$, according to the assumption (3).

3.1.4. Secondary Flow from Suction Inlet to Section y-y

According to the assumption (2), the suction enthalpy h_{s0} of the secondary flow is treated as the stagnant enthalpy. Then, the energy conservation equation is:

$$h_{s0} = h_{sy} + v_{sy}^2/2 \quad (22)$$

where, v_{sy} is velocity of the secondary flow at section y-y, and it equals the speed of sound according to the assumption (3), i.e.,

$$v_{sy} = a_{sy} \quad (23)$$

where, a_{sy} is the speed of sound of the secondary flow at section y-y. It can be expressed as:

$$a_{sy} = Fun(P_{sy}, \rho_{sy}) \quad (24)$$

where,

$$\rho_{sy} = \rho(P_{sy}, h_{sy}) \quad (25)$$

During the expansion process, the isentropic efficiency is defined as

$$\eta_s = (h_{s0} - h_{sy}) / (h_{s0} - h_{sys}) \quad (26)$$

where h_{sys} is the enthalpy of the secondary flow at section y-y after an isentropic expansion from suction to section y-y. It follows:

$$h_{sys} = h(s_{sys}, P_{sy}) \quad (27)$$

where s_{sys} is the entropy corresponding to the enthalpy h_{sys} at section y-y and it equals s_{s0} ; P_{sy} is the pressure at section y-y, which is a parameter used to determine the speed of sound. It can be obtained by an iterative process using Equations (22)–(27).

The secondary flow rate is derived by the following relation:

$$m_s = \rho_{sy} A_{sy} v_{sy} \quad (28)$$

The area A_{sy} follows:

$$A_3 = A_{py} + A_{sy} \quad (29)$$

3.1.5. Mixing and Shock in Constant Section

The mixing starts at section y-y with a uniform pressure P_m . The energy conservation equation can be expressed as:

$$m_p \left(h_{py} + \frac{v_{py}^2}{2} \right) + m_s \left(h_{sy} + \frac{v_{sy}^2}{2} \right) = (m_p + m_s) \left(h_m + \frac{v_m^2}{2} \right) \quad (30)$$

Since the mixing occurs under constant pressure P_m in the mixing chamber whose sectional-area are constant, the momentum conservation equation can be expressed as :

$$\phi_m (m_p v_{py} + m_s v_{sy}) = (m_p + m_s) v_m \quad (31)$$

where ϕ_m is the momentum transfer efficiency which is used to account for the losses during the mixing of the two streams. The density of the mixed flow is:

$$\rho_m = \rho(P_m, h_m) \quad (32)$$

The pressure rises from P_m to P_3 when the mixed flow passes through the shock wave. They meet the following equation [13]:

$$v_m^2 = \frac{P_3 - P_m}{\rho_3 - \rho_m} \left(\frac{\rho_3}{\rho_m} \right) \quad (33)$$

The energy conservation equation can be expressed as:

$$h_3 + v_3^2/2 = h_m + v_m^2/2 \quad (34)$$

Hugoniot equation [21] expresses the thermodynamic parameter relationship while the flow experiences a normal shock wave:

$$h_3 - h_m = \frac{P_3 - P_m}{2} \left(\frac{\rho_m + \rho_3}{\rho_m \rho_3} \right) \quad (35)$$

where,

$$h_3 = h(P_3, \rho_3) \quad (36)$$

3.1.6. Diffuser

The energy conservation equation of the mixed flow exhausting from diffuser is expressed as follows:

$$h_3 + v_3^2/2 = h_{cc} \quad (37)$$

where h_{cc} is the enthalpy at the exit of diffuser when the ejector operates at critical point. Introducing η_d to express the efficiency during the compression process. Then

$$\eta_d = (h_{cs} - h_3) / (h_{cc} - h_3) \quad (38)$$

where h_{cs} is the enthalpy obtained by an ideal isentropic compression process, i.e.,

$$s_{cs} = s_3 \quad (39)$$

During the hypothetic isentropic process, the pressure recovers from P_3 to P_{cs} . The critical condensing pressure P_{cc} , i.e., the pressure at the exit of the diffuser after a real process, is equal to P_{cs} , namely,

$$P_{cc} = P_{cs} = P(h_{cs}, s_{cs}) \quad (40)$$

3.2. Model for the Breakdown Point

3.2.1. Primary Flow from Inlet to Nozzle Exit

The flow process in the nozzle is the same as that at critical point. Thus, Equations (5)–(16) can be used at breakdown point.

3.2.2. Primary Flow from Nozzle Exit to Section x-x

The process is approximately an isentropic process, i.e.,

$$s_{px} \approx s_p \quad (41)$$

According to the assumption (4), at section x-x, the pressure of the primary flow equals that of the secondary flow. i.e. $P_{px} = P_{sx}$. Then,

$$h_{px} = h(P_{px}, s_{px}) \quad (42)$$

The velocity of the primary flow at section x-x, v_{px} , can be obtained by the following energy conservation equation:

$$h_{p1} + v_{p1}^2/2 = h_{px} + v_{px}^2/2 \quad (43)$$

3.2.3. Secondary flow at section x-x

According to the assumption (4), the secondary flow does not mix with the primary flow before section x-x. Thus, the energy of the secondary flow at section x-x equals h_{s0} , i.e.,

$$h_{sx} + v_{sx}^2/2 = h_{s0} \quad (44)$$

At breakdown point, the entrainment ratio equals zero, then it can be assumed that the velocity of the secondary flow is small enough and the pressure of the secondary flow at section x-x equals inlet pressure, i.e., $P_{sx} = P_{s0}$.

3.2.4. Constant Pressure Disturbing

Although the entrainment ratio is equal to zero, the breakdown condensing pressure, P_{cb} , is changed with the pressure of the secondary flow. Obviously, the secondary flow has an effect on P_{cb} , in other words, the secondary flow disturbs the primary flow to influence P_{cb} as assumption (4). To consider this effect, it is assumed that the flow rate of the secondary flow is small enough and equals $1e - 6$ kg/s and introducing the momentum transfer efficiency ϕ_{mb} in momentum conservation equation to account for the losses from section x-x to section m-m.

Applying the energy conservation, gives:

$$m_p \left(h_{px} + \frac{v_{px}^2}{2} \right) + m_s \left(h_{sx} + \frac{v_{sx}^2}{2} \right) = (m_p + m_s) \left(h_m + \frac{v_m^2}{2} \right) \quad (45)$$

The momentum conservation equation is:

$$\phi_{mb} (m_p v_{px} + m_s v_{sx}) = (m_p + m_s) v_m \quad (46)$$

According assumption (4), the constant-pressure disturbing starts at section x-x with a uniform pressure, that is $P_m = P_{px} = P_{sx}$. Then, the density of the flow is:

$$\rho_m = \rho(P_m, h_m) \quad (47)$$

It is assumed the flow experiences a shock wave and the pressure rises from P_m to P_3 . This process can be described by Equations (33)–(36) in critical model.

3.2.5. Diffuser

Assuming the kinetic energy being zero at exit of diffuser, the energy conservation equation is expressed as:

$$h_3 + u_3^2/2 = h_{cb} \quad (48)$$

where h_{cb} is the enthalpy at the exit of diffuser at breakdown point. The isentropic efficiency η_d is defined as:

$$\eta_d = (h_{cs} - h_3) / (h_{cb} - h_3) \quad (49)$$

where h_{cs} is the enthalpy after an ideal isentropic compression process, i.e.,

$$s_{cs} = s_3 \quad (50)$$

The pressure P_{cb} at the exit of the diffuser in a real process is equal to P_{cs} , which is corresponding to the enthalpy h_{cs} . Then

$$P_{cb} = P_{cs} = P(h_{cs}, s_{cs}) \quad (51)$$

3.3. Speed of Sound of Wet Vapor

For wet-vapor working fluid, the flow in the nozzle throat is in two-phase state. Although droplets formed in the throat are so less, they obviously affect the speed of sound of the flow and the ejector performance. Khaled Ameer et al. [22] proposed relations to calculate two-phase speed of sound. They also conducted comparisons with several available methods on two-phase speed of

sound and found that Wood's equation [23] provided relatively good predictions. Tao et al. [24] also found that Wood's equation agrees well with experimental data [25]. Here the Wood's equation is introduced in our models which is

$$a^2 = \frac{1}{\rho} \frac{1}{\left(\alpha / (\rho_v a_v^2) + (1 - \alpha) / (\rho_l a_l^2) \right)} \quad (52)$$

where a_l and a_v are the speed of sound of saturated liquid and vapor, respectively, α is the void fraction of vapor; ρ_l and ρ_v are the densities of saturated liquid and vapor, respectively; ρ is the density of two-phase vapor, it is:

$$\rho = \rho_v \alpha + \rho_l (1 - \alpha) \quad (53)$$

It is assumed that: (1) the gas phase medium and the liquid phase medium are uniformly mixed together; (2) it is an adiabatic process; (3) there is no surface tension and the pressure of liquid phase is equal to that of vapor phase. Then the wood's equation can be deduced as follows:

The bulk modulus of elasticity of the fluid K is the ratio between stress σ and strain ε :

$$K = \frac{\sigma}{\varepsilon} \quad (54)$$

The stress and strain meet the following relationship in the two-phase vapor:

$$\sigma_v = \sigma_l = \sigma \quad (55)$$

$$\varepsilon = \varepsilon_v \alpha + \varepsilon_l (1 - \alpha) \quad (56)$$

The bulk modulus of elasticity follows:

$$K = \rho a^2 \quad (57)$$

Based on Equations (54)–(57), the Wood Equation (52) can be derived.

3.4. Model over Entire Operational Range

As discussed above, the critical point model and breakdown point model are basic models for the model over the entire operational range. We developed the critical point model by modifying the model of Cardemil and Colle [13]. Moreover, we proposed a novel breakdown point model for both dry and wet vapor based on the constant-disturbing assumption. Then, the critical and breakdown points models can be integrated with linear characteristic Equations (1) and (3) and the boundaries Equations (2) and (4) to form the novel model for the entire operational range.

4. Experimental Verification

To obtain more accurate prediction performance, the ejector component efficiencies should be determined carefully since they dramatically influence the validity of an ejector model [26] and the values of the efficiencies depend on the ejector geometry, working fluid and operation conditions [8,9]. In this paper, five efficiencies, η_p , η_s , η_m , ϕ_m and η_d are introduced in critical point model. ϕ_m is used to consider the losses during the mixing of the primary and secondary flows, and η_p , η_m , η_s , η_d are the isentropic efficiencies of the primary flow from inlet to the throat of the nozzle and from the exit of the nozzle to section y-y of the ejector, the efficiency of the secondary flow from suction to hypothetical throat, and the efficiency of the mixed flow in diffuser, respectively. Even though they have physical meaning, they act as a correct factors in generally cases [27]. Thus, they should be determined based on experimental data.

We will employ the experiment data for R134a [28], R141b [8] and R245fa [1] ejectors to determine the efficiencies and verify our models. The physical parameters, safety and environmental data of these

refrigerants are listed in Table 1. R141b is viewed as a higher efficiency refrigerant for an ERS, but it will be phased out due to its ozone depletion potential (ODP). R134a was widely used in the refrigeration systems as an alternative environment friendly refrigerant from point of ODP, toxicity and flammability. Although the US Environmental Protection Agency (EPA) announced the timetable for eliminating the use of it and it will be limited to use according to the EU Regulation 517/2014 due to its high global warming potential (GWP), it can be used in other countries. R245fa is a dry refrigerant, its properties in ODP and GWP are similar with those of R134a, but its applications are less than R134a as its safety class is in B1.

Table 1. Thermophysical prosperities, safety and environmental data of the tested working fluid.

Refrigerant	Molecular Mass	Boiling Point (°C)	Critical Temperature (°C)	Critical Pressure (MPa)	Safety Group	ODP	GWP
R134a	102.03	−26.1	101.1	4.06	A1	0	1370
R141b	116.95	32.0	204.4	4.21	A2	0.12	717
R245fa	134.05	15.1	154.0	3.65	B1	0	1050

A program was developed to calculate the ejector performance over the entire operational range. The flowchart is shown in Figure 5, where the sub-flowcharts for calculating the speed of sound and the parameters of shock wave are shown in Figure 6. The properties of refrigerants are calculated by REFPROP Version 9.0 [20]. EOC analysis method proposed in [17] was employed to analyze the influence of the efficiencies. From [17], larger absolute value of average EOC_u indicates that the efficiency has larger influence on the prediction accuracy of u_c . The same applies to EOC_{pc} and the prediction accuracy of P_{cc} . Based on the experimental data of R134a ejectors in [28], the absolute values of average EOC_u for η_p , η_s , η_m , ϕ_m and η_d are 2.0, 1.11, 0.13, 0, 0, respectively; and those of average EOC_{pc} for η_p , η_s , η_m , ϕ_m and η_d are 0.48, 0.20, 0.12, 0.86, 0.11, respectively. It can be found that, in terms of prediction u_c , the efficiencies η_p and η_s have more influence than the other efficiencies, and, for P_{cc} , the efficiencies η_p and ϕ_m have more impact than the other efficiencies. For R141b, the effects of the efficiencies are similar with those for R134a. Hence, the efficiencies η_m and η_d are taken as constant since entrainment ratio and critical condensing pressure are not sensitive to them. The other efficiencies were determined by sparsity-enhanced optimization [18] based on the foregoing experiment. For the ejector model at breakdown point, three component efficiencies are used, η_p and η_d are taken as those in the critical point model due to the same choking phenomenon and compression process, respectively, and η_{mb} will be expressed as correlation equations as breakdown condensing pressure is sensitive to it.

Figure 6. Sub-flowchart for calculation of the speed of sound and the parameters of shock wave.

4.1. R134a Ejector

R134a is a typical wet-vapor refrigerant. The experimental data [28] are taken to validate the ejector model using R134a as a working fluid. An experimental ejector consists of two parts: the nozzle and the ejector body, which includes suction chamber, constant-area section and diffuser. Thus, nozzles and ejector bodies can interchange. The throat and exit diameters are 2.5 and 3.3 mm for nozzle A, and 2.09 and 2.7 mm for nozzle B, respectively. The diameters of mixing chambers A and B are 4.16 and 3.81 mm, respectively. The efficiency η_m is taken as 0.88 as that in [8] and η_d is adopted as 0.95 as that in [13]. Based on 20 cases of experimental data at critical points, the other efficiencies are derived through sparsity-enhanced optimization:

$$\eta_t = 0.8287 - 0.075 P_r / A_r + 0.155 / A_r \quad (58)$$

$$\eta_s = 1.0036 - 0.99 / (ArPr) \quad (59)$$

$$\phi_m = 0.9808 - 0.5865 / (ArPr) \quad (60)$$

These correlations of efficiencies should be used within the following boundaries:

$$2.77 \leq Ar \leq 3.96, 4.84 \leq Pr \leq 7.06 \quad (61)$$

The comparisons between the theoretical prediction results and the experiment data are shown in Table 2. It can be seen that the theoretical results of critical condensing pressures P_{cc} coincide fairly well with the measured P_{cc} , most of the errors are within $\pm 3\%$ and the maximum prediction error is 3.71%. For critical entrainment ratios u_c , most of them are within $\pm 6\%$ deviation in comparison with the measured u_c and the maximum error is 6.97%. The estimated speed of sound of the primary flow at nozzle throat, a_t , is also reported in Table 2. It can be observed that a_t mainly relates to the saturated temperature t_g and varies slightly with t_e and A_r . This is because that the speed of sound

relates to P_t , h_t and η_t as indicate in Equation (5) to (10), and the efficiency η_t varies with A_r and P_r as indicated in Equation (58).

Table 2. Comparisons between critical point model and the experiment results for R134a ejectors.

A_3/A_t	T_{p0} (°C)	T_{s0} (°C)	Theory P_{cc} (MPa)	Experiment P_{cc} (MPa)	Errors (%)	Theory u_c	Experiment u_c	Errors (%)	a_t (m/s)
AA Ejector									
2.77	75	10	0.8969	0.885	1.35	0.0844	0.0836	1.01	137.2
2.77	75	12.5	0.9203	0.8974	2.55	0.1149	0.1224	−6.14	137.0
2.77	75	15	0.9426	0.9351	0.80	0.1472	0.1561	−5.72	136.7
BB Ejector									
3.32	75	10	0.8461	0.8553	−1.08	0.1929	0.1933	−0.19	136.9
3.32	75	12.5	0.8685	0.8655	0.35	0.2325	0.2209	5.24	136.7
3.32	75	15	0.8906	0.8808	1.11	0.2749	0.257	6.97	136.5
3.32	80	10	0.9071	0.9278	−2.23	0.1402	0.1411	−0.65	134.8
3.32	80	12.5	0.9324	0.9354	−0.32	0.1749	0.1757	−0.45	134.5
3.32	80	15	0.9568	0.9598	−0.31	0.2118	0.2055	3.08	134.2
3.32	85	12.5	0.9964	1.0049	−0.85	0.1235	0.1172	5.35	131.9
3.32	85	15	1.024	1.035	−1.06	0.1562	0.1532	1.97	131.6
BA Ejector									
3.96	75	10	0.7984	0.7994	−0.12	0.3195	0.3248	−1.64	136.6
3.96	75	12.5	0.8204	0.8074	1.62	0.3694	0.3744	−1.32	136.4
3.96	75	15	0.8431	0.8129	3.71	0.4237	0.4446	−4.71	136.3
3.96	80	10	0.8555	0.8649	−1.08	0.2522	0.2635	−4.29	134.4
3.96	80	12.5	0.8797	0.8818	−0.23	0.2966	0.3177	−6.63	134.2
3.96	80	15	0.9039	0.8855	2.07	0.3442	0.3546	−2.94	134.0
3.96	85	10	0.9134	0.9453	−3.37	0.193	0.206	−6.31	131.8
3.96	85	12.5	0.9401	0.9647	−2.55	0.2317	0.2381	−2.68	131.5
3.96	85	15	0.9667	0.9648	0.19	0.2739	0.2782	−1.53	131.3

For the breakdown point, the choking of the primary flow is the same as that at critical point. Thus, the ejector component efficiency η_p is taken as that at critical point, namely, $\eta_d = 0.95$. Based on 12 cases at breakdown points, the optimized result of ϕ_{mb} and its boundaries are:

$$\phi_{mb} = 0.709 + 0.041(Pr / Ar) \quad (62)$$

$$2.77 \leq Ar \leq 3.96, 4.84 \leq Pr \leq 7.06 \quad (63)$$

The theoretical values of breakdown condensing pressure versus experimental data for breakdown points are shown in Figure 7. It can be observed that the theoretical values of breakdown condensing pressure are within $\pm 2\%$ deviation from the experimental data.

For the entire operational range, the theoretical and experimental performances in terms of condensing pressures are illustrated in Figure 8. It is obviously that the theoretical u_c coincide fairly well with the experimental values, and deviations are within $\pm 12\%$ over the entire operational range.

Figure 7. Comparisons of theoretical breakdown condensing pressures with experimental values using R134a as a working fluid.

Figure 8. Comparisons of theoretical entrainment ratios with experimental values using R134a as a working fluid.

4.2. R141b Ejector

In the experiment of Huang et al. [8], they tested the performance of 11 ejectors at critical points. The area ratios A_r of ejectors range from 6.44 to 10.64, the pressures of the primary flow and the secondary flow vary from 0.400 MPa to 0.604 MPa and from 0.040 MPa to 0.047 MPa, respectively.

As we have mentioned before, the efficiencies η_m and η_d are taken as 0.88 and 0.95, respectively. The other efficiencies are optimized based on the experimental data and the methods in [18]. They are:

$$\eta_t = 0.803 - 0.091(P_r/A_r) + 0.25(A_r/P_r) \quad (64)$$

$$\eta_s = 0.9288 + 0.5661(A_r/P_r) \quad (65)$$

$$\phi_m = 0.9232 + 0.0591(P_r/A_r) - 0.08(A_r/P_r) \quad (66)$$

These correlations of efficiencies should be used within the following boundaries:

$$6.44 \leq A_r \leq 10.64, 8.48 \leq P_r \leq 15.13 \quad (67)$$

The theoretical results of ejector performance are compared with the experimental data as shown in Table 3. As can be seen, most of theoretical results of critical condensing pressure P_{cc} are within $\pm 4\%$ deviation in comparison with the measured values and the maximum error is 6.73%. For critical entrainment ratio u_c , the maximum error is 7.70%. The estimated speed of sound of the primary flow at nozzle throat, a_t , is summarized in Table 3. It can be seen that a_t relates to t_g , t_e and A_r although the variations of a_t for R141b are less than those for R134a.

Table 3. Comparisons between critical point model and the experiment results for R141b ejectors.

T_{p0} (°C)	T_{s0} (°C)	Experiment P_{cc} (MPa)	Present P_{cc} (MPa)	Errors (%)	Experiment u_c	Huang [8] u_c	Present u_c	Huang [8] Error (%)	Present Error (%)	a_t (m/s)
AA Ejector										
95	8	0.1424	0.1404	−1.39	0.1859	0.1554	0.1769	−16.43	−4.83	154.9
90	8	0.1287	0.1271	−1.25	0.2246	0.2156	0.2169	−3.99	−3.43	154.4
84	8	0.1147	0.1124	−2.03	0.288	0.288	0.2745	0.23	−4.68	153.8
78	8	0.1027	0.0991	−3.53	0.3257	0.3525	0.3442	8.24	5.68	153.2
95	12	0.1442	0.1438	−0.27	0.235	0.2573	0.234	9.49	−0.44	154.4
90	12	0.1312	0.1303	−0.72	0.2946	0.3257	0.2824	10.54	−4.14	154.0
84	12	0.1167	0.1154	−1.15	0.3398	0.4147	0.352	22.04	3.59	153.5
AB Ejector										
90	8	0.1229	0.1203	−2.12	0.2718	0.2093	0.2551	−22.99	−6.13	154.2
84	8	0.1071	0.1065	−0.59	0.3117	0.3042	0.319	−2.39	2.33	153.6
78	8	0.0912	0.0939	3.00	0.3922	0.4422	0.3956	12.74	0.86	153.0
AG Ejector										
95	8	0.1275	0.1242	−2.58	0.2552	0.2144	0.2573	−15.98	0.80	154.4
90	8	0.1196	0.1125	−5.97	0.304	0.2395	0.3071	−21.22	1.02	154.0
84	8	0.102	0.0996	−2.35	0.3883	0.3704	0.3783	−4.61	−2.56	153.4
78	8	0.0897	0.088	−1.88	0.4393	0.4609	0.464	4.93	5.62	152.9
95	12	0.1279	0.1273	−0.46	0.3503	0.3434	0.3291	−1.97	−6.06	154.0
90	12	0.1167	0.1154	−1.08	0.4034	0.4142	0.3887	2.67	−3.65	153.6
84	12	0.1023	0.1025	0.21	0.479	0.4769	0.4729	12.09	−1.27	153.2
78	12	0.0901	0.0908	0.82	0.6132	0.6659	0.5733	8.60	−6.51	152.6
AC Ejector										
95	8	0.1179	0.1185	0.49	0.2814	0.2983	0.292	6.01	3.75	154.2
90	8	0.1079	0.1073	−0.53	0.3488	0.3552	0.3461	1.84	−0.76	153.8
84	8	0.095	0.0952	0.17	0.4241	0.4605	0.4233	8.58	−0.19	153.3
78	8	0.0816	0.0841	3.08	0.4889	0.5966	0.5144	22.03	5.21	152.8
AD Ejector										
95	8	0.1071	0.1086	1.43	0.3457	0.3476	0.3625	0.56	4.87	153.9
90	8	0.0988	0.0985	−0.27	0.4446	0.4178	0.4245	−6.02	−4.52	153.6
84	8	0.0856	0.0875	2.20	0.5387	0.5215	0.5126	−3.19	−4.84	153.1
78	8	0.0726	0.0775	6.73	0.6227	0.6944	0.6169	11.51	−0.94	152.6
95	12	0.1107	0.1116	0.84	0.4541	0.4708	0.4527	3.67	−0.31	153.6
90	12	0.1008	0.1015	0.65	0.5422	0.5573	0.5255	2.78	−3.07	153.3
84	12	0.089	0.0903	1.48	0.635	0.6906	0.6283	8.75	−1.05	152.8
78	12	0.0772	0.0803	3.97	0.7412	0.8626	0.7494	16.37	1.11	152.4
EG Ejector										
95	8	0.1377	0.1366	−0.83	0.2043	0.1919	0.1991	−6.06	−2.52	154.8
EC Ejector										
95	8	0.1283	0.1303	1.54	0.2273	0.2078	0.2294	−8.57	0.91	154.6
95	12	0.1304	0.1334	2.33	0.304	0.3235	0.296	6.41	−2.64	154.1
ED Ejector										
95	8	0.1212	0.1194	−1.45	0.2902	0.2658	0.2911	−8.39	0.32	154.2
EE Ejector										
95	8	0.1095	0.1111	1.44	0.3505	0.3253	0.3482	−7.20	−0.67	154.0
95	12	0.1095	0.1141	4.20	0.4048	0.4894	0.436	10.55	7.70	153.6
EF Ejector										
95	8	0.1047	0.106	1.24	0.3937	0.3774	0.3892	−4.13	−1.15	153.8
95	12	0.1051	0.109	3.68	0.4989	0.5482	0.4835	9.89	−3.09	153.5
EH Ejector										
95	8	0.0981	0.1004	2.37	0.4377	0.4627	0.4401	5.70	0.55	153.7

4.3. R245fa Ejector

The experimental results of Shestopalov et al. [1] were used to validate our model over the entire operational range. In the experiment, R245fa was used as a working fluid, and 9 ejectors for 11 critical-point cases and 1 ejector for 3 cases over the entire operational range were tested, respectively.

As those for R141b, the efficiencies η_m and η_d are adopted as 0.88 and 0.95 in critical point model, respectively. The other efficiencies were obtained through optimizing them to fit the experiment data. They are:

$$\eta_t = 0.88 - 0.0781(P_r/A_r) + 0.136(A_r/P_r) \quad (68)$$

$$\eta_s = 0.9443 + 0.0095(P_r/A_r) \quad (69)$$

$$\phi_m = 0.9072 + 0.0381(P_r/A_r) - 0.037(A_r/Pr) \quad (70)$$

The boundaries for these correlations of efficiencies are:

$$7.25 \leq Ar \leq 12.89, 10.64 \leq Pr \leq 14.79 \quad (71)$$

The model validation for critical condensing pressure P_{cc} and u_c are illustrated in Figure 9a,b, respectively. As shown in the figures, both P_{cc} and u_c agree fairly well with the experimental results. More specifically, the prediction errors of P_{cc} and u_c are within $\pm 2\%$ and $\pm 5\%$, respectively.

Figure 9. Comparisons of theoretical values of critical condensing pressure and critical entrainment ratios with experimental data using R245fa as a working fluid.

For the breakdown point model, the efficiencies η_t and η_d are taken as those in critical point model. The efficiency ϕ_{mb} was derived by fitting the calculated breakdown condensing pressures P_{cb} to the experimental values in [1]. The obtained optimum correlation and its boundaries are:

$$\phi_{mb} = 0.701 + 0.045(P_r/A_r) - 0.1253(A_r/P_r) \quad (72)$$

$$Ar = 9.56, 10.64 \leq Pr \leq 14.79 \quad (73)$$

For the entire operational range, the experimental and calculated entrainment ratios u in terms of condensing temperatures t_c are plotted in Figure 10. It can be observed that the prediction values coincide with experimental results very well. Further observation finds that the prediction accuracy of breakdown condensing temperatures t_{cb} at breakdown point is higher than that of critical entrainment ratios u_c and critical condensing temperatures t_{cc} . In fact, the prediction errors of t_{cb} are within $\pm 0.5\%$.

Figure 10. Comparisons of theoretical entrainment ratios with experimental values using R245fa as a working fluid.

5. Results and Discussion

For wet vapor refrigerant, droplets will form in the nozzle when the primary flow expands through the nozzle. However, for dry vapor refrigerant, the vapor is in superheated state in the nozzle. Droplets in nozzle will affect ejector performance. In this paper, wet vapor refrigerant R134a and dry vapor refrigerants R141b and R245fa are selected to perform the comparison. A R134a ejector and a R141b ejector were designed to meet the same design conditions at critical point; more specifically, cooling capacity $Q_e = 1.5$ kW, saturated generating temperature $t_g = 85$ °C, saturated evaporating temperature $t_e = 10$ °C and critical condensing temperature $t_{cc} = 36$ °C. The key geometric parameters of the R134a ejector are: $D_t = 2.491$ mm, $D_{p1} = 3.177$ mm, $D_3 = 4.96$ mm. For the R141b ejector, $D_t = 4.261$ mm, $D_{p1} = 7.80$ mm, $D_3 = 10.865$ mm. To simplify the analysis, it is assumed that the losses from the generator and the evaporator to the ejector are ignored, and the inlet pressures P_{p0} and P_{s0} are corresponding to saturated temperatures t_g and t_e , respectively.

Figure 11 shows the critical entrainment ratio u_c versus the superheating degree ΔT_p of the primary flow. It can be observed that for the R134a ejector, u_c is lower than that for the R141b ejector. For the R141b ejector, u_c increases slightly and uniformly with the increasing of ΔT_p , but for the R134a ejector, u_c increases more rapidly at low ΔT_p than at high ΔT_p , with the inflection point being at about $\Delta T_p = 8$ °C. Since $u_c = m_p/m_s$, the increasing of u_c may be induced by the increasing of m_s and/or the decreasing of m_p .

Figure 11. Entrainment ratio versus the primary superheating ΔT_p .

The flow rates m_p and m_s vs. ΔT_p are plotted in Figure 12a,b, respectively, and A_{sy}/A_{py} vs. ΔT_p is shown in Figure 13. As can be seen, the increasing rate of m_s , the decreasing rate of m_p and increasing rate of A_{sy}/A_{py} of the R134a ejector are all lessened after ΔT_p grows higher than 8 °C. The inflection point of these curves are all at $\Delta T_p = 8$ °C. According to the calculation results of the vapor quality, R134a is in wet vapor state at section y-y as $\Delta T_p \leq 8$ °C; after that, it changes into superheated vapor. Thus, the inflection of the variations rate of m_p , m_s , A_{sy}/A_{py} and the resultant change of u_c are induced by the phase change of the primary flow at section y-y.

Figure 12. The flow rates of the primary and secondary flows versus the primary superheating ΔT_p .

Figure 13. Area ratio A_{sy}/A_{py} versus the primary superheating ΔT_p .

For the R141b ejector, the variation rates of u_c , m_p , m_s and A_{sy}/A_{py} are less than those for the R134a ejector, and there is no significant change in the variation rate as shown in Figure 11 to Figure 13. The main reason is that R141b stays at the superheated-vapor state rather than the wet vapor state as ΔT_p is changed. Thus, superheating R134a before it enters the ejector has much higher influence than superheating R141b.

Figure 14 shows the cooling capacity Q_e over the primary superheating ΔT_p . It can be found that Q_e has the same variation trend as m_s since the enthalpies at the inlet and the outlet of the evaporator have no variation. Superheating the primary flow is helpful in increasing Q_e for a R134a ejector (Q_e increased by 10% at $\Delta T_p = 20^\circ\text{C}$). However, for a R141b ejector, Q_e shows no change.

The coefficient of performance (COP) is often employed in evaluating the performance of a refrigeration system. For an ERS, it is defined as the ratio of cooling capacity Q_e to energy input which includes generating heat Q_g and pump power W_p . Generally, since W_p is much less than Q_g , it has less influence on COP. Figure 15 depicts COP over the superheating temperature ΔT_p . For R134a, COP has no significant variation with ΔT_p , although Q_e increases with the increasing of ΔT_p as shown in Figure 14. This is because superheating the primary flow will cost more energy. In other words, the increasing rate of Q_g is close to that of Q_e . For R141b, COP decreases with rise

in ΔT_p . This lies in that superheating the primary flow costs more energy, whereas, the cooling capacity Q_e is almost constant.

Figure 14. Cooling capacity versus the primary superheating ΔT_p .

Figure 15. The coefficient of performance COP versus the primary superheating ΔT_p .

Figure 16 illustrates the critical condensing temperature t_{cc} v.s. the primary superheating ΔT_p . It can be seen that for the R134a ejector, t_{cc} has a slight increase when ΔT_p increases, but for the R141b ejector, t_{cc} remains almost constant. The difference may be mainly induced by the different variation of v_t , which equals to the speed of sound at the nozzle. The calculation results indicate that v_t increases by 14% and 4% for the R134a ejector and the R141b ejector, respectively, when ΔT_p changes from 0 to 20 °C. Thus, the increment of t_{cc} for the R134a ejector is larger than that for the R141b ejector. When ΔT_p varies from 0 to 20 °C, the increments of the outlet temperature of the ejector are about 26 °C and 17 °C for the R134a ejector and R141b ejector, respectively. It should be noticed that the above t_{cc} is not the outlet temperature of the diffuser but the saturated temperature which is corresponding to the critical condensing pressure P_{cc} .

Figure 16. Critical condensing temperature versus the primary superheating ΔT_p .

Besagni et al. [26] studied the optimum ejector performance, which is obtained by adjusting area ratio A_r to a suitable value in relation to the variations of generating, evaporating and condensing temperature. This is significant for the design of an ERS. However, for operating an ERS, it seems more helpful to find the coupled influence of operating conditions and superheating on the performance of the ejector at fixed A_r .

Figure 17 illustrates the variations of ejector performance with saturated generating temperature t_g using R134a as a working fluid. It can be seen that, for the same t_g , the critical entrainment ratio u_c at $\Delta T_p = 10$ °C is larger than that at $\Delta T_p = 0$ °C, but COPs are almost the same at both $\Delta T_p = 0$ °C and $\Delta T_p = 10$ °C. As those have indicated in Figures 11 and 14, except the rise in u_c , superheating the primary flow can also cause the rise in cooling capacity Q_e . In terms of critical condensing temperature t_{cc} , superheating the primary flow can increase it slightly when $t_g > 85$ °C as shown in Figure 17b. Thus, for wet-vapor R134a, it can be concluded that superheating the primary flow has insignificant influence on COP, but can increase the cooling capacity Q_e . In fact, further observation on Figure 17 finds that the effect of the change of t_g is much more significant than that of superheating the primary flow. Namely, COP and u_c both increase sharply with the decrease of t_g . Therefore, for a given ERS, decreasing the saturated generating temperature t_g can result in significant increase in both u_c and COP when the actual condensing temperature t_c is lower than t_{cc} .

Figure 17. Critical entrainment ratio, COP and critical condensing temperature versus generating temperature using R134a as a working fluid.

Figure 18 depicts the ejector performance v.s. the saturated generating temperature t_g using R141b as a working fluid. It can be observed that when $\Delta T_p = 10$ °C, the critical entrainment ratio u_c has a diminutive rise and COP has a slight decrease relative to those at $\Delta T_p = 0$ °C, but t_{cc} has hardly any variation when the primary flow is superheated at $\Delta T_p = 10$ °C. Referring to Figures 11 and 14, it can be deduced that Q_e can not increase for each t_g in the range from 70 to 95 °C. Thus, for an ERS using dry-vapor refrigerant R141b, superheating primary flow could not improve its performance.

Figure 18. Critical entrainment ratio, COP and critical condensing temperature versus generating temperature using R141b as a working fluid.

Figure 19 shows the ejector performance v.s. the saturated evaporating temperature t_e using R134a as a working fluid. For each t_e , the critical entrainment ratio u_c and critical condensing temperature t_{cc} at $\Delta T_p = 10$ °C are both larger than those at $\Delta T_p = 0$ °C, but COP are almost the same for the two conditions. Referring to Figures 11 and 14, it can be deduced that Q_e can be larger at $\Delta T_p = 10$ °C than that $\Delta T_p = 0$ °C for each t_e . Thus, for saturated generating temperature $t_g = 85$ °C, superheating primary flow can lead to some increase in Q_e and t_{cc} of an ERS using R134a as a working fluid. However, COP, u and t_{cc} all rapidly increase with rise in t_e . This means increasing of t_e has more significant influence on the performance of an ERS than superheating the primary flow. However, it is certain that for each t_e , superheating the primary flow can further improve the cooling capacity Q_e .

Figure 19. Critical entrainment ratio, COP and critical condensing temperature versus evaporating temperature using R134a as a working fluid.

Figure 20 illustrates the ejector performance v.s. the evaporating temperature t_e using R141b as a working fluid. It is found that for the ejector with fixed A_r , the critical entrainment ratio u_c at $\Delta T_p = 10^\circ\text{C}$ is a little larger than that at $\Delta T_p = 0^\circ\text{C}$ and COP is slightly lower than that at $\Delta T_p = 0^\circ\text{C}$ for each t_e . As shown in Figure 20b, t_{cc} has hardly any variation when ΔT_p changes from 0 to 10°C . Thus, superheating the primary flow has little influence on the performance of an ERS using R141b as a working fluid. However, it can be observed that the performance, namely u , COP and t_{cc} , of an ERS can be obviously improved with a rise in t_e .

Figure 20. Critical entrainment ratio, COP and critical condensing temperature versus evaporating temperature using R141b as a working fluid.

For the breakdown point model, R134a and R245fa are used as working fluids to analyze the ejector performance vs. the primary superheating ΔT_p . Figure 21a,b show the breakdown condensing temperature t_{cb} in terms of the primary superheating ΔT_p for a R134a ejector and a R245fa ejector, respectively. t_{cb} is corresponding to the breakdown condensing pressure P_{cb} . It can be observed that t_{cb} increases with the rise in ΔT_p . For the R134a ejector, t_{cb} has up to 2% rise at $\Delta T_p = 10^\circ\text{C}$, but for a R245fa ejector, the increasing of t_{cb} is insignificant. This is mainly because the variation rate of sound speed in the nozzle throat is different for the two ejectors. It is known that the speed of sound sharply increases as the working fluid changes from wet vapor to dry saturated vapor. Hence, the increasing rate of the sound speed in the nozzle throat of the R134a ejector, is larger than that of the R245fa ejector since R134a changes from two-phase state to vapor state as ΔT_p rises from 0 to 10°C , but for R245fa, it maintains at superheating vapor state. Further observations indicate that the increasing rate is similar for different t_e . This indicates that t_e has no significant influence on the variation rate of t_{cb} when ΔT_p is changed.

Figure 21. Breakdown condensing temperature versus the primary superheating ΔT_p for R134a and R245fa ejectors.

6. Conclusions

This paper proposed a novel model for the performance predictions of dry and wet vapor ejectors over the entire operational range. The model was obtained by integrating the linear characteristic equations of ejector with critical and breakdown points models, which were developed based on the assumptions of constant-pressure mixing and constant-pressure disturbing. To improve the accuracy of the models, equations of the real gas property and the two-phase speed of sound were introduced and the ejector component efficiencies were determined through EOC analysis and sparsity-enhanced optimization. The theoretical results of the model coincided fairly well with the experimental data for both dry and wet refrigerants over the entire operational range.

It was also found that superheating the primary flow before it enters the ejector can improve the ejector performance. For the R134a ejector, when the primary superheating ΔT_p changed from 0 to 20 °C, the critical entrainment ratio u_c and cooling capacity Q_e could be increased by 19% and 10%, respectively. For the R141b ejector with the same design conditions as the R134a ejector, u_c increased insignificantly with the enlarging ΔT_p . For the breakdown point, the condensing temperature t_{cb} had some rise when ΔT_p increased. For both wet and dry working fluids, critical entrainment ratios u_c , coefficients of performance COP and critical condensing temperatures t_{cc} could be improved significantly with rising in saturated evaporating temperature t_e . However, for a given ERS, decreasing the saturated generating temperature t_g can result in significant increase in both u_c and COP when the actual condensing temperature decreases. If the primary flow is superheated together with the increase of t_e or the decrease of t_g , Q_e and u_c can be further improved for wet working fluid R134a.

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Abbreviations

The following abbreviations are used in this manuscript:

COP	the coefficient of performance
CRS	compressor-based refrigeration system
EOC	the effect of the change (of efficiency)
ERS	ejector refrigeration system

Nomenclature

a	speed of sound (m/s)
A	area (m ²)
d	diameter (m)
h	enthalpy (kJ/kg)
K	bulk modulus of elasticity (N/m ²)
m	mass flow rate (kg/s)
P	pressure (MPa)
s	entropy (kJ/kg K)
t	temperature (°C)

T	temperature (K)
u	entrainment ratio
v	velocity (m/s)
α	void fraction of vapor
ε	strain
η	efficiency relating to isentropic efficiency
ρ	density (kg/m ³)
σ	stress (N/m ²)
ϕ	efficiency account for losses
ω	infinitesimal
Subscripts	
b	breakdown
c	condensation, critical point
d	diffuser
e	expansion, evaporation
g	generation
l	liquid
m	mixing flow
p	primary flow
p_0	primary flow at inlet of ejector
p_1	primary flow at nozzle exit
r	ratio
s	secondary flow, isentropic process
s_0	secondary flow at inlet of ejector
t	throat
v	vapor
y	position of the hypothetical throat

References

1. Shestopalov, K.; Huang, B.; Petrenko, V.; Volovyk, O. Investigation of an experimental ejector refrigeration machine operating with refrigerant R245fa at design and off-design working conditions. Part 1. Theoretical analysis. *Int. J. Refrig.* **2015**, *55*, 201–211.
2. Umair, M.; Akisawa, A.; Ueda, Y. Performance evaluation of a solar adsorption refrigeration system with a wing type compound parabolic concentrator. *Energies* **2014**, *7*, 1448–1466.
3. Akiloroaya, V.; Samali, B.; Fakhar, A.; Pishghadam, K. A review of different strategies for HVAC energy saving. *Energy Convers. Manag.* **2014**, *77*, 738–754.
4. Keenan, J.H.; Neumann, E.P. A simple air ejector. *ASME J. Appl. Mech.* **1942**, *9*, A75–A81.
5. Keenan, J.H. An investigation of ejector design by analysis and experiment. *J. Appl. Mech.* **1950**, *17*, 299–309.
6. Munday, J.T.; Bagster, D.F. A new ejector theory applied to steam jet refrigeration. *Ind. Eng. Chem. Process. Des. Dev.* **1977**, *16*, 442–449.
7. Eames, I.W.; Aphornratana, S.; Sun, D.W. The jet-pump cycle: A low cost refrigerator option powered by waste heat. *Heat Recover. Syst. CHP* **1995**, *15*, 711–721.
8. Huang, B.J.; Chang, J.M.; Wang, C.P.; Petrenko, V.A. A 1-D analysis of ejector performance. *Int. J. Refrig.* **1999**, *22*, 354–364.
9. Besagni, G.; Mereu, R.; Chiesa, P.; Inzoli, F. An integrated lumped parameter-CFD approach for off-design ejector performance evaluation. *Energy Convers. Manag.* **2015**, *105*, 697–715.
10. Khennich, M.; Sorin, M.; Galanis, N. Exergy flows inside a one phase ejector for refrigeration Systems. *Energies* **2016**, *9*, 212.
11. Chen, W.; Shi, C.; Zhang, S.; Chen, H.; Chong, D.; Yan, J. Theoretical analysis of ejector refrigeration system performance under overall modes. *Appl. Energy* **2017**, *185*, 2074–2084.
12. Lear, W.E.; Parker, G.M.; Sherif, S.A. Analysis of two-phase ejectors with Fabri choking. *Arch. Proc. Inst. Mech. Eng. C-J. Mech.* **2002**, *216*, 607–621.

13. Cardemil Cardemil, J.M.; Colle, S. A general model for evaluation of vapor ejectors performance for application in refrigeration. *Energy Convers. Manag.* **2012**, *64*, 79–86.
14. Chen, Y.M.; Sun, C.Y. Experimental study of the performance characteristics of a steam-ejector refrigeration system. *Energy Convers. Manag.* **1997**, *15*, 384–394.
15. Xu, X.; Chen, G.; Tang, L.; Zhu, Z.; Liu, S. Experimental evaluation of the effect of an internal heat exchanger on a transcritical CO₂ ejector system. *J. Zhejiang Univ.-Sci. A* **2011**, *12*, 146–153.
16. Chen, W.; Liu, M.; Chong, D.; Yan, J.; Little, A.B.; Bartosiewicz, Y. A 1D model to predict ejector performance at critical and sub-critical operational regimes. *Appl. Therm. Eng.* **2013**, *36*, 1750–1761.
17. Li, F.; Tian, Q.; Wu, C.; Wang, X.; Lee, J.M. Ejector performance prediction at critical and subcritical operational modes. *Appl. Therm. Eng.* **2017**, *115*, 444–454.
18. Li, F.; Wu, C.; Wang, X.; Tian, Q.; Teo, K.L. Sparsity-enhanced optimization for ejector performance prediction. *Energy* **2016**, *113*, 25–34.
19. Chen, S.L.; Yen, J.Y.; Huang, M.C. Experimental investigation of ejector performance based upon different refrigerants. *ASHRAE Trans.* **1998**, *104*, 153–160.
20. Lemmon, E.; Huber, M.; McLinden, M. *NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP*, version 9.1; National Institute of Standards and Technology: Gaithersburg, MD, USA, 2010.
21. Anderson J. *Modern Compressible Flow*, 2nd ed.; McGraw-Hill: New York, NY, USA, 2002.
22. Ameur, K.; Aidoun, Z.; Ouzzane, M. Modeling and numerical approach for the design and operation of two-phase ejectors. *Appl. Therm. Eng.* **2016**, *109*, 809–818.
23. Wood, A.B. *A Textbook of Sound*, 3rd ed.; Neill and Co. Ltd: London, UK, 1957.
24. Tao, B.; Chen, D.H.; Che, C.X.; Wang, X.M. Study on the sound velocity in a gas-liquid flow. *J. Appl. Acoust.* **2015**, *34*, 373–376.
25. Henry, R.E.; Grolmes, M.A.; Fauske, H.K. *Pressure-Pulse Propagation in Two-Phase One- and Two-Component Mixtures*; ANL: Chicago, IL, USA, 1971.
26. Besagni, G.; Mereu, R.; Leo, G.D.; Inzoli, F. A study of working fluids for heat driven ejector refrigeration using lumped parameter models. *Int. J. Refrig.* **2015**, *58*, 154–171.
27. Varga, S.; Oliveira, A.C.; Diaconu, B. Numerical assessment of steam ejector efficiencies using CFD. *Int. J. Refrig.* **2009**, *32*, 1203–1211.
28. Li, F. Study on Ejectors and a Solar Ejector-Refrigeration System with Dual Energy-Storage Tanks and Multiple Ejectors. Ph.D. Thesis, Taiyuan University Technology, Taiyuan, China, 2016.

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