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Analysis and Comparison of Two-Phase Ejector Performance Metrics for R134a and CO_2 Ejectors

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ABSTRACT

Two-phase ejectors have been gaining increased attention in recent years due to their ability to directly improve the COP of the cycle. Of common interest in two-phase ejector studies is how the ejector improves cycle COP. However, less emphasis is often given to the performance of the two-phase ejector itself. The amount of COP improvement offered by an ejector cycle is very strongly influenced by the performance of the two-phase ejector; thus, it is important to understand the operation and performance of the two-phase ejector. Defining the performance of a two-phase ejector is not as straightforward as for an isentropic expander because there are multiple fluid streams in an ejector and because it is difficult to obtain flow properties at some locations in the ejector. As a result, there are a variety of performance metrics that have been proposed for use with ejectors in general and specifically for two-phase ejectors.

In the present study, several different metrics that have been proposed for measuring the performance of ejectors are presented and analyzed. Performance metrics that were originally proposed for single-phase ejectors as well as those proposed specifically for two-phase ejectors are both considered. A simple numerical ejector model is used to simulate ejector operation and calculate the performance of the ejector based on the various performance metrics. The various ejector performance metrics are then compared based on the numerical results. Experimental data for R134a and CO_2 two-phase ejectors is also presented, and the ejector performance metrics are compared based on the available experimental data as well. It is seen that R134a and CO_2 offer somewhat similar ejector performance, though the CO_2 ejector does seem to have noticeably better performance. CO_2 has significantly higher throttling loss than R134a, meaning that there is more work available for the two-phase ejector to recover with CO_2 and larger potential COP improvement for a CO_2 cycle.

1. INTRODUCTION

A two-phase ejector is a device that is capable of recovering the expansion power that is generally lost during an isenthalpic expansion process in the conventional vapor-compression cycle. A two-phase ejector uses the expansion of a high-pressure liquid (motive stream) to entrain and increase the pressure of a low-pressure vapor (suction stream). In a refrigeration cycle, a two-phase ejector can be used to entrain and compress the vapor at the evaporator outlet, resulting in higher compressor suction pressure and improved cycle performance. The layout of the standard two-phase ejector cycle and its representation on a pressure-specific enthalpy diagram are shown in Figure 1.

CO_2 has very high throttling loss, in the range of temperatures commonly associated with air conditioning cycles, compared to most other common refrigerants, and as a result, the use of an ejector with CO_2 offers the potential for very large COP improvement. Elbel and Hrnjak (2008) observed simultaneous COP and capacity improvements of 7 and 8 %, respectively, with a CO_2 two-phase ejector cycle. Similar but higher COP improvement was observed at matched capacity by Nakagawa *et al.* (2011) and Lee *et al.* (2011); these studies observed 26 and 15 % COP improvement, respectively. Lower pressure fluids have less potential for improvement with two-phase ejector cycles; Lawrence and Elbel (2014) observed 5 % COP improvement with R134a and 6 % with R1234yf.

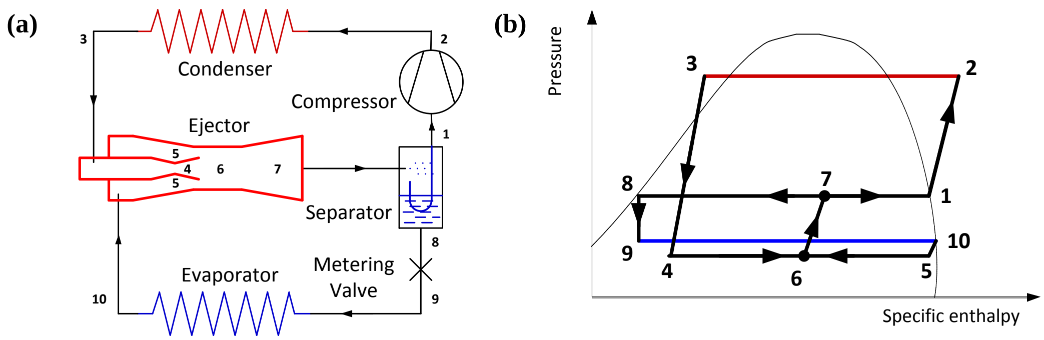

Figure 1: (a) Cycle layout and (b) pressure-specific enthalpy diagram of standard two-phase ejector cycle

Two-phase ejector studies commonly focus on the COP improvement that the ejector can provide to the cycle. However, less focus is often given to the performance of the ejector itself, despite the fact that ejector performance has a very strong impact on the COP of the cycle. Thus, it is important to understand how ejector performance is quantified. This paper focuses on several different performance metrics that have been proposed for quantifying the performance of a two-phase ejector. Ejector performance metrics are presented and analyzed qualitatively in terms of their ability to measure two-phase ejector performance. A numerical model of a two-phase ejector and experimental data from R134a and CO_2 ejectors are then used to compare the performance metrics quantitatively.

2. EJECTOR PERFORMANCE METRICS

2.1 Single-Phase Ejector Performance Metrics

There are several performance metrics that are commonly used when working with single-phase ejectors. Equations (1) and (2) show the mass entrainment ratio and the suction pressure ratio, respectively, which are often associated with measuring the performance of single-phase ejectors (Chunnanond and Aphornratana, 2004) but are also very useful for describing the performance of two-phase ejectors. The mass entrainment ratio is defined as the ratio of the suction mass flow rate to the motive mass flow rate; it is a direct measure of the amount of mass that the ejector can entrain. The suction pressure ratio is defined as the ratio of the diffuser outlet pressure to the suction inlet pressure; it is a direct measure of the pressure lift that the ejector can provide to the entrained stream.

$$\Phi_m = \frac{\dot{m}_{sn}}{\dot{m}_{mn}} \tag{1}$$

$$\Pi_s = \frac{P_{diff,out}}{P_{sn,in}} \tag{2}$$

The objective of using an ejector is to both entrain a fluid and to increase its pressure. However, the mass entrainment ratio and suction pressure ratio are each only able to describe one of these effects, and thus, in order to adequately describe two-phase ejector performance, it is necessary to consider both parameters at the same time. It would be more desirable to have a single performance parameter, such as an ejector efficiency, which could capture both the mass entrainment and the pressure lift effects at the same time. Efficiencies for ejectors are generally defined as a quantity representing actual power recovery over a quantity representing a maximum available power recovery. Such definitions may actually describe the effectiveness of the ejector, but the term ejector efficiency is more commonly found in the literature. An efficiency that is commonly used for single-phase ejectors and defined by ASHRAE (1983) is shown in Equation (3).

$$\eta_{ASHRAE} = \frac{(\dot{m}_{mn} + \dot{m}_{sn})}{\dot{m}_{mn}} \frac{h_{diff,out} - h_{sn,in}}{h_{mn,in} - h_{mn,out,isen}} \tag{3}$$

It is important to note that while defining an ejector efficiency is a convenient way to simultaneously capture both mass entrainment and pressure lift effects of the ejector, it cannot fully define the performance of the ejector in the way a set of two performance parameters, such as the entrainment and pressure ratios, is able to. Though they exit as a single stream of fluid at a single state, the two streams of fluid enter the ejector separately and undergo two different processes before becoming a single stream. The way two separate fluid streams undergo two different processes cannot be fully described by a single numerical value; thus, none of the ejector efficiencies that will be discussed in this paper are actually able to completely describe ejector performance by themselves. This is different from an expander device, which is the other commonly considered device for recovering expansion work. An expander involves only a single stream of fluid undergoing a single expansion process and can be fully described by a single value of isentropic efficiency.

2.2 Two-Phase Ejector Efficiency Definitions

An efficiency definition specifically intended for two-phase ejectors was proposed by Nakagawa and Takeuchi (1998) and is shown in Equation (4). This efficiency is defined as the energy gain associated with an isochoric pressure increase of both motive and suction streams between the suction inlet pressure and the diffuser outlet pressure, assuming that both streams start at the suction inlet state, less the initial kinetic energy of the suction stream, divided by the total enthalpy that would be supplied by the motive stream via an isentropic expansion from the motive inlet pressure to the mixing (motive outlet) pressure. The efficiency definition of Nakagawa and Takeuchi (1998) is similar to that of ASHRAE (1983), but energy gain associated with isochoric pressure increase has been used instead of specific enthalpy difference in the numerator. Note that this efficiency definition is essentially proportional to the quantity $(1 + \Phi_m)$, as the kinetic energy contribution is generally negligible compared to the pressure increase term.

$$\eta_{Nakagawa} = \frac{(\dot{m}_{mn} + \dot{m}_{sn}) \frac{P_{diff,out} - P_{sn,in}}{\rho_{sn,in}} - \dot{m}_{sn} \frac{{V_{sn,in}}^2}{2}}{\dot{m}_{mn}\left(h_{mn,in} - h_{mn,out,isen}\right)} \tag{4}$$

Another efficiency definition for two-phase ejectors was proposed by Butrymowicz *et al.* (2005) and is shown in Equation (5). This efficiency is defined as the isothermal compression power required by the suction stream to be compressed from the suction inlet pressure to the diffuser outlet pressure, assuming the suction fluid to be an ideal gas, divided by the expansion power provided by the motive stream as it expands from the motive inlet pressure to the diffuser outlet pressure, assuming the motive fluid density is constant throughout the expansion process, plus the kinetic energy of the motive flow at the motive nozzle outlet.

$$\eta_{Butrymowicz} = \Phi_m \frac{\frac{P_{sn,in}}{\rho_{sn,in}} \ln\left(\frac{P_{diff,out}}{P_{sn,in}}\right)}{\frac{P_{mn,in} - P_{diff,out}}{\rho_{mn,in}} + 0.5{V_{mn,out}}^2} \tag{5}$$

Elbel and Hrnjak (2008) proposed an efficiency for two-phase ejectors defined as the enthalpy consumed by an isentropic compression of the suction stream from the suction inlet pressure to the diffuser outlet pressure divided by the enthalpy supplied by an isentropic expansion of the motive stream from the motive inlet pressure to the diffuser outlet pressure, as shown in Equation (6). Note that the assumption of isentropic compression describes a limiting process and is essentially the minimum amount of power that could be recovered by the ejector; in reality, the compression process is not isentropic, meaning that additional power is required for the actual compression process. Thus, this definition of ejector efficiency can be thought of as a conservative estimate of power recovery efficiency, as it likely under predicts the actual power recovery efficiency. Note also that the efficiencies of Nakagawa and Takeuchi (1998) and Butrymowicz *et al.* (2005) both require knowledge of the mixing section pressure, which can be hard to determine experimentally due to the high-speed, two-phase flow and non-equilibrium effects in the mixing section. However, the efficiency of Elbel and Hrnjak (2008) does not require knowledge of the mixing section pressure.

$$\eta_{Elbel} = \frac{\dot{W}_{rec}}{\dot{W}_{rec,max}} = \Phi_m \frac{h_{s,out,isen} - h_{s,in}}{h_{m,in} - h_{m,out,isen}} = \Phi_m \frac{h\left(P_{diff,out}, s_{s,in}\right) - h_{s,in}}{h_{m,in} - h\left(P_{diff,out}, s_{m,in}\right)} \tag{6}$$

Ozaki *et al.* (2004) proposed an efficiency definition, shown in Equation (7), very similar to that of Elbel and Hrnjak (2008) except the expansion process was assumed to be from motive inlet pressure to mixing pressure. This difference will yield a higher amount of power available for recovery for the Ozaki *et al.* (2004) definition.

$$\eta_{Ozaki} = \Phi_m \frac{h\left(P_{diff,out}, s_{s,in}\right) - h_{s,in}}{h_{m,in} - h\left(P_{mix}, s_{m,in}\right)} \tag{7}$$

2.3 Analysis of Two-Phase Ejector Efficiencies

Köhler *et al.* (2007) derived the same efficiency definition as that of Elbel and Hrnjak (2008) as the product of an isentropic expander efficiency and an isentropic compressor efficiency. An isentropic efficiency for the hypothetical process of extracting work from the motive fluid with an expander operating between the motive inlet pressure and the diffuser outlet pressure is given in Equation (8). An isentropic efficiency for the hypothetical process of passing the suction fluid through a compressor operating between the suction inlet pressure and the diffuser outlet pressure is given in Equation (9). If the expander device is directly connected to the compressor, such that the power extracted from the motive fluid is the power provided to the suction fluid (assuming no losses while transferring power between components), then the product of the two isentropic efficiencies describes the efficiency of the overall process, shown in Equation (10).

$$\eta_{exp} = \frac{h_{m,in} - h_{m,out}}{h_{m,in} - h_{m,out,isen}} \tag{8}$$

$$\eta_{cp} = \frac{h_{s,out,isen} - h_{s,in}}{h_{s,out} - h_{s,in}} \tag{9}$$

$$\eta_{exp}\eta_{cp} = \frac{h_{m,in} - h_{m,out}}{h_{m,in} - h_{m,out,isen}} \frac{h_{s,out,isen} - h_{s,in}}{h_{s,out} - h_{s,in}} \tag{10}$$

If the power extracted by the expander is set equal to the power consumed by the compressor, shown in Equation (11), then the resulting expression can be used to find an expression for entrainment ratio, shown in Equation (12), and this expression for entrainment ratio can be substituted into Equation (10) to show that the expander-compressor efficiency is equal to the efficiency of Elbel and Hrnjak (2008), as shown in Equation (13).

$$\dot{m}_m\left(h_{m,in} - h_{m,out}\right) = \dot{m}_s\left(h_{s,out} - h_{s,in}\right) \tag{11}$$

$$\Phi_m = \frac{\dot{m}_s}{\dot{m}_m} = \frac{h_{m,in} - h_{m,out}}{h_{s,out} - h_{s,in}} \tag{12}$$

$$\eta_{exp}\eta_{cp} = \Phi_m \frac{h_{s,out,isen} - h_{s,in}}{h_{m,in} - h_{m,out,isen}} = \eta_{Elbel} \tag{13}$$

This hypothetical expander-compressor system makes sense as an analogy for an ejector, as the operating principle of the ejector is the exchange of power between an expanding stream and a stream being compressed. However, it may not initially be clear why the motive stream is only expanded to the diffuser outlet pressure in the above analysis and not to the mixing section pressure, as happens in a real ejector and is assumed in the efficiency of Ozaki *et al.* (2004). The extra power that is available because of the further expansion from diffuser outlet pressure to mixing pressure, theoretically, only serves to recompress the motive fluid from the mixing pressure back to the diffuser outlet pressure; only the power from the expansion of the motive fluid from inlet to diffuser outlet pressure can be transferred to the suction fluid and thus be considered useful power available for recovery.

All of the efficiencies presented above can be viewed as power recovery efficiencies, as they all represent some actual power gained during the compression process over some theoretical maximum power provided by the expansion process. Note, however, that the efficiencies of Elbel and Hrnjak (2008), Butrymowicz *et al.* (2005), and Ozaki *et al.* (2004) are all proportional to the entrainment ratio Φ_m while, as noted above, the efficiency of Nakagawa and Takeuchi (1998) is proportional to the quantity $(1+\Phi_m)$. The efficiency of Nakagawa and Takeuchi (1998) accounts for the power gain of the suction stream as well as the motive stream in the mixing section and diffuser (assuming both streams to begin at the suction inlet state), whereas the other three efficiency definitions only account for the power gain of the suction stream. As a result, the efficiency of Nakagawa and Takeuchi (1998) would be expected to consistently yield a higher value than the other three efficiencies. It is worth noting again that the objective of using an ejector is to transfer power from the motive stream to the suction stream; thus, it makes sense that the useful power gain in numerator of each of the efficiency definitions be that of just the suction stream.

Interestingly, it can also be shown that the efficiency definition of Elbel and Hrnjak (2008) can be viewed as a reversible entrainment ratio efficiency, defined as the actual entrainment ratio over the entrainment ratio that would be achieved by a reversible ejector for the same inlet and outlet pressures. This reversible entrainment ratio efficiency was originally proposed by Elrod (1945) and is shown in Equation (14).

$$\eta_{Elrod} = \frac{\Phi_m}{\Phi_{m,rev}} \tag{14}$$

To see how the efficiency of Elbel and Hrnjak (2008) is equivalent to that of Elrod (1945), the reversible entrainment ratio $\Phi_{m,rev}$ must first be calculated. If the expression for entrainment ratio in Equation (12) is modified to assume that the compression and expansion processes happen reversibly (isentropically), then the expression for reversible entrainment ratio is obtained, as shown in Equation (15). Equation (15) for reversible entrainment ratio can then be substituted into the efficiency of Elrod (1945) in order to obtain the efficiency of Elbel and Hrnjak (2008), as shown in Equation (16).

$$\Phi_{m,rev} = \frac{h_{m,in} - h_{m,out,isen}}{h_{s,out,isen} - h_{s,in}} \tag{15}$$

$$\eta_{Elrod} = \frac{\Phi_m}{\Phi_{m,rev}} = \frac{\Phi_m}{\left(\dfrac{h_{m,in} - h_{m,out,isen}}{h_{s,out,isen} - h_{s,in}}\right)} = \Phi_m \frac{h_{s,out,isen} - h_{s,in}}{h_{m,in} - h_{m,out,isen}} = \eta_{Elbel} \tag{16}$$

3. COMPARISON OF TWO-PHASE EJECTOR EFFICIENCIES

3.1 Numerical Comparison

In order to gain better understanding of how the various efficiencies for two-phase ejectors presented above compare to each other, a numerical model of an ejector was developed. The numerical model is used in this section to calculate the output of the ejector based on the specified inlet fluids states. The Kornhauser (1990) model is used to simulate ejector performance. This model is a 1-D two-phase ejector model based on the assumption of homogeneous equilibrium; the fluid is assumed to be in thermodynamic equilibrium at all points, and fluid properties are assumed to be constant across a cross-section at all points. The motive nozzle, suction nozzle, and diffuser of the ejector are each modeled with their own prescribed value of isentropic efficiency; any sources of irreversibility in the components can be accounted for by these efficiencies. The model assumes that the mixing of motive and suction flows occurs at constant pressure, and this pressure must be prescribed as an input to the model. The mass flow rates of the motive and suction streams must also be specified as inputs for the model.

For the present simulation, the isentropic efficiencies of the motive nozzle, suction nozzle, and diffuser were all assumed to be unity. The mixing section pressure is used as an independent variable in the analysis so that ejector performance can be optimized with respect to mixing pressure; the optimum mixing section pressure occurs when motive and suction streams enter the mixing section at the same velocity, resulting in no shear losses between the two streams. An ejector cannot realistically be constructed such that its component efficiencies are unity and its mixing pressure is always optimized. However, perfectly efficient components and an optimized mixing pressure will result in ideal (reversible) ejector performance, meaning that the ejector efficiency would be expected to take a value of unity; this will be further investigated for each of the efficiencies below. The motive inlet fluid was set to be saturated liquid at 45ºC, and the suction inlet fluid was set to be saturated vapor at 5ºC. R134a was used as the working fluid. The entrainment ratio was set to be 0.5.

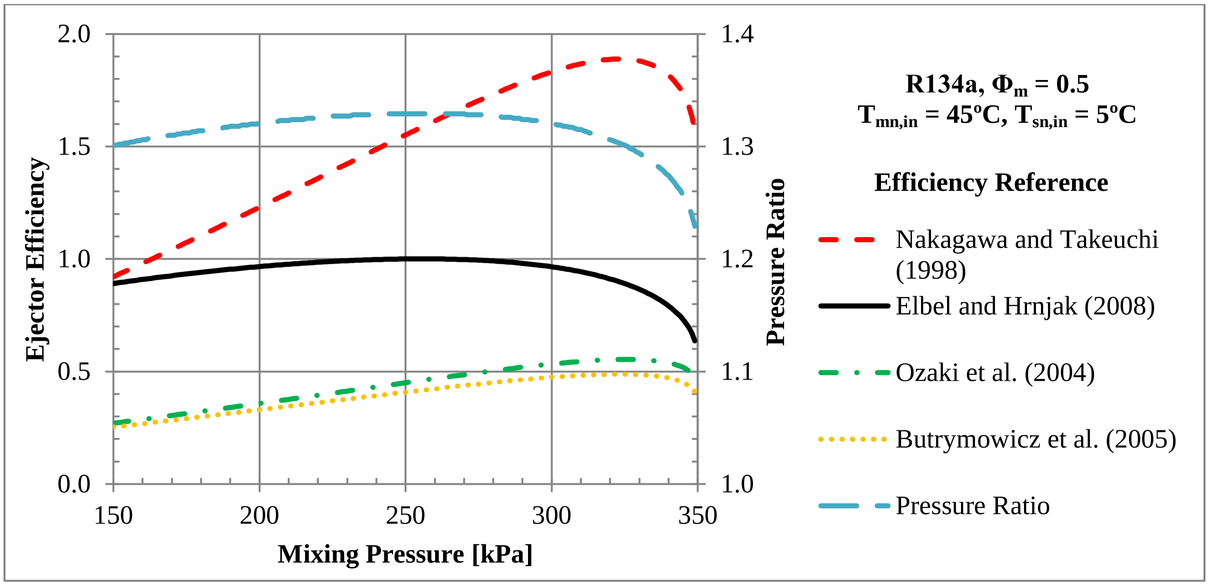

Figure 2: Numerical comparison of two-phase ejector efficiencies and ejector pressure ratio with variation of mixing section pressure.

Figure 2 shows how the efficiency definitions of Nakagawa and Takeuchi (1998), Butrymowicz *et al.* (2005), Elbel and Hrnjak (2008), and Ozaki *et al.* (2004), as well as the suction pressure ratio from Equation (2), compare to each other over a range of mixing pressures. It can be seen from the plot that the efficiency of Nakagawa and Takeuchi (1998) yields the largest value for the entire range of mixing pressures. As discussed above, the efficiency of Nakagawa and Takeuchi (1998) accounts for the pressure rise of both the motive and suction streams assuming that both streams start at the suction state. The efficiency does also account for the additional available power due to the further expansion of the motive stream from diffuser to mixing pressure, meaning that it makes sense to account for the power to recompress the motive stream in the numerator. However, the motive stream is partially liquid when it is recompressed in the mixing section and diffuser; liquid requires less specific power to compress than vapor, meaning that this efficiency is overestimating the power required to compress both streams. This explains why the efficiency of Nakagawa and Takeuchi (1998) always yields a higher value than the other efficiencies; it also explains why, as seen in the plot, this efficiency can yield values greater than unity.

On the other hand, the efficiencies of Ozaki *et al.* (2004) and Butrymowicz *et al.* (2005) are always the lowest in value. The efficiency of Ozaki *et al.* (2004) is the same as that of Elbel and Hrnjak (2008) except the power of the motive stream expanding to the mixing pressure is used in the Ozaki *et al.* (2004) efficiency rather than the power of the motive stream expanding to the diffuser pressure, as in the efficiency of Elbel and Hrnjak (2008); thus, it makes sense that the efficiency of Ozaki *et al.* (2004) would always be lower than that of Elbel and Hrnjak (2008). The efficiency of Butrymowicz *et al.* (2005) shows a value similar to, though slightly less than, the efficiency of Ozaki *et al.* (2004). The efficiency of Butrymowicz *et al.* (2005) assumes isothermal compression of the suction stream, which actually results in slightly higher specific compression power than the isentropic assumption in the efficiency of Ozaki *et al.* (2004). However, the efficiency also assumes that the power provided by the motive stream is equal to the kinetic energy at the motive nozzle outlet, which for a perfect nozzle is equal to the power provided by an isentropic expansion from motive inlet to mixing pressure, plus the energy associated with an isochoric pressure decrease from motive inlet to diffuser outlet pressure. The kinetic energy term is generally greater than the pressure decrease term in this definition, so the denominator of the Butrymowicz *et al.* (2005) efficiency is slightly larger than that of the Ozaki *et al.* (2004); however, the compression power of the suction stream is also slightly larger in the Butrymowicz *et al.* (2005) efficiency, resulting in fairly similar values for the two efficiency definitions.

It can be seen that each efficiency shows a maximum value on the range of entrainment ratios. Recall that the component efficiencies are each unity for this simulation, so it would make sense that at optimum mixing pressure, the ejector achieves a perfect efficiency value, which is generally thought of as being unity. However, only the efficiency of Elbel and Hrnjak (2008) has a maximum value of unity. The maximum value of the Nakagawa and Takeuchi (1998) efficiency is greater than unity for the reason discussed above. The maximum value of the Ozaki *et al.* (2004) efficiency is less than unity because, as described above, the efficiency accounts for the additional power due to expansion from diffuser to mixing pressure in the denominator, but the numerator only accounts for the compression power of the suction stream. The maximum value of the Butrymowicz *et al.* (2005) efficiency is also less than unity due to the isothermal compression assumption and the pressure decrease term in the denominator.

It can also be seen that the optimum mixing pressure that results in the maximum ejector efficiency is different for different efficiency definitions. The efficiency of Elbel and Hrnjak (2008) shows its maximum value at the same mixing pressure as the suction pressure ratio, meaning that this efficiency is at a maximum when the pressure increase of the suction stream is at a maximum for a fixed mass entrainment and inlet conditions. On the other hand, the efficiency definitions of Nakagawa and Takeuchi (1998), Ozaki *et al.* (2004), and Butrymowicz *et al.* (2005), each of which are functions of mixing section pressure, all show their maximum values at a mixing pressure that is only slightly less than the suction pressure of the ejector. A lower mixing pressure will result in a larger value of the denominator in each of these efficiencies; thus, these efficiencies will tend to be lower at lower mixing pressures even though the maximum pressure lift for a fixed mass entrainment does occur at lower mixing pressure.

3.2 Experimental Comparison

The above mentioned two-phase ejector efficiency definitions can also be compared using experimental data. The experimentally determined efficiency values for an R134a ejector are shown in Figure 3 over a range of entrainment ratios. Lawrence and Elbel (2014) provide details of the R134a ejector design and dimensions and the experimental facility that was used to obtain the data. Note that ejector performance is strongly influenced by the geometry of the ejector. However, the intent of this study is to compare ejector performance definitions rather than to study ejector geometry effects; thus, all of the data points in Figure 3 were obtained with the same ejector geometry. Several of the efficiency definitions require knowledge of the mixing pressure, which was assumed to be 10 kPa lower than the suction inlet pressure, corresponding to a 1 K decrease in saturation temperature. The numerical results in Figure 3 show that a mixing section pressure 10 kPa lower than the suction inlet pressure will result in near maximum values for the efficiency definitions of Nakagawa and Takeuchi (1998), Ozaki *et al.* (2004), and Butrymowicz *et al.* (2005), which are the three efficiencies that are dependent on mixing pressure.

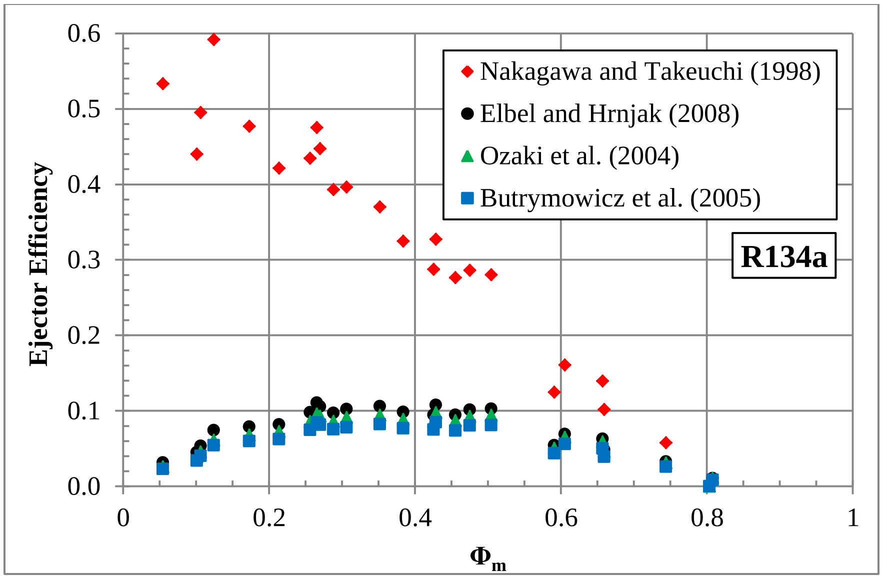

Figure 3: Experimentally determined ejector performance in terms of various two-phase ejector efficiencies for an R134a ejector.

Figure 3 shows that three of the ejector efficiency definitions show very similar values and behavior with respect to varying entrainment ratio, but the efficiency definition of Nakagawa and Takeuchi (1998) shows significantly different values and a different trend compared to the other three. Similar to the numerical results discussed above, the efficiency of Nakagawa and Takeuchi (1998) is the highest over the range of conditions, and the efficiencies of Ozaki *et al.* (2004) and Butrymowicz *et al.* (2005) are always the lowest over the range of conditions. The reasons for the different relative magnitudes of the four different efficiencies are discussed in the preceding section on numerical results.

Because of the pump-like nature of the ejector, higher entrainment ratio means lower pressure increase provided to the suction stream; there is generally a trade-off between the mass entrainment effect and the pressure rise that an ejector can provide. Since high mass entrainment and high pressure lift are both desired when working with two-phase ejectors, it makes sense that the most desirable ejector performance is achieved at an intermediate entrainment ratio, when both mass entrainment and pressure increase are relatively high. The efficiency definitions of Elbel and Hrnjak (2008), Ozaki *et al.* (2004), and Butrymowicz *et al.* (2005) all show their maximum values at an intermediate entrainment ratio. On the other hand, the efficiency of Nakagawa and Takeuchi (1998) shows its maximum value at the lowest entrainment ratio. This is due to the fact that, neglecting the small kinetic energy contribution in the denominator, this efficiency is proportional to the quantity $(1 + \Phi_m)$ while the other three efficiencies are proportional to Φ_m. Thus, at zero mass entrainment, when pressure rise is at its highest, the Nakagawa and Takeuchi (1998) efficiency can still take on a non-zero value whereas the other efficiencies must be zero because entrainment ratio is zero.

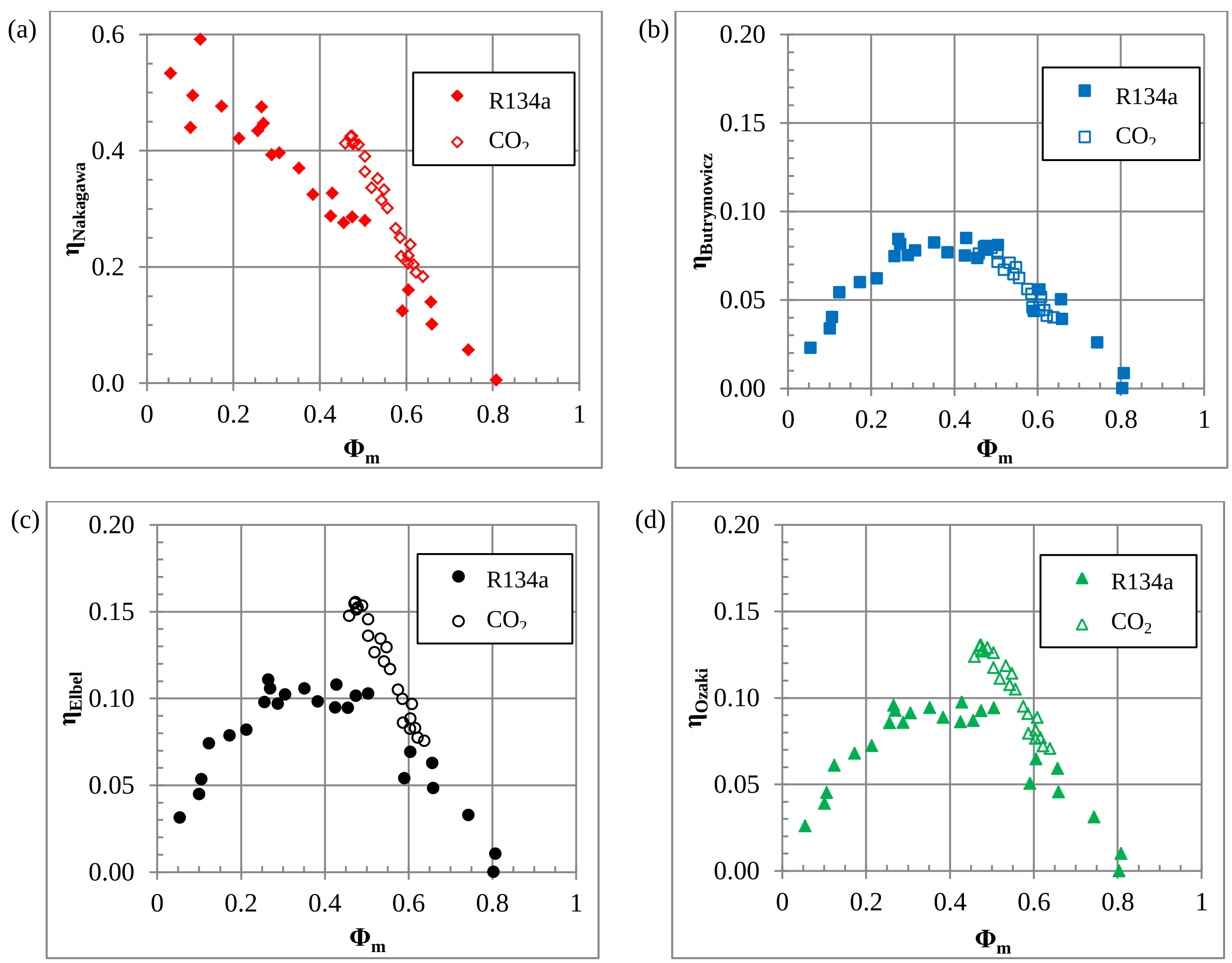

Figure 4: Comparison R134a and CO_2 ejector performance for varying entrainment ratio using the ejector efficiency definition of: (a) Nakagawa and Takeuchi (1998); (b) Butrymowicz *et al.* (2005); (c) Elbel and Hrnjak (2008); (d) Ozaki *et al.* (2004).

Figure 4 compares the experimentally determined efficiency values between the R134a ejector described above and a CO_2 ejector; Elbel and Hrnjak (2008) provide details of the CO_2 ejector design and dimensions and the experimental facility. CO_2 is more commonly used in two-phase ejector systems because of the fluid's high throttling loss, especially for transcritical operation. Though the range of entrainment ratios for which data is available is not as wide for the CO_2 ejector, it can clearly be seen from Figure 4 that the CO_2 ejector can achieve greater efficiency than the R134a ejector based on three of the four efficiency definitions; for the efficiency definition of Butrymowicz *et al.* (2005), the CO_2 data points seem to take on very similar values as the R134a data points. The reason for this is not clear, though it may be that the pressure difference term in the denominator of the efficiency is much larger for the case of CO_2; CO_2 experiences a much larger pressure decrease through the nozzle of the ejector compared to R134a, but the density of the motive fluid is similar for both fluids. It can also be seen that, as was the case with R134a, the efficiency of Nakagawa and Takeuchi (1998) produces significantly higher values with the CO_2 ejector than the other three efficiency definitions produce.

As a result of CO_2's higher throttling loss, there is greater opportunity to recover power with a two-phase ejector, and thus, greater opportunity to improve the COP of CO_2 systems compared to R134a systems. However, all four of the two-phase ejector efficiency definitions mentioned above represent different ways to quantify some amount of power actually recovered in the ejector divided by some maximum theoretical power that could be recovered; the higher power recovery of the CO_2 ejector should not have an influence on the efficiency values reported in Figure 4. Thus, it appears that the CO_2 ejector is actually a more effective expansion device than the R134a ejector. In fact, CO_2 ejector efficiency, according to the definition of Elbel and Hrnjak (2008), has been reported as high as 31 % by Banasiak *et al.* (2012). Though the amount of available data for R134a ejectors is far more limited, the authors feel that R134a ejectors cannot reasonably achieve efficiency values as high as those already reported for CO_2 ejectors. One possible explanation for the lower efficiency of R134a ejectors is the lower absolute vapor density, which can result in greater frictional losses in the ejector. Another possible reason for R134a ejectors' lower efficiency is the greater difference between liquid and vapor densities compared to CO_2; the larger difference between the two phases may result in less efficient mixing and greater shearing losses during the mixing process in the ejector.

4. CONCLUSIONS

Several different efficiency definitions proposed for use with two-phase ejectors have been presented and discussed in this paper. The different efficiency definitions all represent different ways to quantify some amount of power recovered by an ejector divided by some theoretical maximum power that is available for recovery; however, each efficiency involves different assumptions in order to determine these quantities, and as a result, the efficiencies can take very different values for the same ejector operating point. Thus, when an ejector efficiency is reported, it is important to understand how that efficiency is defined and what the limits of the efficiency are.

A numerical model was used to compare the different efficiency definitions. It was seen that even for reversible ejector performance, only the efficiency of Elbel and Hrnjak (2008) achieved a value of unity at its optimum mixing pressure; it was also seen that this efficiency was the only one that showed its maximum value at the point of largest pressure increase. It was seen experimentally with an R134a ejector that the efficiency of Nakagawa and Takeuchi (1998) takes significantly higher values and displays different behavior with respect to entrainment ratio compared of the other three efficiencies. It was also seen experimentally that in addition to being able to recover a larger amount of power, a CO_2 ejector can also achieve greater efficiency than an R134a ejector. Possible explanations for the higher efficiency of CO_2 ejectors are the higher vapor density of CO_2 (lower frictional losses) and the smaller difference between liquid and vapor densities (more efficient mixing) compared to R134a.

REFERENCES

American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE), 1983, *Handbook: Equipment*, Chapter 13: Steam jet refrigeration equipment, Atlanta, GA, USA.

Banasiak, K., Hafner, A., Andresen, T., 2012, Experimental and numerical investigation of the influence of the two-phase ejector geometry on the performance of the R744 heat pump, *Int. J. Refrig.*, vol. 35: p. 1617-1625.

Butrymowicz, D., Karwacki, J., Trela, M., 2005, Investigation of two-phase ejector in application in compression refrigeration systems, *Int. Conf. Thermophysical Prop. Trans. Proc. Ref.*, IIR, Vicenza, Italy.

Chunnanond, K., Aphornratana, S., 2004, Ejectors: Applications in refrigeration technology, *Renew. Sustain. Energy Reviews*, vol. 8, pp. 129-155.
Elbel, S., Hrnjak, P., 2008, Experimental validation of a prototype ejector designed to reduce throttling losses encountered in transcritical R744 system operation, *Int. J. Refrig.*, vol. 31: p. 411-422.
Elrod, H.G., 1945, The theory of ejectors, *J. Appl. Mech.*, vol. 12: p. A170-A174.
Köhler, J., Richter, C., Tegethoff, W., Tischendorf, C., 2007, Experimental and theoretical study of a CO_2 ejector refrigeration cycle, Presentation at VDA Winter Meeting, Saalfelden, Austria.
Kornhauser, A.A., 1990, The use of an ejector as a refrigerant expander, *Proc. 1990 USNC IIR-Purdue Refrig. Conf.*, IIR, West Lafayette, IN, USA.
Lawrence, N., Elbel, S., 2014, Experimental investigation of an alternate two-phase ejector cycle suitable for use with low-pressure refrigerants, *Int. J. Refrig.*, vol. 38: p. 310-322.
Lee, J.S., Kim, M.S., Kim, M.S., 2011, Experimental study on the improvement of CO_2 air conditioning system performance using an ejector, *Int. J. Refrig.*, vol. 35: p. 1614-1626.
Nakagawa, M., Takeuchi, H., 1998, Performance of two-phase ejector in refrigeration cycle, *Proc. 3rd Int. Conf. Multiphase Flow*, Lyon, France.
Nakagawa, M., Marasigan, A.R., Matsukawa, T., 2011, Experimental analysis on the effect of internal heat exchangers in transcritical CO_2 refrigeration cycle with two-phase ejector, *Int. J. Refrig.*, vol. 35: p. 1577-1586.
Ozaki, Y., Takeuchi, H., Hirata, T., 2004, Regeneration of expansion energy by ejector in CO_2 cycle, *Proc. 6th IIR Gustav Lorentzen Conf. Nat. Working Fluids*, IIR, Glasgow, UK.

NOMENCLATURE

Symbols

h	specific enthalpy [kJ/kg]
$\dot{m}$	mass flow rate [kg/s]
P	pressure [kPa]
s	specific entropy [kJ/kg-K]
T	temperature [K]
V	velocity [m/s]
$\dot{W}$	power [W]

Greek Symbols

η	ejector efficiency
Π_s	suction pressure ratio
ρ	density [kg/$^{m3]}$
Φ_m	mass entrainment ratio

Subscripts

ASHRAE	referring to a publication by ASHRAE (1983)
Butrymowicz	referring to a publication by Butrymowicz *et al.* (2005)
cp	compressor
diff	diffuser
Elbel	referring to a publication by Elbel and Hrnjak (2008)
Elrod	referring to a publication by Elrod (1945)
exp	expander
in	inlet of component
isen	isentropic process
m	motive stream
max	maximum
mix	mixing section
mn	motive nozzle
Nakagawa	referring to a publication by Nakagawa and Takeuchi (1998)
out	outlet of component
Ozaki	referring to a publication by Ozaki *et al.* (2004)
rec	recovered by ejector
rev	reversible process
s	suction stream
sn	suction nozzle

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