Utilization of Ejector for Decrease of Compressor Discharge Pressure in HVAC&R Applications

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ABSTRACT

In vapor compression refrigeration systems, two-phase refrigerant ejectors have been widely considered to increase compressor suction pressure leading to improved system performance. Nevertheless, ejectors could be utilized in an unconventional way for reducing the compressor discharge pressure and temperature. In this case, the principle of the condensing ejector could be used. The two-phase condensing ejector principle has been used a few decades ago using steam-water mainly for underwater propulsion and liquid metal MHD power generation systems. Recently, some researchers implemented this principle in the case of R22 and R-507 refrigerants. However, with the help of condensing ejector principles, there might be an opportunity to achieve a certain portion of the compression work of the system by the ejector for a wide range of operating conditions as well as different refrigerants. This paper discusses the feasibility study of this innovative utilization of two-phase ejectors considering a pertinent ejector cycle. A numerical modeling study has been performed for different refrigerants, for example, R134a, R1234yf, R1233zd(E) to explore the potential of an ejector as a way of reducing compressor outlet pressure and temperature. Preliminary modeling results from the considered ejector cycle seem to be promising and reveal that it might be possible to achieve reduced compressor discharge pressure as well as temperature in a wide range of entrainment ratios. The system, as well as the ejector performance, have also been reported for this ejector cycle.

1. INTRODUCTION

Two-phase refrigerant ejectors are mainly used in vapor compression refrigeration systems to recover some of the power that could have been lost during the throttling process. One of the most common utilizations of ejectors so far is to increase the compressor suction pressure for decreasing the compression work in the system. However, there might be some other novel ways through which ejectors could be utilized in HVAC&R applications. One such innovative application of ejectors could be to achieve reduced compressor discharge pressure and temperature. The higher compressor discharge temperature could be a crucial issue for certain refrigerants. For some refrigerants, the high discharge temperature from the compressor might have some negative impacts on the corresponding HVAC&R system. In this case, the reliability of the compressor might be affected, and the oil breakdown might occur as oil could lose the lubrication film due to the higher compressor discharge temperature. Besides, the chemical decomposition of the oil could happen earlier due to the running of the system at a higher compressor discharge temperature for a longer timeframe. Therefore, the use of an ejector to achieve reduced compressor exit pressure as well as temperature might potentially benefit different HVAC&R systems.

An ejector works on the momentum transfer principle between two different fluid streams; one is the motive fluid, and another is the suction fluid. Typically, in a two-phase refrigerant ejector, the motive fluid is expanded in a converging-diverging nozzle or sometimes, only a converging nozzle. This expansion of the motive fluid through a nozzle creates a low-pressure region at the motive nozzle outlet, thus facilitating to entrain of the suction fluid through the suction chamber. Eventually, the motive and the suction fluids are mixed in the mixing section. The motive and the suction fluids can be both liquid and vapor. Finally, the mixed flow is decelerated through the diffuser, thereby, achieving a higher pressure at the diffuser outlet than the pressure at the suction chamber inlet. Current research reports the significant system performance improvement from the standard ejector cycle by achieving reduced compression work in the system (Elbel and Hrnjak, 2008), (Lucas and Koehler, 2012).

Condensing ejector principle could be used to decrease the refrigerant pressure and temperature at the compressor outlet. An analytical, as well as experimental study on the two-phase condensing ejector, was first initiated by Miguel and Brown (1964). They experimented with steam-salt water, and their application was mainly for underwater propulsion and liquid metal MHD power generation systems. Miguel and Brown (1964) stated that there exists a very high vapor condensation rate, and they assumed that the vapor completely condenses inside the constant-area mixing section of the ejector. In their study, it is reported a shock wave occurring at the mixing section which they referred to as the “condensation shock”. Later Levy and Brown (1972) concluded through their experimental results that the condensation shock must occur for the complete vapor condensation inside the ejector, and it is also affected by the motive fluid inlet velocity. Recently, Bergander (2006) first attempted the principle of the condensing ejector using the R22 refrigerant. Bergander (2006) proposed an ejector cycle to achieve a certain portion of the required compression work through the ejector, thus improving the system performance. The first experimental attempt reported 16% higher COP than the standard vapor compression system although Bergander (2006) showed that a maximum of 38% improvement in COP could be possible theoretically. Later Bergander *et al.* (2010) experimented with R-507 refrigerant using the same ejector cycle and reported that the improvement in COP is somewhat limited for that particular refrigerant. In his study, he also hypothesized that different refrigerants might offer potential benefits through this unconventional ejector cycle.

This paper presents a numerical study to assess an ejector’s potential for achieving reduced compressor discharge pressure as well as temperature. The ejector cycle proposed by Bergander (2006) is simulated with the help of an appropriate numerical modeling scheme using different refrigerants, for instance, R134a, R1234yf, R1233zd(E). The principle of the condensing ejector and the associated ejector cycle patented by Bergander (2006) is explained in section 2. Section 3 of this manuscript describes the ejector modeling procedure for the decrease of compressor outlet pressure and temperature. Preliminary modeling results in terms of the system and the ejector performance metrics are represented in section 4. These modeling results reveal that it could be possible to utilize an ejector as a way of achieving reduced pressure and temperature at the compressor outlet in the case of different refrigerants.

2. CONDENSING EJECTOR PRINCIPLE AND EJECTOR CYCLE

2.1 Principle of Condensing Ejector

Figure 1 illustrates the schematic of the condensing ejector principle as stated in Miguel and Brown (1964). The high-pressure liquid is accelerated through the converging nozzle whereas, the high-velocity vapor enters through the suction chamber. There exists a high relative velocity difference between these two fluid streams, and different thermodynamic states at the inlet of the ejector could be utilized advantageously. These two fluid streams start to mix with each other in the converging mixing section. Due to the very high relative velocity difference, a large value of the heat transfer coefficient is achieved from these fluids leading to a higher vapor condensation rate inside the ejector. The uncondensed vapor and the liquid stream flow through the constant area mixing section, and a sharp pressure rise, as well as a shock wave, is noticed at this point. The rapid change in density of the fluid streams causes this sharp pressure rise. This condensation shock produces a completely liquid state (section y), and once again the pressure rises of the liquid stream as it flows through the diffuser (section 4). Miguel and Brown (1964) also reported that the exit stagnation pressure could be higher than that of pressure at both inlets of the ejector.

2.2 Ejector Cycle for Condensation Shock

The ejector cycle proposed by Bergander (2006) is demonstrated in Figure 2(a) and the associated P-h diagram is shown in Figure 2(b). In this ejector cycle, the flow is split at the condenser outlet (point 7), and a certain portion of the liquid flow gets pumped and fed as the motive flow (point 8) to the ejector. The remaining flow at the condenser outlet is expanded through a metering valve and flows through the evaporator (point 9). The high-pressure liquid flow (point 8) generates suction of the compressed vapor (point 2) from the compressor outlet to the ejector suction chamber. The liquid and the vapor streams are mixed in the mixing section (point 5), and the pressure of the mixed flow rises up to the condensation pressure (point 6). Thus, a certain portion of the compression work could be achieved by the ejector itself, thereby, unloading the compressor. The key advantage of this ejector cycle could be to achieve reduced compressor discharge pressure as well as temperature because the compressor exit pressure is not achieved at the same level as the condenser pressure. In addition, if the compressor pressure ratio could be decreased, there arises an opportunity to achieve reduced refrigerant temperature at the compressor outlet. The reduced compressor discharge temperature could be beneficial for certain refrigerants and that might allow a wide range of system operations. However, there exists a tradeoff for the system performance improvement in the considered ejector cycle. The pump work should be less compared to that of the compressor, and the ejector efficiency needs to be reasonable for better system operation.

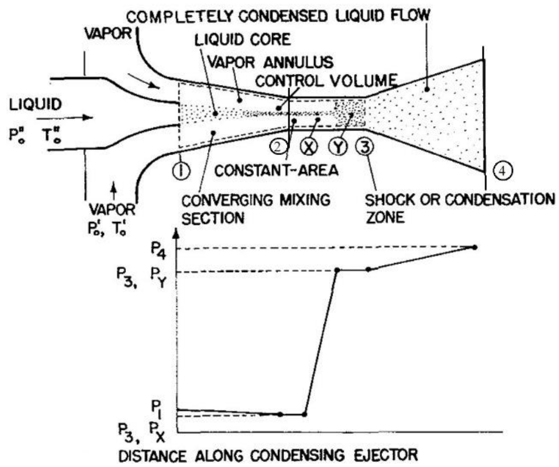

Figure 1: Schematic diagram of the condensing ejector (Miguel and Brown, 1964)

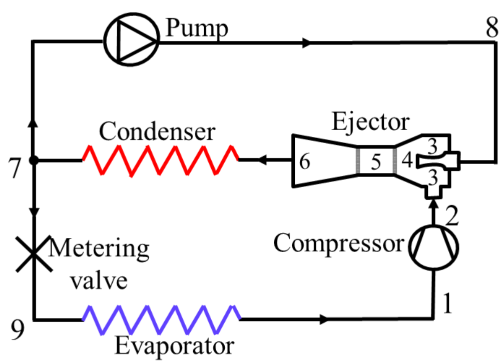

Figure 2(a): Schematic diagram of the ejector cycle proposed by Bergander (2006)

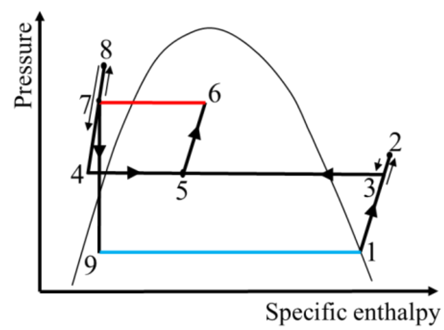

Figure 2(b): P-h diagram of the associated ejector cycle for compressor discharge pressure decrease

3. EJECTOR MODELING PROCEDURE

3.1 Numerical Modeling of the Ejector Cycle

The ejector cycle concerning the compressor discharge pressure decrease illustrated in Figure 2(a) is simulated under suitable assumptions. The considered numerical model investigates a steady-state system operation to assess the ejector's potential for this unconventional use in the case of different refrigerants, for example, R134a, R1234yf, R1233zd(E). In this numerical modeling scheme, the evaporator and the condenser temperature are kept fixed at 10 °C (283 K) and 40 °C (313 K), respectively, whereas, a subcooling of 5 K is assumed at the condenser outlet. The mass flow rate through the evaporator as well as through the ejector suction nozzle is governed by assuming a fixed evaporator cooling capacity of 5 kW. The power of the compressor and pump is determined assuming a fixed isentropic efficiency of 80% and 100%, respectively, in the case of all the refrigerants. The motive inlet state of the ejector is set by assuming a fixed pressure at the pump inlet. As different refrigerants have different saturation pressures, the motive inlet pressure for different refrigerants is assumed to be the saturation pressure for the same corresponding temperature.

The simulation procedure proposed by Kornhauser (1990) is modified for the considered ejector cycle and implemented in the numerical modeling scheme. The fixed isentropic efficiency values are considered for ejector components (motive nozzle, suction chamber, diffuser). In the case of R134a, these fixed efficiencies are considered based on the experimental studies, whereas, a reasonable assumption for these efficiency values of ejector components is made in the case of R1234yf and R1233zd(E) refrigerants. A relation between the suction nozzle inlet pressure and the ejector mixing section pressure is set as an input to the simulation procedure. Regarding the mixing section pressure, it is assumed that the saturation pressure corresponds to the 1 K saturation temperature drop from the ejector suction chamber inlet. Besides, the motive flow rate ratio (r) which is defined as the ratio of motive mass flow rate to the total mass flow rate is also given as an input to the numerical model. Equations for the ejector components as well as for the system are solved with the help of the Engineering Equation Solver (EES). The simulation procedure gives output for the pressure lift from the ejector as well as the compressor outlet temperature. Figure 3 shows the detailed ejector modeling procedure for the considered ejector cycle.

Inputs: $P_{mn,in}, h_{mn,in}, h_{sn,in}, \eta_{mn}, \eta_{sn}, \eta_{diff}, r$
Other inputs: relation between $P_{sn,in}$ and P_{ms}

⇩

Motive nozzle: $h_{mn,out,is} = f(P_{ms}, s_{mn,in})$

$h_{mn,out} = h_{mn,in} + \eta_{mn}(h_{mn,out,is} - h_{mn,in})$ [Isentropic efficiency definition]

$v_{mn,out} = \sqrt{2(h_{mn,in} - h_{mn,out})}$ [Energy conservation]

⇩

Suction nozzle: $h_{sn,out,is} = f(P_{ms}, s_{sn,in})$

$h_{sn,out} = h_{sn,in} + \eta_{sn}(h_{sn,out,is} - h_{sn,in})$ [Isentropic efficiency definition]

$v_{sn,out} = \sqrt{2(h_{sn,in} - h_{sn,out})}$ [Energy conservation]

⇩

Mixing section: $v_{diff,in} = r v_{mn,out} + (1-r) v_{sn,out}$ [Linear momentum conservation]

$h_{diff,in} = r h_{mn,in} + (1-r) h_{sn,in} - 0.5 v_{diff,in}^2$ [Energy conservation]

$s_{diff,in} = f(P_{ms}, h_{diff,in})$

⇩

Diffuser: $h_{diff,out,is} = h_{diff,in} + 0.5 \eta_{diff} v_{diff,in}^2$ [Isentropic efficiency definition]

$h_{diff,out} = h_{diff,in} + 0.5 v_{diff,in}^2$ [Energy conservation]

⇩

Outputs: $P_{lift}, T_{sn,in}$

Figure 3: Simulation procedure for the ejector model of compressor outlet pressure decrease

3.2 Ejector Performance Parameters

The performance of an ejector can largely affect the overall system performance in HVAC&R systems. Most often, the ejector performance is quantified by two parameters; the mass entrainment ratio and the pressure lift. The mass entrainment ratio (ϕ_m) for the ejector is defined as the ratio of the suction fluid mass flow rate ($\dot{m}_{sn}$) to the motive fluid mass flow rate ($\dot{m}_{mn}$). Besides, pressure lift also characterizes the ejector performance which is the difference between the pressure at the ejector outlet and the suction chamber inlet pressure. The pressure lift from an ejector can be different for different refrigerants depending upon the saturation pressure levels. A good ejector should have both a larger mass entrainment ratio as well as a higher pressure lift to deliver improved performance at the system level.

$$\text{Mass entrainment ratio, } \phi_m = \frac{\dot{m}_{sn}}{\dot{m}_{mn}} = \frac{1}{r} - 1 \tag{1}$$

$$\text{Pressure lift, } P_{lift} = P_{diff,out} - P_{sn,in} \tag{2}$$

The ejector efficiency (η_{ej}) is another performance metric for the ejector which characterizes the possible amount of work recovery from the isenthalpic throttling process. Researchers defined this parameter in different ways in the contemporary literature. However, the definition of ejector efficiency proposed by Elbel and Hrnjak (2008) is used in the present study. According to Elbel and Hrnjak (2008), ejector efficiency (η_{ej}) is defined as the ratio of the amount of work recovered from an ejector to the maximum amount of work recovery possible through the same ejector. The work recovery of the ejector is calculated considering the isentropic compression of the suction fluid stream from the inlet of the ejector suction to the ejector diffuser outlet. Moreover, the isentropic expansion of the motive fluid stream from the motive nozzle inlet to the ejector diffuser is considered the theoretical or maximum amount of recoverable work potential from an ejector. Refrigerant properties at the considered states are used for the determination of the ejector efficiency by using equation (3).

$$\text{Ejector efficiency, } \eta_{ej} = \frac{\dot{m}_{sn}}{\dot{m}_{mn}} \left[\frac{h(P_{diff,out}, s_{sn,in}) - h_{sn,in}}{h_{mn,in} - h(P_{diff,out}, s_{mn,in})} \right] = \frac{\dot{W}_{rec}}{\dot{W}_{rec,max}} \tag{3}$$

4. NUMERICAL MODELING RESULTS FOR THE EJECTOR CYCLE

4.1 Numerical Modeling Results for the System Performance

The compressor outlet temperature, as well as the pressure, is solved through the developed numerical modeling scheme. One of the main objectives of this study is to find out the potential of an ejector to achieve reduced compressor discharge temperature. Figures 4(a) and 4(b) represent the compressor outlet temperature variation with the entrainment ratio in the case of R134a and R1233zd(E) refrigerants, respectively. A comparative analysis is also performed between the simple vapor compression cycle (considering no ejector) and the considered ejector cycle to assess the improvement potential in terms of compressor discharge temperature. Both Figures 4(a) and 4(b) reveal that the compressor outlet temperature increases with the increase in entrainment ratio from the ejector. In the considered ejector model, the flow at the compressor outlet is the suction flow that is considered constant by the fixed cooling capacity. Therefore, the increase in entrainment ratio means a less amount of motive flow rate that eventually increases the compressor outlet pressure as well as the temperature. In the case of a simple vapor compression cycle, no ejector is involved, and for the fixed evaporator and condenser temperature, a constant compressor outlet temperature is attained. From the comparative analysis, it is obvious that for the considered refrigerants, the decrease of compressor outlet temperature is possible in a wide range of entrainment ratios.

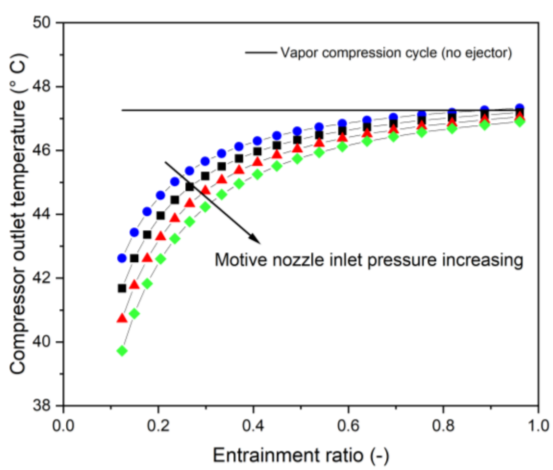

Figure 4(a): Variation of compressor outlet temperature with entrainment ratio for R134a

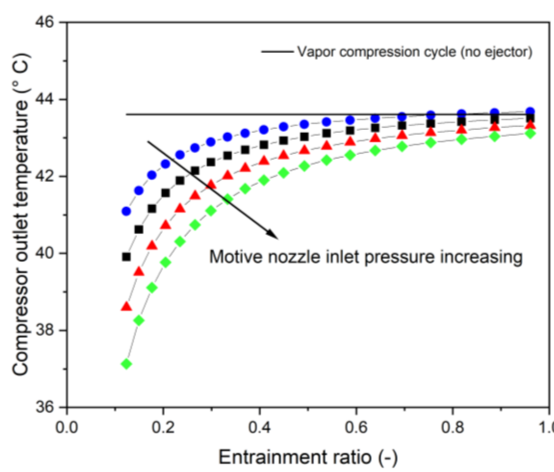

Figure 4(b): Variation of compressor outlet temperature with entrainment ratio for R1233zd(E)

Figure 4(a) shows that a maximum decrease of around 8 °C in compressor discharge temperature is possible in the case of the considered ejector cycle for R134a than that of the simple vapor compression cycle. Whereas, in the case of R1233zd(E), numerical modeling results show the maximum decrease of around 6.5 °C (Figure 4(b)) in compressor outlet temperature. Furthermore, from both Figures 4(a) and 4(b) it is observed that in the case of the higher motive nozzle inlet pressure, the compressor outlet temperature becomes lower. This is obvious because the motive fluid is the subcooled liquid in this considered ejector model, and a higher motive inlet pressure increases the specific enthalpy at the motive nozzle inlet. This effect is much more prominent in the case of a low entrainment ratio of the ejector; because now the motive mass flow rate of the ejector becomes significantly higher compared to that of the suction mass flow rate. Consequently, the compressor outlet temperature, as well as pressure, become lower to perform the ejector operation. These modeling results seem to be promising and practically it might be feasible to use this ejector cycle with a good experimental approach for different refrigerants.

A representative compressor, as well as pump power, variation with the entrainment ratio is shown in Figure 5 for R134a refrigerant. The system performance is evaluated in terms of Coefficient of Performance (COP) which depends upon the compressor as well as pump power. From Figure 5, it is observed that the pump power exponentially decreases with the increase in entrainment ratio. The pump power is governed by the motive fluid mass flow rate and the enthalpy difference for the isentropic compression through the pump. In this case, during isentropic compression by the pump (process 7-8; Figure 2(b)), the enthalpy difference does not change with the entrainment ratio as the refrigerant states are fixed at both pump inlet and outlet. Besides, the suction mass flow rate is assumed to be constant by the cooling capacity in this ejector model, eventually leading to a sharp decrease in the motive mass flow rate for a higher entrainment ratio. This reduced ejector motive mass flow rate decreases the required pump power considerably. Whereas, the enthalpy difference during the isentropic compression (process: 1-2; Figure 2(b)) dictates the required compressor power in the system. As the entrainment ratio of the ejector increases, the compressor outlet pressure tends to approach the condenser pressure. In addition, the difference in the compressor outlet temperature becomes less in the case of a higher entrainment ratio than in the case of lower mass entrainment. This effect is notable in Figure 5, where it is shown that the rate of compressor power increase is high in the case of a low entrainment ratio, but it increases at a lower rate when the entrainment ratio reaches above 0.4. As the compressor outlet pressure and temperature change very little above this entrainment ratio, the enthalpy difference during the isentropic compression (process: 1-2; Figure 2(b)) does not increase considerably compared to the case of a lower value of entrainment ratio. The reason for the little change in enthalpy difference comes from the thermodynamic properties of the considered refrigerant. With a careful investigation of the P-h diagram, it is found that the slope of the isentropic lines does not change notably when they tend to approach the condenser pressure. Consequently, the required compressor power does not significantly increase when the entrainment ratio reaches above 0.4 for the considered ejector cycle.

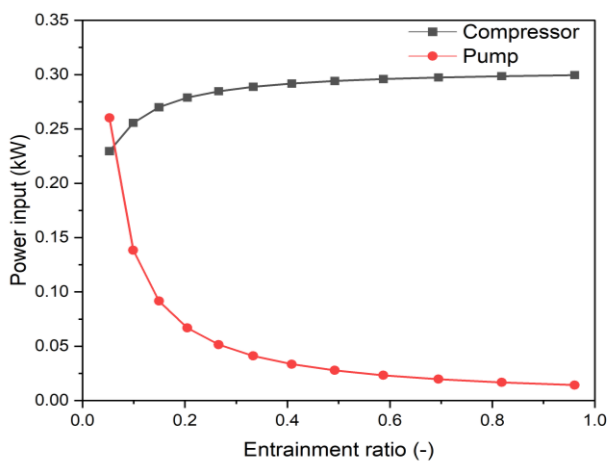

Figure 5: Variation of compressor and pump input power with entrainment ratio for R134a

Figure 6 illustrates a representative variation of the Coefficient of Performance (COP) with the entrainment ratio in the case of R134a. These COP values are thought to be the maximum achievable COP from the considered ejector cycle. COP values from the ejector model are compared to the COP that could be achieved from a simple vapor compression cycle, where no ejector is used. From Figure 6, it is obvious that the COP achieved from the considered ejector cycle becomes less than that of the simple vapor compression cycle (no ejector used). However, the COP values reach relatively close to the COP achieved when no ejector is used in the case of a higher entrainment ratio.

This indicates that it might be possible to achieve a comparable range of COP from the ejector cycle also. In the considered ejector model, several assumptions are considered, for instance, the motive inlet pressure, mixing pressure, and isentropic efficiency values of ejector components. It might be possible to achieve improved system performance from a good experimental approach controlling these experimental parameters. Moreover, the study conducted by Bergander *et al.* (2010) also reported the limitations in COP improvement in the case of R-507.

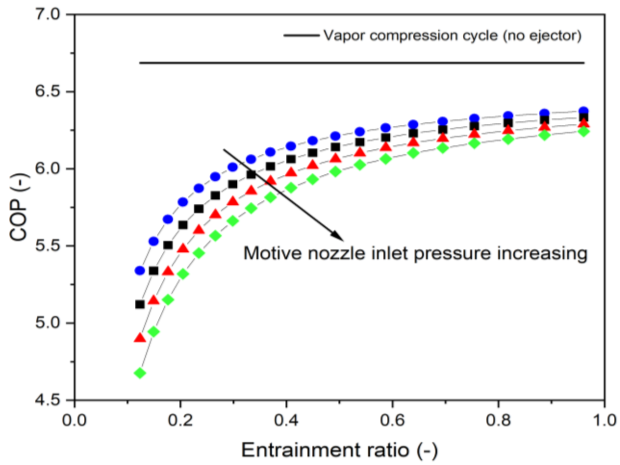

Figure 6: Variation of COP with entrainment ratio for R134a

The effect of the entrainment ratio on the system COP reveals that there exists an increasing trend of the system COP with the increase in entrainment ratio. This trend agrees with the trend reported by Bergander *et al.* (2010). From Figure 6, it is evident that the overall system COP becomes lower in the case of higher motive nozzle inlet pressure because in this case, more pump power is required to perform the ejector operation. Furthermore, the COP increases sharply up to the range of entrainment ratio of 0.4 above which the rate of increase in COP is not significant. This trend can be well explained with the help of compressor and pump power variation represented in Figure 5. Figure 5 reveals that the pump power dramatically decreases up to the entrainment ratio of 0.4. Whereas the evaporator capacity remains constant for all the cases, and the compressor power increases relatively less than that of the pump power in this range of entrainment ratio leading to a notable increase in the system COP. An interesting observation from Figure 6 is that the system performance gets improved in the case of a higher mass entrainment ratio. An experimental investigation of this ejector cycle could render necessary insights regarding the selection of an optimum entrainment ratio for improved system performance.

4.2 Numerical Modeling Results for the Ejector Performance

Numerical modeling results reveal the ejector performance in terms of pressure lift and ejector efficiency that have been defined earlier in section 3.2. Figure 7 represents a comparison of pressure lift from the ejector in a wide range of entrainment ratios for R134a, R1233zd(E), and R1234yf. From Figure 7, it is obvious that for these investigated refrigerants, a reasonable amount of pressure lift might be achievable in a broad range of entrainment ratios. The amount of pressure lift largely depends on the thermodynamic properties, especially, the saturation pressure level for different refrigerants. A comparative analysis among these considered refrigerants reveals that the R1234yf could offer larger work recovery potential than R134a and R1233zd(E) with the implementation of this ejector cycle for the underlying conditions. With a careful analysis of modeling results, it is found that the refrigerant which offers a larger potential for the decrease of compressor discharge temperature could also achieve a higher amount of pressure lift. This observation can be explained by the achievable compressor discharge pressure and temperature for these refrigerants. The larger decrease in compressor outlet temperature eventually decreases the pressure ratio as well as the compressor exit pressure. Consequently, a higher pressure lift from the ejector becomes possible for achieving the same level of condensation pressure. This trend of pressure lift for the considered ejector cycle might provide insights regarding the experimentation as well as the ejector design optimization. Another notable thing from the pressure lift characteristics is that the pressure lift decreases with the increase in entrainment ratio. Contemporary literature also shows a similar trend of the pressure lift variation with the mass entrainment ratio. Future studies could focus on choosing the optimum ejector performance parameters for the experimental investigation of this ejector cycle.

The variation of ejector efficiency with the mass entrainment ratio in the case of R134a is illustrated in Figure 8. The maximum ejector efficiency value in Figure 8 is in the similar order that has been reported by Lawrence and Elbel (2015) in the case of a standard two-phase ejector cycle. Figure 8 illustrates that the ejector efficiency first starts to increase and increases up to the entrainment ratio of around 0.15. Above the entrainment ratio of 0.15, the ejector efficiency sharply decreases for a higher mass entrainment ratio. With the increase in entrainment ratio, the motive mass flow rate decreases as the cooling capacity is assumed to be constant in this ejector cycle. In addition, the isentropic expansion of the motive fluid does not change irrespective of the entrainment ratio. However, the enthalpy difference for calculating the work recovery of the ejector sharply declines with the increase in entrainment ratio. The reason for this decrease comes from the attainment of a higher compressor suction pressure eventually approaching the condenser pressure. Consequently, the pressure lift becomes also lower that has been reported in Figure 7. So, the small enthalpy difference for the isentropic compression from the suction nozzle inlet to the ejector diffuser outlet leads to a decrease in ejector efficiency in the case of a higher entrainment ratio.

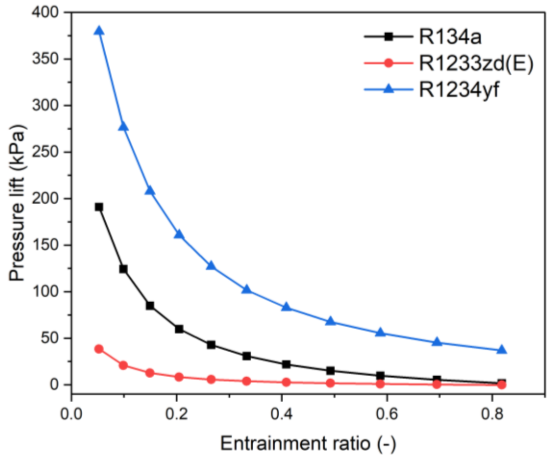

Figure 7: Comparison of pressure lift for different refrigerants

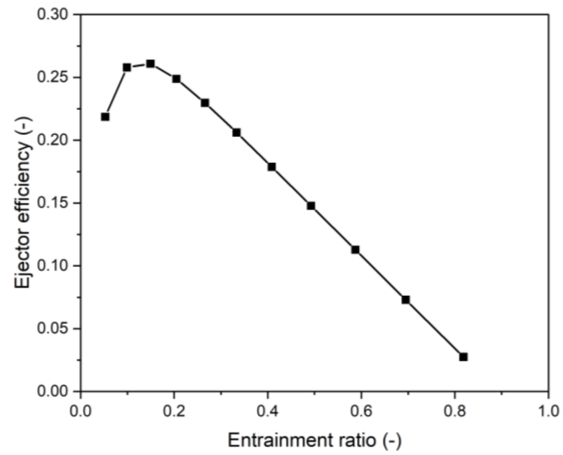

Figure 8: Variation of ejector efficiency with entrainment ratio for R134a

5. CONCLUSIONS

This manuscript presents a numerical modeling study using a pertinent ejector cycle patented by Bergander (2006) to assess the potential of an ejector as a way of achieving reduced compressor discharge pressure as well as the temperature. A numerical modeling scheme was developed to investigate the system as well as the ejector performance regarding this unconventional ejector cycle. Numerical modeling results showed the possibility of reducing compressor outlet temperature by around 8 °C and 6.5 °C, for R134a and R1233zd(E), respectively compared to that of the simple vapor compression system. A comparative study was also performed on the Coefficient of Performance (COP) between this unconventional ejector cycle and the standard vapor compression cycle. Preliminary modeling results with underlying assumptions showed some limitations in the COP improvement from the considered ejector cycle. However, there might be an opportunity to achieve better system performance through an experimental investigation. In this study, the ejector performance was also explored through the considered numerical modeling approach, and it is found that a reasonable amount of pressure lift could be attainable in the case of different refrigerants. These preliminary modeling results seem to be promising for this novel utilization of ejectors in a wide range of operating conditions. However, numerical results revealed the possibility of optimum ejector efficiency in the case of lower mass entrainment. Overall, for a better system as well as the ejector performance, the entrainment ratio needs to be chosen carefully. Future studies could focus on experimentation with this ejector cycle to gain better insights regarding the ejector's potential to decrease the compressor discharge pressure and temperature.

NOMENCLATURE

COP	Coefficient of Performance	(-)
h	specific enthalpy	($kJ.kg^{-1}$)
$\dot{m}$	mass flow rate	($kg.\ s^{-1}$)
P	pressure	(kPa)
P_{lift}	pressure lift	(kPa)
r	motive flow rate ratio	(-)
s	specific entropy	($kJ.kg^{-1}.K^{-1}$)
T	temperature	(°C)
v	velocity	($m.s^{-1}$)
$\dot{W}$	work rate	(kW)
η	efficiency	(-)
Φ_m	entrainment ratio	(-)

Subscript

diff	diffuser
ej	ejector
in	inlet
is	isentropic
mn	motive nozzle
ms	mixing section
out	outlet
sn	suction chamber
rec	recovered
rec,max	maximum recovery potential

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