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Numerical Study on the Design of Microchannel Evaporators for Ejector Refrigeration Cycles

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ABSTRACT

Two-phase ejectors are devices capable of improving the performance of refrigeration and air conditioning cycles by means of expansion work recovery. Ejector studies often focus on the design and performance of the two-phase ejector and the effect it can have on the performance of the ejector cycle. However, the ejector is not the only component of the system that can have a significant influence on the performance of the ejector cycle. Recent experimental work has shown that the effect of evaporator design on ejector cycle performance can be quite significant, though there is very little research available on the relation between evaporator design and ejector cycle performance. In this paper, a numerical model of a microchannel air-to-refrigerant evaporator, capable of accounting for heat transfer and pressure drop effects, is developed and used to investigate the effect that different evaporator dimensions have on the performance of ejector cycles. The model has been previously validated with experimental data in previous studies. There are two ejector cycles of interest: The standard ejector cycle, in which the ejector is used to directly lift the pressure of fluid exiting the evaporator before the compressor suction, allowing the work recovered in the ejector to directly supplement the compressor work. Figure 1(a) shows a schematic of the standard ejector cycle, which is the most commonly considered vapor compression cycle with an ejector; in this cycle, the ejector is used to directly lift the pressure of fluid exiting the evaporator before the compressor suction, allowing the work recovered in the ejector to directly supplement the compressor work. Figure 1(b) shows a schematic of the ejector recirculation cycle, which uses the ejector pump additional liquid through the evaporator and improve evaporator performance but not to directly supplement the compressor work. Feeding excess liquid to the evaporator eliminates dryout at the end of the evaporator and increases mass flux, which improves refrigerant-side heat transfer throughout the evaporator, though it can also increase refrigerant pressure drop. Lawrence and Elbel (2015a) showed numerically that high-throttling loss fluids (CO₂) should use the ejector to directly supplement compressor power while lower-throttling loss fluids (R134a,
R410A) should use the ejector to improve evaporator performance. Note that the standard ejector cycle can also use the ejector to recirculate excess liquid through the evaporator and improve evaporator performance but at the expense of lower ejector pressure lift (less direct unloading of compressor).

Elbel and Lawrence (2016) provide a review of recent two-phase ejector research and note that ejector studies very commonly focus on the design and performance of the two-phase ejector and the effect that ejector performance has on the cycle. However, recent research has shown that design of other components in the ejector cycle, such as the evaporator, also have a very significant effect on the overall performance of the ejector cycle and how it compares to an expansion valve (DX) cycle. Studies investigating how the evaporator affects ejector cycle performance are quite limited. Pottker and Hrnjak (2015) noted that the standard ejector cycle can also improve refrigerant distribution among parallel tubes in the evaporator because the flash vapor is bypassed around the evaporator; they demonstrated experimentally that bypassing flash vapor can account for half the COP improvement provided by the ejector cycle. Minetto et al. (2013) noted that poor compressor oil return can result in a large amount of oil being recirculated through the evaporator, which can dramatically decrease evaporator performance in the cycles. It should be noted that these experimental investigations, while certainly meaningful, were performed with only a single evaporator; they do not account for the effect that evaporator geometry could be playing in their results. Lawrence and Elbel (2016) experimentally compared the performance of the two ejector cycles discussed here using two different microchannel evaporators. They showed the same cycle with a different evaporator could yield very different performance; the standard ejector cycle performed better with an evaporator with lower refrigerant-side cross-sectional area, while the ejector recirculation cycle performed better with an evaporator with greater refrigerant-side cross-sectional area. However, no study has thoroughly investigated how evaporator geometry affects the performance of ejector cycles. Thus, the objective of this study is to numerically investigate the effect that microchannel evaporator geometry has on the performance of the standard ejector and ejector recirculation cycles. The effect of microchannel port diameter and number of refrigerant-side passes are investigated. Guidelines for optimizing the performance of the evaporator in ejector cycles are provided. The investigation is performed with refrigerants R410A and CO₂ (R744) in order to compare the effect of evaporator design in ejector cycles for different refrigerants.

2. NUMERICAL MODEL

In order to investigate the effect of evaporator geometry on ejector cycle performance, a finite volume model of a microchannel evaporator based on the conservation of mass, momentum, and energy and able to account for heat transfer and pressure drop effects was developed. Each tube was divided into 200 equally sized volumes. Correlations for air-side heat transfer coefficient, refrigerant-side two-phase pressure drop, and refrigerant-side two-phase heat transfer coefficient were selected from the literature. The correlation of Park and Jacobi (2009) for air-side heat transfer coefficient of louvered-fin heat exchangers was used. The two-phase pressure drop multiplier correlation from Friedel (1979) was used to calculate two-phase pressure drop. The flow boiling heat transfer coefficient of Chen (1966) was used with R410A, while the flow pattern-based heat transfer model of Cheng et al. (2006) was used with CO₂ due to the very different flow boiling behavior of this fluid. The effectiveness-NTU method was used to determine heat transfer in each element. Further description and validation of the evaporator model can be found in Lawrence and Elbel (2015a, 2015b).

Figure 1: Cycle layout diagram of (a) standard ejector cycle and (b) ejector recirculation cycle.
The evaporator used in the simulations was a single-slab, microchannel heat exchanger with 24 parallel tubes. The tubes were divided into multiple passes for some cases. The overall height (tube length) and width of the evaporator were 267 and 240 mm, respectively. The each microchannel tube had a thickness and depth of 1.9 and 18.8 mm, respectively, and 19 microchannel ports. The tube pitch was 9.8 mm. The hydraulic diameter of the ports was varied during the simulations with a maximum diameter of 0.98 mm. The fin height and depth were 7.9 and 21 mm, respectively, and the fin pitch was 1.4 mm. The total air-side heat transfer area was 1.58 m$^2$ for all cases, while the refrigerant-side heat transfer and cross-sectional areas varied depending on port hydraulic diameter and number of refrigerant-side passes. Figure 2 shows a schematic of the microchannel heat exchanger described here.

![Schematic of microchannel evaporator](image)

**Figure 2:** Schematic and dimensions of microchannel evaporator used in numerical study.

The remaining system components were simulated using thermodynamic state point analysis. The capacity of the system was set to 1.0 kW for all cases. The condensing temperature was set to 45°C with 1 K of subcooling for R410A. For CO$_2$, the gas cooler outlet temperature was set to 44°C and the gas cooler pressure was set to 100 bar. An internal heat exchanger (IHX) was not included in the CO$_2$ system. Constant compressor isentropic and mechanical efficiencies of 70 and 80 %, respectively, were assumed. The evaporator air inlet state was set to be dry air at 27°C and 0.100 m$^3$s$^{-1}$ flow rate. The ejector was simulated using the 1-D, homogeneous equilibrium model of Kornhauser (1990), which required the assumption of isentropic efficiencies of individual ejector components and the assumption of a constant mixing section pressure. The efficiencies of the motive nozzle, suction nozzle, and diffuser were assumed to be 0.60, 0.60, and 0.55, respectively. The mixing pressure was assumed to be 20 kPa lower than the ejector suction pressure for R410A and 100 kPa lower than the ejector suction pressure for CO$_2$ (corresponding to a 1 K drop in saturation pressure for each fluid). The computer program Engineering Equation Solver or EES (F-Chart Software, 2015) was used to obtain property information for the air and refrigerants and to iteratively solve the generally non-linear system of equations.

## 3. NUMERICAL INVESTIGATION OF EVAPORATOR DESIGN

### 3.1 Effect of Microchannel Port Hydraulic Diameter

Figure 3(a) compares the relative COP of the standard ejector and DX cycles for varying microchannel port hydraulic diameter. The relative COP is defined as the COP at the point of interest divided by the maximum COP of the DX cycle. Figure 3(b) shows the overall heat transfer coefficient-area product (UA) of the evaporator and refrigerant-side pressure ($\Delta P$) of the evaporator in the two cycles. In comparison to the DX cycle, the ejector cycle has lower refrigerant mass flow rate and lower inlet quality to the evaporator due to the expansion vapor being bypassed around the evaporator in the ejector cycle. This results in both lower refrigerant-side heat transfer coefficient (resulting in lower UA) and lower pressure drop in the standard ejector cycle, as seen in Figure 3(b), with
the net result being a lower evaporator outlet pressure (worse overall evaporator performance) in the ejector cycle than in the DX cycle. The standard ejector cycle is still able to improve COP compared to the DX cycle due to the work recovery and pressure lift of the ejector. It can be seen from Figure 3(a) that there is an optimum microchannel port hydraulic diameter that results in maximum COP for each cycle due to the trade-off between increasing pressure drop and increasing UA as hydraulic diameter decreases (refrigerant mass flux increases); the optimal hydraulic diameter is smaller in the ejector cycle because of the lower mass flow rate and lower inlet quality of this cycle. The results presented here show that the amount of COP improvement obtained with an ejector cycle can be very dependent on the design of the evaporator. Additionally, the evaporator designs that result in maximum ejector cycle COP and maximum ejector cycle COP improvement compared to the DX cycle are not the same. Systems with smaller evaporators for the given capacity (resulting in greater refrigerant mass flux) would benefit more from implementing the standard ejector cycle. However, if one is free to design an evaporator for a system, the objective should be to maximize ejector cycle COP, not to maximize COP improvement compared to a baseline cycle.

Figure 3: Comparison of standard ejector and DX cycle (a) relative COP and (b) evaporator UA and refrigerant-side pressure drop with R410A; evaporator outlet state for both cycles is 3 K superheat.

3.2 Effect of Number of Passes and Evaporator Flow Rate
The above section has shown that changing refrigerant mass flux by adjusting hydraulic diameter can be used to maximize the COP of the standard ejector cycle. This section will investigate further improvement of the ejector cycle evaporator by increasing the number of refrigerant-side passes and by overfeeding the evaporator in order to increase refrigerant mass flux. Figure 4 shows four different evaporator configurations that were investigated, labeled as Evaporator A through D. Each evaporator configuration had a different number of refrigerant-side passes and thus a different number of microchannel tubes per pass, as shown in the figure. All evaporators still had 24 total microchannel tubes, and all air-side dimensions were the same for all evaporators. The refrigerant was R410A.

Figure 4: Configurations of Evaporator A (1 refrigerant pass) through D (4 refrigerant passes).
Figure 5(a) compares the relative COP of the standard ejector cycle as a function of circulation number for the four different evaporators, while Figure 5(b) shows the same for the ejector recirculation cycle. The circulation number is defined as the total evaporator liquid flow rate divided by the flow rate of vaporized liquid (Lawrence and Elbel, 2015a), as shown in Equation (1). The circulation number quantifies the rate at which liquid is fed to the evaporator in a liquid overfeed or liquid recirculation system. All the results shown in Figure 5 are for the optimum hydraulic diameter for the given evaporator configuration, circulation number, and system conditions, with the maximum possible hydraulic diameter assumed to be 0.98 mm due to the dimensions of the microchannel tube. All conditions were compared to the same baseline point (maximum DX cycle COP from Figure 3(a)).

\[
\text{Circulation Number} = n = \frac{m_{\text{evap,liquid, total}}}{m_{\text{evap, vaporized}}} \tag{1}
\]

It can be seen from Figure 5(a) that COP increases with increasing number of passes for the standard ejector cycle due to the increase of mass flux (increased heat transfer coefficient). As shown above, decreasing hydraulic diameter can also be used to increase refrigerant mass flux and heat transfer coefficient (increases UA or effectiveness); however, refrigerant-side heat transfer area is decreased at the same time (decreases UA or effectiveness). On the other hand, increasing the number of passes allows mass flux and heat transfer coefficient to be increased without decreasing refrigerant-side heat transfer area, offering the opportunity for greater improvement in COP. It can also be seen from Figure 5(a) that each evaporator has an optimum circulation number that maximizes COP; this will be discussed further in the following section. The standard ejector cycle can achieve up to 9.5 % COP improvement if a larger number of refrigerant passes (4) is used and the hydraulic diameter and circulation number are optimized.

**Figure 5: Relative COP of (a) standard ejector and (b) ejector recirculation cycles as functions of circulation number (evaporator mass flow rate) for different evaporator configurations with R410A.**

Figure 5(b) shows that the ejector recirculation cycle COP increases with increasing circulation number for all evaporators. As mentioned above, increasing the mass flux through liquid overfeed will increase refrigerant heat transfer coefficient and evaporator UA or effectiveness, though refrigerant-side pressure drop will also increase at higher mass flux. At high enough circulation number, excessive pressure drop would eventually outweigh improved heat transfer performance, resulting in an eventual maximum in COP at some optimal circulation number. Interestingly, it is seen that unlike the standard ejector cycle, the highest COP, and thus the best evaporator performance, is obtained with Evaporator C (3 passes) and high circulation number; the higher pressure drop obtained with Evaporator D outweighs the higher heat transfer coefficient. However, at low circulation number (less than 1.7), Evaporator D does yield higher COP than Evaporator C; this shows that first optimizing evaporator design (number of passes and tube diameter) and then optimizing evaporator operation (circulation number) or vice versa will not necessarily yield the maximum obtainable COP. In order to achieve the highest obtainable COP, how the evaporator is designed and how the evaporator is operated must all be optimized simultaneously to find the optimal balance of refrigerant-side heat transfer coefficient and pressure and thus achieve the highest COP.
Comparing Figures 5(a) and 5(b) shows that with Evaporator A, the COP of both cycles increases with increasing circulation number; however, the ejector recirculation cycle achieves higher COP than the standard ejector cycle at high circulation number (above 2.5), while the standard ejector cycle achieves higher COP than the ejector recirculation cycle at lower circulation number. Since the circulation number can be freely set in a system, it would make sense to choose the ejector recirculation cycle and operate at high circulation number when using Evaporator A. On the other hand, the standard ejector cycle achieves higher COP than the ejector recirculation cycle when using any of the other evaporators. This shows that the standard ejector cycle can ultimately achieve higher COP than the ejector recirculation cycle when the evaporator design and circulation number are optimized for each cycle.

Figure 6(a) shows the effectiveness of the different standard ejector cycle evaporator configurations, while Figure 6(b) shows the refrigerant-side pressure drop. As discussed above, the refrigerant-side heat transfer coefficient and evaporator effectiveness (UA) increase with increasing circulation number (increasing mass flux), and a greater number of passes yields further improvement in evaporator effectiveness. Interestingly, the pressure drop does not necessarily increase with increasing circulation number for all conditions. At higher mass flux, as obtained with Evaporators C and D, pressure drop always increases with circulation number, but the pressure drop of Evaporator B shows a minimum at a circulation of 1.3 while the pressure drop of Evaporator A decreases for all circulation numbers. This may be attributed to more liquid and lower average quality throughout the tube. Similar to the numerical results shown here, Lawrence and Elbel (2016) observed experimentally with R410A and a very similar evaporator that overfeeding the evaporator in an ejector cycle to a moderate extent could noticeably improve heat transfer performance without noticeably increasing refrigerant-side pressure drop. For the case of Evaporator D, the increased pressure drop outweighs the increased heat transfer coefficient, causing a decrease in COP with increasing circulation number. It can also be seen from Figure 6(b) that Evaporator B has significantly lower pressure drop than Evaporator A. This is due to the fact that microchannel port hydraulic diameter is optimized for each point. Evaporator A must use much smaller hydraulic diameter compared to Evaporator B in order to improve heat transfer performance, but the smaller hydraulic diameter of Evaporator A also increases pressure drop. Evaporator B has double the flow rate through a given tube compared to Evaporator A and can achieve better heat transfer performance with a larger hydraulic diameter, allowing for a significant reduction in pressure drop. This again indicates that it is better to increase mass flux and improve evaporator heat transfer performance by increasing the number of refrigerant-side passes than by reducing microchannel port hydraulic diameter.

Figure 7(a) shows the effectiveness of different ejector recirculation cycle evaporator configurations, while Figure 7(b) shows the refrigerant-side pressure drop. It can be seen that the trends are similar as those observed with the standard ejector cycle. Evaporator effectiveness increases with increasing circulation number and increasing number of refrigerant-side passes. Additionally, as also observed with the standard ejector cycle, the pressure drop shows a minimum with Evaporator B and continuously decreases with Evaporator A.
Figure 7: Effectiveness and (b) refrigerant-side pressure drop of evaporator in ejector recirculation cycle as a function of circulation number for different evaporator configurations with R410A.

Note that the numerical model did not account for the effect of refrigerant distribution among parallel microchannel tubes in a header; headers that have two-phase refrigerant entering them (headers in DX and ejector recirculation cycles and intermediate headers in standard ejector cycle) may not achieve uniform distribution of liquid and vapor among all parallel tubes, which would decrease the performance of the evaporator. The results presented in this paper do not investigate the effect of air-side dimensions on evaporator and cycle performance. Any change in dimensions that decreases air-side area or air-side heat transfer coefficient would clearly decrease COP for all cycles. Additionally, a decrease in air-side area or heat transfer coefficient will also increase air-side resistance, meaning that refrigerant-side changes would have less of an effect on overall evaporator and cycle performance.

3.3 Guidelines for Ejector Cycle Evaporator Design and Operation

The above results have shown specific evaporator dimensions and liquid feeding rates that optimize the performance of the ejector cycles for the specific system simulated in the model. However, these results are only necessarily applicable to the system of interest here. It would be more useful if the results were generalized in order to aid in the design and operation of an evaporator in any ejector system, as will be discussed in this section. Table 1 summarizes several evaporator performance parameters of the standard ejector and ejector recirculation cycles for the four different evaporator configurations at the point of maximum COP (or highest observed COP) for each case.

Table 1: Evaporator performance parameters of standard ejector and ejector recirculation cycles for four evaporator configurations at point of maximum (or highest observed) COP for each case.

<table>
<thead>
<tr>
<th></th>
<th>Standard Ejector Cycle</th>
<th>Ejector Recirculation Cycle</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$\varepsilon_{\text{evap}}$ (-)</td>
<td>$\Delta P_{\text{evap}}$ (kPa)</td>
</tr>
<tr>
<td>Evap A</td>
<td>0.714</td>
<td>14.1</td>
</tr>
<tr>
<td>Evap B</td>
<td>0.741</td>
<td>8.5</td>
</tr>
<tr>
<td>Evap C</td>
<td>0.749</td>
<td>13.3</td>
</tr>
<tr>
<td>Evap D</td>
<td>0.768</td>
<td>20.1</td>
</tr>
</tbody>
</table>

It can be seen from Table 1 that the evaporator pressure drop at the optimal point for each evaporator configuration was in the range of 8 – 20 kPa for the standard ejector cycle and in the range of 14 – 44 kPa for the ejector recirculation cycle. The overall optimal point for the standard ejector cycle (Evaporator D) had a pressure drop of 20 kPa, while the overall optimal point for the ejector recirculation cycle (Evaporator C) had a similar pressure drop of 26 kPa; these pressure drops correspond to a drop in saturation temperature of approximately 0.6 – 0.7 K. This pressure drop at the optimal point is a consequence of the increased heat transfer coefficient/increased pressure drop
trade-off that has been discussed above. Note that with the ejector recirculation cycle, the pressure drop of 44 kPa (1.2 K drop in saturation temperature) obtained with Evaporator D was seen to result in lower overall COP despite the higher effectiveness with this evaporator. Thus, it seems that there is a limiting pressure drop corresponding to at most 1.0 K drop in saturation temperature for the given fluid that should not be exceeded when designing and operating an evaporator in an ejector cycle.

Despite the similar pressure drop at the optimal points in the two ejector cycles, the cycles do have very different mass flux and circulation number at their optimal points. The performance of the standard ejector cycle is generally optimized when the mass flux through the given evaporator is in the range of 55 – 60 kg m\(^{-2}\) s\(^{-1}\) (for Evaporators B through D), while the performance of ejector recirculation cycle is optimized with a mass flux of 192 kg m\(^{-2}\) s\(^{-1}\) (Evaporator C). Furthermore, the optimal circulation number for the standard ejector cycle with Evaporator D was 1.0 (no overfeed), while the optimal circulation number for the ejector recirculation cycle was always at least 4.0 (very high overfeed) for each evaporator. This means that the way the ejector evaporator should be operated is very different in the two different ejector cycles.

As noted above, the standard ejector cycle can use the work recovered in the ejector to either directly unload the compressor by lifting compressor suction pressure or to improve evaporator performance by liquid overfeed. Figure 5(a) demonstrates that each evaporator has a different optimum circulation number, corresponding to an optimum evaporator flow rate. This optimum is a result of the trade-off increasing evaporator performance and decreasing ejector pressure lift as circulation number increases. Increasing circulation number means more of the work recovered in the ejector is used to pump excess liquid through the evaporator and improve evaporator performance, but less work is available to lift the pressure of the vapor (which provides the capacity) and directly unload the compressor. The highest observed COP with Evaporator D is achieved with a circulation number of 1.0 (no overfeed). Thus, it seems most beneficial to use as much of the work as possible for pressure lift (direct compressor unloading) and use the design of the evaporator rather than overfeed to improve evaporator performance in the standard ejector cycle. Note that this is not the case in the ejector recirculation cycle. The ejector recirculation cycle can only use the ejector to pump excess liquid through the evaporator (not to directly unload the compressor). Thus, it is beneficial to operate with a large amount of overfeed in the ejector recirculation cycle (in combination with an optimized evaporator design) to improve evaporator performance as much as possible.

Finally, the mass flux has been adjusted three different ways in the above results: Number of refrigerant-side passes; port hydraulic diameter; and circulation number. It is best to use the number of refrigerant-side passes to adjust the mass flux first and then use the circulation number in order to further control and achieve the desired mass flux. Comparing Evaporators C and D for the ejector recirculation cycle shows that too many passes results in too high mass flux and too high pressure drop, ultimately lowering cycle performance despite the higher effectiveness. However, comparing Evaporators A and B for each cycle shows that too few passes can severely hurt evaporator effectiveness; Evaporator B can achieve greater effectiveness than Evaporator A with a lower refrigerant-side pressure drop. There is an optimum number of passes to use for each cycle and condition. Because increasing mass flux by decreasing port hydraulic diameter also decreases refrigerant-side heat transfer area, it seems best to leave the hydraulic diameter large and use the number of passes and circulation number to achieve the desired mass flux.

Based on above discussion, the guidelines for the design and operation of microchannel evaporators in ejector cycles can be summarized as follows:

- The evaporator should be designed and operated so that the drop in saturation temperature due to refrigerant pressure drop does not exceed 1.0 K.
- The optimal mass flux seems to be around 55 – 60 kg m\(^{-2}\) s\(^{-1}\) for the standard ejector cycle, while the optimal mass flux for the ejector recirculation cycle is significantly higher (~200 kg m\(^{-2}\) s\(^{-1}\)).
- An increased number of refrigerant-side passes should be used initially to adjust refrigerant mass flux; the amount of liquid overfeed should then be controlled to achieve the desired mass flux; the hydraulic diameter of the ports should generally be kept large to provide greater heat transfer area.
- The ejector recirculation cycle should be operated with a large amount of overfeed and with an optimized number of refrigerant-side passes.
- The standard ejector cycle should be operated with a low amount of overfeed (to maximize pressure lift of the ejector) and should use a large number of refrigerant-side passes to improve evaporator heat transfer.
3.4 Comparison of R410A and CO₂
All of the above results have used R410A as the refrigerant. The model was also used to investigate the effect of microchannel port hydraulic diameter with refrigerant CO₂ for the conditions described above. The relative COP for the standard ejector and DX cycles is shown in Figure 8(a), while the evaporator UA and refrigerant-side pressure drop are shown in Figure 8(b) for the two cycles. It can be seen that similar to the R410A results, there is an optimum hydraulic diameter that maximizes COP, and the optimum hydraulic diameter is smaller for the ejector cycle than for the DX cycle. Up to 6.4% COP improvement is achieved with the ejector cycle when comparing the two cycles at optimum hydraulic diameter. Again, greater COP improvement (up to 9.0%) can be achieved by comparing the two cycles with a hydraulic diameter that is too small for either cycle. The relatively small COP improvement reported here is due to the relatively low efficiency of the simulated ejector. Figure 8(b) shows that the pressure drop behavior with CO₂ is similar to that observed with R410A, though because of its high working pressure, CO₂ is less sensitive to evaporator pressure drop compared to other refrigerants. On the other hand, the evaporator UA trend with the CO₂ cycles differs significantly compared to what was observed with R410A. CO₂ achieves very high heat transfer coefficient at low quality because of its very strong nucleate boiling. This means that the standard ejector cycle, which bypasses vapor around the evaporator and enters at very low quality, can actually achieve better heat transfer performance than the DX cycle despite the lower mass flux, as seen in Figure 8(b) for smaller hydraulic diameter.

![Figure 8: Comparison of standard ejector and DX cycle (a) relative COP and (b) evaporator UA and refrigerant-side pressure drop with refrigerant CO₂; evaporator outlet state for both cycles is 3 K superheat.](image)

It can be seen from Figure 8(a) that the COP of the CO₂ ejector cycle is not as sensitive to hydraulic diameter as the COP of the R410A ejector cycle was. This may be due to the large ejector work recovery and pressure lift in the CO₂ ejector cycle; the COP of the ejector cycle is dominated by the pressure lift of the ejector cycle and the effect of evaporator performance is less significant in determining overall cycle performance. It can be concluded from this that evaporator design plays a critical role in the performance of ejectors cycles for low pressure refrigerants (R410A, hydrocarbons, ammonia, R134a) but may not be as critical when operating the CO₂ standard ejector cycle; issues such as proper cycle control and design of efficient ejectors would have a more significant effect on the overall performance of a CO₂ ejector cycle.

4. CONCLUSIONS

This paper has presented the results of a numerical investigation on the design of microchannel evaporators in ejector cycles. It has been seen that ejector cycle performance can be very sensitive to evaporator design (number of refrigerant-side passes and port hydraulic diameter) and operation (liquid feeding rate or circulation number). It was seen that for the standard ejector cycle, the ejector should be used for pressure lift (direct compressor unloading) rather than liquid overfeed, and the design of the evaporator should be used to improve evaporator performance. On the other hand, for the ejector recirculation cycle, the ejector should be used to provide a large amount of overfeed to
the evaporator and improve evaporator performance in combination with an optimized evaporator design. The standard ejector cycle could ultimately achieve higher COP than the ejector recirculation cycle once evaporator design was optimized for each cycle with the R410A system considered here; however, the ejector recirculation cycle would be expected to perform better at conditions of lower ambient temperature or with a lower work recovery fluid (R134a, ammonia), as discussed by Lawrence and Elbel (2015a). General guidelines for the design and operation of an evaporator in an ejector cycle have been extracted from the numerical results and discussed above. Future work should focus on expanding the numerical investigation to systems with different capacities (which would require very different flow rates), different refrigerants, and different heat exchanger types (round tube) in order to provide more general guidelines on the design and operation of the evaporator for any ejector system.

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NOMENCLATURE

Symbols and Abbreviations

\[ \begin{align*}
    COP & : \text{Coefficient of Performance (-)} \\
    d_h & : \text{hydraulic diameter (mm)} \\
    DX & : \text{expansion valve cycle} \\
    m & : \text{mass flow rate (kg s}^{-1}) \\
    n & : \text{circulation number (-)} \\
    T & : \text{temperature (K)} \\
    UA & : \text{overall heat transfer coefficient-area product (kW K}^{-1}) \\
    \Delta P & : \text{pressure drop (kPa)} \\
    \varepsilon & : \text{effectiveness (-)} \\
\end{align*} \]

Subscripts

\[ \begin{align*}
    \text{jec} & : \text{ejector} \\
    \text{evap} & : \text{evaporator}
\end{align*} \]

REFERENCES


