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Experimental and Numerical Study on the Performance of R410A Liquid Recirculation Cycles with and without Ejectors

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ABSTRACT

Liquid recirculation is a method of improving evaporator performance by supplying the evaporator with more liquid than is evaporated. The excess liquid, which is generally provided by means of an externally-driven, mechanical pump, eliminates dryout and increases overall heat transfer coefficient in the evaporator. Liquid recirculation is common in large-scale industrial refrigeration systems, where the additional cost and complexity of the pump is justified by the large size of the system. However, liquid recirculation has yet to gain widespread application in small-scale systems, such as residential or automotive air conditioning. The use of an ejector to provide the recirculation effect rather than a mechanical pump can help liquid recirculation systems gain consideration for small-scale systems. However, the performance of liquid recirculation cycles with and without ejectors for small-scale systems has yet to be thoroughly investigated.

This paper presents the results of an experimental and numerical study of the effect of liquid recirculation on the performance of air condition cycles. The refrigerant R410A, commonly considered for residential systems, is used as the working fluid. A numerical model of a liquid recirculation cycle, capable of accounting for heat transfer and pressure drop effects in the evaporator, was developed and used to predict the effect that recirculation ratio (ratio of total evaporator mass flow rate to mass flow rate of vaporized liquid) has on cycle COP. A liquid recirculation cycle with a mechanical pump was constructed and tested at various evaporator mass flow rates in order to see experimentally the effect of recirculation ratio on the COP of the cycle. Cycles in which the work recovered by the expansion process in the ejector is used to provide the pumping effect, instead of the mechanical pump, are then presented and analyzed, and the ability of the ejector to act as a viable replacement for the mechanical pump in liquid recirculation systems is discussed.

1. INTRODUCTION

Liquid recirculation, also commonly called liquid overfeed, is a method often employed in large-scale industrial refrigeration systems in order to improve evaporator performance. In a liquid recirculation cycle, more liquid is sent to the evaporator than is actually evaporated, meaning that the outlet of the evaporator is a two-phase fluid rather than a superheated vapor, as is the case in direct expansion (DX) system operation. The absence of liquid at the outlet of the evaporator (dryout) results in decreased refrigerant-side heat transfer coefficient and increased refrigerant-side pressure drop in the dryout region. Thus, an evaporator that is overfed with liquid can result in increased refrigerant-side heat transfer coefficient and potentially decreased refrigerant-side pressure drop (at lower mass fluxes). Feeding only liquid into the evaporator can also improve refrigerant distribution for evaporators that have inlet headers. As a result of these effects, liquid recirculation can result in improved evaporator performance, higher evaporation pressure, and higher system COP compared to a DX cycle.

The amount of liquid that is being fed to the evaporator can have a significant effect on how much COP improvement can be gained from a liquid recirculation cycle. Higher mass flow rate in the evaporator generally results in higher heat transfer coefficient but also higher pressure drop. Thus, there is an optimum evaporator mass flow rate that will result in the maximum recirculation cycle COP. A common way to quantify the amount of liquid overfeed is with the recirculation ratio, defined as the total evaporator mass flow rate to the mass flow rate of vaporized refrigerant and shown in Equation (1). If the inlet state of the evaporator is close to a saturated liquid state, then the recirculation ratio is approximately the inverse of the evaporator outlet quality. Another way to quantify the amount of liquid overfeed is with the overfeed ratio, defined as the ratio of liquid to vapor mass flow rate at the evaporator outlet and shown in Equation (2).

$$R = \frac{\dot{m}_{evap,total}}{\dot{m}_{evap,vaporized}} = \frac{\dot{m}_{evap,total}\left(h_v\left(P_{evap,out}\right) - h_{evap,in}\right)}{\dot{Q}_{evap}} \approx \frac{1}{x_{evap,out}} \tag{1}$$

$$OR = \frac{\dot{m}_{evap,out,liquid}}{\dot{m}_{evap,out,vapor}} = \frac{1 - x_{evap,out}}{x_{evap,out}} \approx R\left(1 - x_{evap,out}\right) \tag{2}$$

Generally, an externally driven mechanical pump is used to provide the liquid recirculation effect necessary for this type of cycle. A recirculation cycle using a mechanical pump is called a forced recirculation cycle and is shown in Figure 1. The pump draws liquid from a liquid-vapor separator and supplies it to the inlet of the evaporator. The outlet of the evaporator then returns to the liquid-vapor separator, allowing for two-phase flow at the outlet of the evaporator. The pumping effect in a recirculation cycle can also be provided by an ejector (as discussed in this paper), gravity, or high-pressure vapor from the discharge of the compressor. Lorentzen (1958) provides a summary of the different systems that can be used to overfeed the evaporator with liquid. Stoecker (1998) provides a detailed overview of the design and operation of liquid recirculation systems.

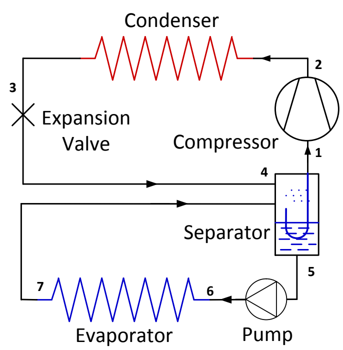

Figure 1: Forced liquid recirculation cycle in which a mechanically driven pump is used to provide excess liquid to the evaporator.

Despite the being a commonly implemented concept for refrigeration applications, there is limited quantitative information available in the open literature on how much liquid should be recirculated and how much improvement (capacity and COP) can be achieved through liquid recirculation. Giuliani *et al.* (1999) reported an experimental COP improvement of 13 % comparing a forced liquid recirculation cycle to a DX cycle with R134a as the working fluid; recirculation ratio was fixed at 2 for all of their tests. They also showed that a forced recirculation cycle with an R134a/R32 mixture resulted in a 12 % decrease in COP compared to the DX cycle due to the composition shift of the zeotropic mixture in the liquid-vapor separator. Bivens *et al.* (1997) showed that the same composition shift problem would be expected in a liquid recirculation system with R407C as the refrigerant.

As mentioned above, there is limited information on liquid recirculation cycles available in the open literature. It is unclear from the literature what values of recirculation ratio should be used in these cycles and how much performance improvement can be expected from a recirculation cycle. In order to determine reasonable recirculation ratios and the corresponding COP improvement, a numerical model will be used to investigate the performance of an R410A forced liquid recirculation cycle. Experimental data obtained with an R410A forced recirculation cycle will also be presented and discussed.

2. ANALYSIS

2.1 Numerical Model for Prediction of Recirculation Cycle Performance

The ASHRAE Handbook (2002) provides recommendations for recirculation ratio for various fluids, as shown in Table 1. However, even for a single fluid, the recirculation ratio that results in best cycle performance depends on a variety of factors such as evaporator type and dimensions as well as system operating conditions. It is important to understand how various factors affect the optimum recirculation ratio and the corresponding potential for COP improvement.

Table 1: Recirculation ratios for various systems recommended by ASHRAE (2002).

Refrigerant	Recirculation Ratio
Ammonia (downfeed, large diameter tubes)	6 to 7
Ammonia (upfeed, small diameter tubes)	2 to 4
R22	3
R134a	2

In order to predict the COP improvement that can be obtained with a liquid recirculation cycle and to investigate the effects that certain parameters have on recirculation cycle performance, a numerical model of the forced recirculation cycle was developed using the program Engineering Equation Solver or EES (F-Chart, 2013). This program was used to iteratively solve the set of non-linear equations and to obtain R410A property information for required by the model. A finite volume model of a microchannel evaporator able to account for varying refrigerant heat transfer coefficient and pressure drop was developed. Table 2 shows the system parameters that were used in the numerical model. The condensing temperature as well as the efficiencies of the compressor and the mechanical pump were fixed for all points. It was also assumed that distribution was uniform in the inlet header of the evaporator, meaning that each of the parallel microchannel tubes received the same mass flow rate and inlet quality. R410A was used as the refrigerant. Correlations for air-side heat transfer coefficient, refrigerant-side two-phase heat transfer coefficient, and refrigerant-side two-phase pressure drop were selected from the open 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 from Friedel (1979) was used to calculate two-phase pressure drop, and the flow pattern map based correlation for flow boiling heat transfer coefficient from Wojtan *et al.* (2005) was used to calculate the two-phase heat transfer coefficient in the evaporator.

Table 2: Operating parameters of simulated forced recirculation cycle.

Parameter	Value
Cooling capacity	2.0 kW
Condensing temperature	45^{o}C
Evaporator air temperature	27^{o}C
Evaporator air flow rate	150 L/s
Pump isentropic efficiency	0.30
Compressor isentropic efficiency	0.70
Compressor mechanical efficiency	0.80

The evaporator investigated was a single-pass microchannel heat exchanger with 23 parallel tubes. Each microchannel tube had 19 ports, each with a hydraulic diameter of 0.54 mm. The overall dimensions of the evaporator were 265 mm by 241 mm (height by width); the heat exchanger had a depth of approximately 21 mm. Louvered fins were used on the air-side to improve heat transfer; the fin pitch was 1.4 mm (approximately 18 fins per inch). The total refrigerant-side heat transfer area was 0.196 m^2, the refrigerant-side cross-sectional area was 100.1 mm^2, and the total air-side heat transfer area was 1.573 m^2. Further details of the evaporator dimensions can be found in Hoehne and Hrnjak (2004).

2.2 Numerical Results

The performance of the forced recirculation cycle, compared to a DX cycle with the same system conditions and evaporator geometry and an evaporator outlet superheat of 5 K, is shown in Figure 2(a). The COP of the recirculation cycle considers both the compressor power and pump power, as shown in Equation (3), though the pump power was nearly negligible. Results are reported in terms of the COP Ratio, defined as the ratio of the COP of the recirculation cycle over the COP of the DX cycle and shown in Equation (4). It can be seen that COP Ratio increases with recirculation ratio with a maximum COP improvement upwards of 10 %, though the COP Ratio does not increase much after a recirculation ratio of approximately 2.3. Figure 2(b) shows the corresponding evaporator pressure drop and UA value for the system. The evaporator UA value shows similar trend as the COP Ratio, meaning that the COP improvement is mainly being caused by improvement of UA value (due to the recirculation effect). The pressure drop in the evaporator seems to be small in this case and likely does not have a significant effect on the COP Ratio, though the effect of pressure drop could be more significant in a different evaporator. Generally, pressure drop is proportional to the square of mass flow rate (recirculation ratio), but because of the decreasing average quality in the tubes as recirculation ratio increases, the evaporator pressure drop appears to be almost linearly related to recirculation ratio.

$$COP_{Recirc} = \frac{\dot{Q}_{evap}}{\dot{W}_{cp} + \dot{W}_{pump}} \tag{3}$$

$$COP\ Ratio = \frac{COP_{Recirc}}{COP_{DX}} \tag{4}$$

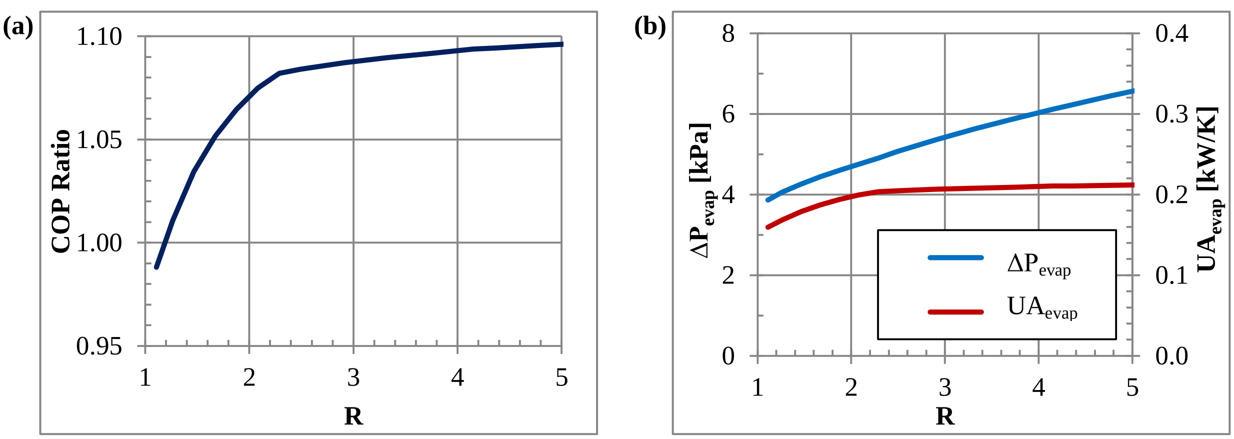

Figure 2: Performance of R410A forced liquid recirculation cycle at varying recirculation ratio shown in terms of (a) COP Ratio and (b) pressure drop and UA value of evaporator.

Figure 2(a) also shows that at very low recirculation ratios (less than approximately 1.2) the COP Ratio is less than unity, meaning that the recirculation cycle actually has lower COP than the DX cycle. This is due to the fact that at very low recirculation ratio, the UA value of the recirculation cycle can be lower than that of the DX cycle. The absence of vapor at the inlet of the evaporator in the recirculation cycle yields lower velocity at the beginning of the evaporator, which decreases refrigerant-side heat transfer coefficient compared to the DX cycle at the beginning of the evaporator. Additionally, the greater specific enthalpy difference across the evaporator in the recirculation cycle (due to liquid rather than two-phase fluid entering the evaporator) results in lower mass flow rate for the same capacity compared to the DX cycle (at very low recirculation ratio); this results in lower mass flux for the recirculation cycle, which also decreases refrigerant-side heat transfer coefficient. Thus, there are two different effects that both reduce refrigerant-side heat transfer coefficient and UA value of the evaporator in the recirculation cycle, resulting in lower evaporation pressure and lower COP of the recirculation cycle compared to the DX cycle (at very low recirculation ratio). Note that once recirculation ratio becomes large enough, the high mass flux increases the refrigerant-side heat transfer coefficient for the recirculation cycle above that of the DX cycle, resulting in COP improvement. Note also that these results do not account for any potential maldistribution of liquid in the evaporator inlet header due to the presence of vapor in the DX cycle, though not all evaporator designs have a distribution problem.

The amount of COP improvement that can be achieved is dependent on the geometry and conditions of the evaporator. If the hydraulic diameter of the microchannel ports is reduced from 0.54 mm to 0.39 mm (refrigerant-side cross-sectional area is approximately halved), the mass flux for a given recirculation ratio and for the DX cycle will essentially double. The results for a smaller hydraulic port diameter of 0.39 mm (larger mass flux) are compared to the results for 0.54 mm hydraulic diameter in Figure 3; the air-side dimensions of the evaporator as well as all system operating conditions were unchanged. It can be seen in Figure 3 that the evaporator with a larger mass flux can only achieve less than 3 % COP improvement. This is because the refrigerant-side heat-transfer coefficient, and thus the UA value, of the evaporator are already higher in the DX cycle due to the higher initial mass flux; there is less opportunity to improve UA value by increasing mass flux (with recirculation) when initial mass flux is higher to begin with. As a result, evaporators that have a smaller refrigerant-side cross-sectional area, and thus a higher mass flux, offer less opportunity for improvement with liquid recirculation cycles. On the other hand, Figure 3 also shows that the evaporator with the larger mass flux also experiences lower penalty at very low recirculation ratios. This is again due to the fact that the initial mass flux is already higher, and the decrease in UA value at low recirculation ratio is not as significant when mass flux is higher.

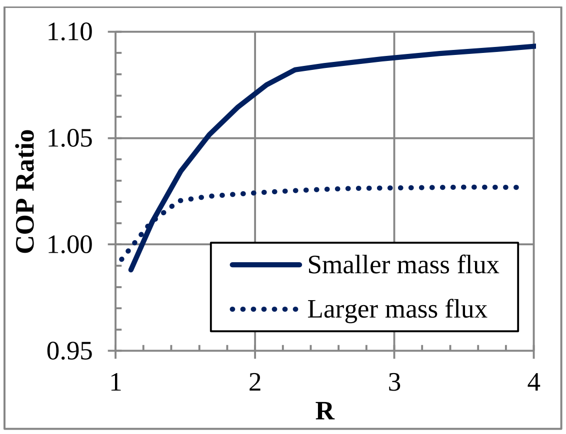

Figure 3: Effect of microchannel port diameter (or mass flux) on COP Ratio of R410A forced liquid recirculation cycle; larger initial mass flux means less opportunity for COP improvement but also reduced penalty for operating at very low entrainment ratio.

2.3 Liquid Recirculation Cycles with Ejectors

It can be seen from the above analysis that a liquid recirculation cycle offers the potential for significant COP improvement in air conditioning and refrigeration systems, making it an attractive method for COP improvement in these systems. As stated above, liquid recirculation systems are commonly found in industrial refrigeration applications. However, these systems have yet to gain widespread attention for use in smaller systems, such as residential refrigeration or air conditioning, likely due to the added cost and complexity of using a pump to circulate liquid, which may not be justified for such small systems. However, if the liquid were able to be circulated by another means, such as an ejector, then it is possible that liquid recirculation could be applied to smaller, more common systems. Figure 4(a) shows the ejector recirculation cycle, in which the ejector driven by high-pressure liquid from the condenser is used to recirculate liquid from the separator back to the inlet of the evaporator. This cycle was originally patented by Phillips (1938).

Because of the increased interest in ejector cycles in recent years, this cycle has received some attention in the open literature. Dopazo and Fernández-Seara (2011) performed an experimental investigation of an ammonia ejector recirculation cycle with a plate evaporator and observed recirculation ratios between 2 and 4. Minetto *et al.* (2014) reported COP improvement up to 13 % with a CO_2 ejector recirculation cycle in which the vapor generated during the expansion process was bypassed around the plate-and-fin evaporators; their ejector recirculation cycle operated with a recirculation ratio very close to unity. Li *et al.* (2014) investigated an R134a ejector recirculation cycle in which the ejector was used to recirculate liquid through a falling film chiller (evaporator); they observed an optimum recirculation ratio of 1.13. Note that the evaporator design used in the study of Li *et al.* (2014) is very different from the evaporators generally used in small-scale applications, and the optimum recirculation ratio for R134a systems with other evaporator designs may be higher than what was shown in that study.

The standard two-phase ejector cycle, shown in Figure 4(b) and originally patented by Gay (1931), has received considerable attention in recent years because of its ability to improve COP by means of expansion work recovery in the ejector. CO_2 has received the majority of attention in two-phase ejector studies because of its large potential for improvement through expansion work recovery. Elbel (2011) provides a review of CO_2 ejector studies. Generally, the evaporator in this cycle is assumed to operate with superheated or saturated vapor at the exit. However, there is no need to evaporate all of the liquid in the evaporator in this cycle, as the ejector is capable of pumping liquid, vapor, or a two-phase fluid. Thus, if excess liquid is fed to the evaporator, the standard two-phase ejector cycle can also be used as a liquid recirculation cycle. Disawas and Wongwises (2004) reported COP improvement of 5 to 10 % with R134a when operating the standard two-phase ejector cycle as a liquid recirculation cycle; they also observed that the ejector was capable of providing recirculation ratios between 1.5 and 3.5.

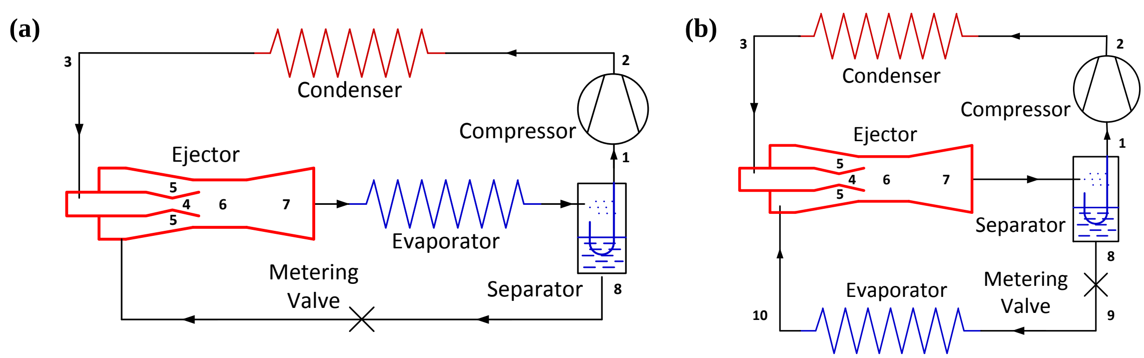

Figure 4: Liquid recirculation cycles in which an ejector is used in place of a mechanical pump: (a) Ejector recirculation cycle in which an ejector is used to circulate excess liquid through the evaporator; (b) standard two-phase ejector cycle, which can also be used as a recirculation cycle if the ejector is used to pump excess liquid through the evaporator.

Having presented three different recirculation cycles, it may be interesting to investigate different practical aspects of the cycles that may be important for the cycles' application to small-scale systems. If a reliable ejector can be designed that can provide proper entrainment over a range of conditions, then it would be more desirable to use an ejector to pump the liquid than a mechanical pump; this gives a clear advantage to the two ejector cycles, especially for small-scale applications. The literature presented above shows that two-phase ejectors can indeed be designed to entrain a significant amount of liquid and achieve reasonable recirculation ratios.

The forced liquid recirculation cycle and the standard ejector cycle should both result in little or no vapor at the inlet of the evaporator; the ejector recirculation cycle, however, must have vapor present at the inlet of the evaporator due to the expansion process taking place in the ejector. There are two effects that this vapor could have on the evaporator performance: First, the presence of vapor could cause significant maldistribution in the evaporator header, which would hurt evaporator performance; second, the presence of vapor will increase the velocity at the beginning of the evaporator tubes, which for many fluids will result in higher refrigerant-side heat transfer coefficient (though pressure drop would also increase). Thus, the effect of the vapor at the inlet of the evaporator is not entirely clear and may depend of the evaporator design.

The optimum recirculation ratio (resulting in the highest COP for the given conditions) may not necessarily be the same for each of the cycles. The forced recirculation and ejector recirculation cycles would be expected to have similar optimum recirculation ratios, though the presence of vapor at the evaporator inlet could change the optimum point slightly in the ejector recirculation cycle. Note that in the ejector recirculation cycle, the compressor suction pressure is essentially set by the evaporation pressure, and the pressure lift of the ejector is only used to overcome any pressure drop in the evaporator and liquid recirculation loop; any additional pressure lift that the ejector may provide is lost in the metering valve.

This is different from the standard ejector cycle; in the standard ejector cycle, overfeeding the evaporator with additional liquid can increase the evaporation pressure, but the ejector can still provide further benefit to the cycle by lifting the compressor suction pressure above the evaporation pressure. Thus, the ejector serves two purposes in the standard ejector cycle: Improve evaporator performance by overfeeding the evaporator with liquid (resulting in higher evaporator pressure) and increase the compressor suction pressure above that of the evaporator. Greater mass flow through the evaporator generally means greater increase in evaporator pressure but also lower pressure increase provided by the ejector; thus, the trade-off between higher evaporator pressure and lower ejector pressure increase may result a maximum COP at some optimum recirculation ratio. It is possible that the optimum recirculation ratio for the standard ejector cycle may be lower than that of the other two recirculation cycles.

One potential problem that is often encountered with ejector cycles is what happens to the cycle if the ejector does not work properly (meaning the ejector does not entrain sufficient or any mass) due to either poor ejector design or operation at conditions not favorable to the ejector (off-design operation). It can be seen from Figure 4(b) that if the ejector in the standard ejector cycle does not entrain mass, the system will lose some or all of its flow through the evaporator, resulting in severely reduced capacity and COP. On the other hand, if the ejector in the ejector recirculation cycle does not pump liquid from the separator vessel, there will still be mass flow through the evaporator (being pulled through by the compressor); thus, even if the ejector does not work, the ejector recirculation would not be expected to perform any worse than a DX cycle. Note that the forced recirculation cycle would not be expected to encounter this problem, as the pump would be expected to work regardless of the system operating conditions.

Finally, because of the presence of the separator vessel in each of the cycles, not all flow ultimately returns to the compressor, meaning that there may be difficulties returning oil to the compressor. If the oil tends to collect with the liquid phase in the separator vessel in each of the cycles, then it may just be continually circulated through the evaporator and fail to ever return to the compressor. Failure to return sufficient oil to the compressor can damage the compressor, and circulating too much oil through the evaporator can degrade the performance of the evaporator. The different advantages and disadvantages of the three different liquid recirculation cycles discussed here are summarized in Table 3 below.

Table 3: Comparison of different aspects of the three different liquid recirculation cycles presented above.

	Forced Recirculation Cycle	**Standard Two-Phase Ejector Cycle**	**Ejector Recirculation Cycle**
Liquid pumping device	Mechanical pump	Ejector	Ejector
Vapor present at evaporator inlet?	No vapor	Little or no vapor	Yes
Optimum recirculation ratio	High	Low	Highest (no $P_{lift,ejec}$ benefit)
Reduced capacity (below DX) at ejector off-design conditions?	N/A	Yes	No
Oil trapped in evaporator loop?	Possible	Possible	Possible

3. EXPERIMENTAL FACILITY AND METHODS

The experimental test facility was modified from an existing system used for previous ejector experiments in order to construct a forced liquid recirculation system. A full description of the previous ejector system is given by Lawrence and Elbel (2014). Two closed-loop wind tunnels house the condenser and the evaporator. The evaporator was of microchannel design similar to the evaporator described in the numerical modeling section above. The experimentally tested evaporator had 31 parallel microchannel tubes instead of 23 and a microchannel port hydraulic diameter of 0.77 mm; all other dimensions were the same. The condenser was a round-tube-plate-fin heat exchanger. Variable speed blowers and electric heaters allowed for control of the air flow rate and the air inlet temperature to the condenser and evaporator. A single-piston, rotary-type compressor suitable for use with R410A with a capacity of approximately 2 kW was used on the refrigerant-side. Type-T thermocouples, differential pressure transducers, and flow nozzles were used for air-side measurements. Type-T thermocouples, differential and absolute pressure transducers, and Coriolis-type mass flow meters were used for refrigerant-side measurements. A power transducer was used to measure the electrical power supplied to the compressor. A liquid-vapor separator with a height of approximately 18 cm, a diameter of approximately 14 cm, and an internal volume of approximately 2800 cm^3 was used in the system. The separator had a sight-glass that allowed for visualization of the liquid and vapor; it was ensured during tests that the liquid level was sufficiently high to prevent vapor from exiting at the liquid port of the separator but not so high that an excessive amount of liquid would exit at the vapor port. A positive displacement pump with a maximum flow rate of 21 L/min and a maximum pressure rise of approximately 3 bar was used to circulate the liquid refrigerant through the evaporator. Detailed information of the uncertainty of the instruments and the system can be found in Elbel and Hrnjak (2008).

Because of the presence of the two-phase fluid at the refrigerant-side outlet of the evaporator, the specific enthalpy could not be determined based on temperature and pressure readings. Thus, it was not possible to directly obtain two independent energy balances when operating the forced liquid recirculation cycle described here. Thus, the cooling capacity and COP of the system were determined via air-side measurements. With the air-side evaporator capacity known, the refrigerant outlet enthalpy and the recirculation ratio in Equation (1) were able to be determined.

Data was collected at steady-state conditions on the forced recirculation cycle. Air flow rate and inlet temperature (in both the condenser and evaporator) and the compressor speed were kept the same for all tests. The condenser air inlet temperature was 35°C, and the air flow rate was 300 L/s. The evaporator air inlet temperature was 27°C, and the air flow rate was 100 L/s; the evaporator was run under dry conditions. The compressor speed was 40 Hz (approximately half of the maximum speed and capacity of the compressor). R410A was used as the working fluid. The system was run at several different recirculation ratios by varying the pump speed. Difficulty was encountered in obtaining stable pump operation at very low pump speed, so results are reported for higher recirculation ratios.

4. EXPERIMENTAL RESULTS

The results of the experimental investigation of a forced liquid recirculation system with R410A are shown in Figure 5. Figure 5(a) shows that at the fixed compressor speed, the capacity does not seem to vary significantly, as it was approximately 0.9 kW for all the data points presented here.

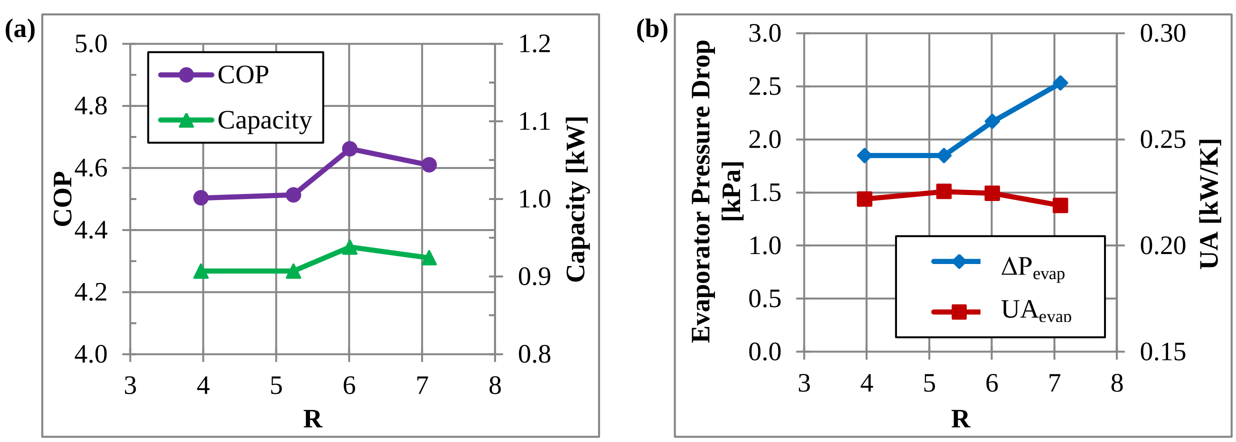

Figure 5: Experimental results for the performance of the forced liquid recirculation cycle with a microchannel evaporator and R410A as the refrigerant presented in terms of (a) COP and evaporator capacity and (b) evaporator pressure drop and evaporator UA value.

Figure 5(a) also shows the COP of the system for several different recirculation ratios. It can be seen that at the range of recirculation ratios tested, the COP does not offer much variation, as would be expected given the high values of recirculation ratio, though the COP does seem to show a slight optimum near a recirculation ratio of approximately 6. Note that the optimum recirculation ratio for this system would be likely to change if the system were tested at higher capacity, as the capacity used for these tests (less than 1 kW) is small for the microchannel evaporator used in the system. Note also that the COP seems to follow a similar trend as the capacity, as the compressor power was very similar for all tests.

Figure 5(b) shows the performance of the evaporator in the forced recirculation cycle in terms of pressure drop and UA value. It can be seen from Figure 5(b) that the evaporator UA value does not show significant variation over the range of entrainment ratios, as would be expected at higher recirculation ratios. The numerical results above show that the UA value does not change significantly after a recirculation ratio of about 2.3. The UA value seems to drop for the highest recirculation ratio, meaning that refrigerant-side heat transfer coefficient actually decreases with increasing recirculation ratio. The reason for this is not clear, though it may be due to the presence of a very large amount of liquid throughout the evaporator at such high recirculation ratio, resulting in very low velocity throughout the evaporator. It can also be seen from Figure 5(b) that the evaporator pressure drop increases with increasing recirculation ratio, due to the increased mass flow rate at higher recirculation ratio. The increasing pressure drop combined with what appears to be slightly decreasing UA value at very high recirculation ratios (greater than 6 for this specific system and capacity) yields an optimum recirculation ratio that results in the highest COP.

5. SUMMARY AND CONCLUSIONS

This study has reported numerical and experimental results on the performance of a forced liquid recirculation cycle with R410A as the refrigerant and has discussed the possibility of implementing liquid recirculation cycles that use an ejector to recirculate the liquid rather than a mechanical pump. A finite volume model of the evaporator capable of accounting for pressure drop and heat transfer coefficient effects was used to show the effect of recirculation ratio on the performance of the forced recirculation cycle. It was seen that the forced liquid recirculation cycle with a microchannel evaporator could achieve upwards of 10 % COP improvement, though when the mass flux was doubled, the COP improvement was reduced to less than 3 %. Thus, it can be concluded that systems that use refrigerants and heat exchangers in which the baseline (or DX) mass flux is low can gain the most benefit from liquid recirculation cycles.

A forced liquid recirculation cycle was constructed and tested; a range of recirculation ratios was tested by varying the pump speed. The COP did not show much variation over the range of recirculation ratios, but the recirculation ratios achieved experimentally were in the range of 4 to 7, where the COP would not be expected to show much variation. There was a slightly higher COP near a recirculation ratio of about 6, meaning that due to the competing effects of higher pressure drop and higher UA value as recirculation ratio increases, the COP does show a maximum at some optimum recirculation ratio. It should be noted that the experimentally observed optimum recirculation ratio may not be typical of all R410A recirculation systems, as the system capacity was small for the microchannel evaporator used in the system.

In order to make liquid recirculation more suitable for small-scale applications, liquid recirculation cycles in which the mechanical pump was replaced by an ejector were presented. It was seen that the recirculation cycles considered in this paper each offer their own advantages and disadvantages, but it was not clear if there was a recirculation cycle that was the best choice in every situation. The best choice for recirculation cycle may depend on the refrigerant and application being considered. Further analysis of the different options for recirculation cycles should be a topic of additional study.

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NOMENCLATURE

Symbols and Abbreviations

COP	coefficient of performance [-]
DX	direct expansion
h	specific enthalpy [kJ/kg]
$\dot{m}$	mass flow rate [g/s]
OR	overfeed ratio [-]
P	pressure [kPa]
$\dot{Q}$	cooling capacity [kW]
R	recirculation ratio [-]
UA	product of overall heat transfer coefficient and area [kW/K]
$\dot{W}$	power
x	quality [-]

Subscripts

cp	compressor
evap	evaporator
in	inlet of component
out	outlet of component
Recirc	forced recirculation cycle
v	saturated vapor

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