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Neal Lawrence
ndlawre2@illinois.edu

Stefan Elbel

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Experimental and Analytical Investigation of Automotive Ejector Air-Conditioning Cycles Using Low-Pressure Refrigerants

Neal LAWRENCE, Stefan ELBEL*

Department of Mechanical Science and Engineering,
University of Illinois at Urbana-Champaign,
1206 West Green Street, Urbana, IL, 61801, USA
Phone: (217) 244-1531, Fax: (217) 333-1942, Email: elbel@illinois.edu

* Corresponding Author

ABSTRACT

In recent years, ejectors have received much attention because of their ability to reduce throttling losses and increase the efficiency of stationary and mobile air-conditioning systems. While much of the initial research was carried out with high pressure fluids, such as carbon dioxide, it was soon discovered that ejectors can also offer significant advantages in systems that utilize low pressure working fluids. Because throttling losses are generally less significant for low pressure refrigerants, the dominant improvement mechanism in these systems comes mostly from the system’s low pressure side, where reduced mass flow rates and higher suction pressures cause less pressure drop. Systems with microchannel evaporators can benefit the most because using an ejector can significantly reduce refrigerant mal-distribution and yield better thermodynamic utilization of the available surface areas.

This paper presents experimental and numerical results obtained with a realistic air-conditioning system for a small size vehicle. Numerical simulations with R134a and R1234yf were performed to determine the effect that inefficiency of the liquid-vapor separator in the standard two-phase ejector cycle could have on the performance of the cycle. The results show that a realistic amount of separator inefficiency can have a significant impact on the performance of the two-phase ejector cycle, and that R1234yf gains 1 to 3 % greater benefit from the ejector than R134a but is impacted more severely by separator inefficiency. A two-phase ejector system concept was realized in which evaporation occurred on two different temperature levels without the use of a liquid-vapor separator. Better matched temperature glides between air and refrigerant streams result, which drives the reduction of exergetic losses and further increases system efficiency. Conventional and ejector system tests were carried out with R134a refrigerant. The ejector cycle used was able to improve the COP by as much as 10 % over a conventional cycle. However, this relatively large improvement is partly due to the fact that the cycle setup and operating conditions were slightly more favorable to the ejector cycle.

1. INTRODUCTION

One of the thermodynamic losses of the conventional vapor-compression refrigeration cycle is the isenthalpic throttling of the high pressure fluid to the lower pressure required for evaporation. This isenthalpic expansion decreases the COP of the vapor-compression cycle, compared to the Carnot cycle, which has an isentropic expansion, due to the decrease in cooling capacity and the lost opportunity for expansion work recovery. A two-phase ejector is a combined pump-expansion device that can be used to recover this expansion work and improve the COP of the cycle. A two-phase ejector takes a high pressure (motive) flow of fluid and expands it through a nozzle to increase its momentum. The momentum of the motive flow is then used to increase the momentum of a lower pressure (suction) flow through momentum transfer. The two flows mix and are then decelerated in the mixing section and diffuser to increase pressure. The result is a two-phase flow exiting the ejector at a pressure higher than the initial pressure of the suction flow; the ejector provides a pressure increase to the lower pressure...
flow. Ejectors and other types of expansion devices allow the expansion process to approach an isentropic expansion, reducing the thermodynamic loss of the cycle.

Gay (1931) proposed a refrigeration cycle that used a two-phase ejector to recover some of the expansion work that would have otherwise been lost in the conventional cycle. Kornhauser (1990) developed a thermodynamic model for the expansion process in the ejector and found theoretical COP improvements for the ejector cycle of 21% over the conventional cycle with R12 as the working fluid. Elbel and Hrnjak (2008) showed experimentally that two-phase ejectors can be beneficial for refrigeration systems with high pressure working fluids (such as carbon dioxide) because of the large potential for expansion work recovery. The use of lower pressure refrigerants will reduce the potential for expansion work recovery. However, systems operating with lower pressure refrigerants can still benefit from the use of an ejector due to better liquid distribution and lower mass flow rate (resulting in lower pressure drop) in the evaporator. Early experimental results with an ejector system using R134a by Harrell and Kornhauser (1995) showed an ejector system could achieve COP improvements of 3.9 to 7.6% over the conventional cycle. More recently, Disawas and Wongwises (2004) experimentally investigated a two-phase ejector cycle with R134a and noticed improvements of similar magnitude.

Analysis of the standard ejector refrigeration cycle generally assumes a perfect liquid-vapor separator; it is generally assumed that all of the vapor that enters the separator exits at the vapor port and that all of the liquid that enters the separator exits at the liquid port. However, depending on the type and size of the separator, this may not necessarily be the case. It is possible that the effect of separator inefficiency, that is, some amount of vapor exiting the separator at the liquid port or some amount of liquid exiting the separator at the vapor port, can significantly degrade the performance of the ejector cycle such that it is no longer beneficial to use a two-phase ejector. Because of the potential harmful effects of the liquid-vapor separator, it may be useful to consider two-phase ejector cycles that operate without the use of a liquid-vapor separator; cycles of this sort have been previously proposed by Burk et al. (2006) and Hiroshi et al. (2010).

One of the objectives of this paper is to investigate the effect that inefficiency of the liquid-vapor separator in the two-phase ejector cycle can have on the performance of the cycle. An analytical model of the two-phase ejector cycle with varying liquid and vapor separation efficiencies will be used to calculate the performance of the cycle, and the performance of the ejector cycle will then be compared to that of a conventional vapor compression cycle at the same operating conditions. R134a and R1234yf have been chosen as the working fluids for the numerical study. An ejector cycle without the liquid-vapor separator will also be investigated experimentally with R134a.

2. ANALYSIS

2.1 Standard Ejector Cycle Model
A thermodynamic simulation model was developed to analyze the performance of the standard ejector cycle with a liquid-vapor separator. The LMTD in the heat exchangers, pressure loss in the tubing and heat exchangers, and heat transfer in the tubing were not taken into account since the model would only be used for comparison with a conventional system. The system layout of the standard two-phase ejector cycle is given in Figure 1 (a). The corresponding P-h diagram for the cycle can be seen in Figure 1 (b).

![Figure 1](image_url)

**Figure 1:** (a) System layout of standard two-phase ejector cycle with liquid-vapor separator, (b) Pressure-specific enthalpy diagram of standard ejector cycle with liquid vapor separator
The ejector was modeled using the thermodynamic model developed by Kornhauser (1990). This is an iterative model with equations based on the conservation of mass, momentum, and energy, and it assumes homogeneous equilibrium at all points in the ejector. This model required the assumption of isentropic efficiencies of three ejector components: The motive nozzle (\(mn\)), the suction nozzle (\(sn\)), and the diffuser (\(diff\)). The definitions of these efficiencies are given in equations (1) through (3). Efficiencies of 0.8, 0.8, and 0.75 were assumed for the motive nozzle, suction nozzle, and diffuser respectively. The pressure in the mixing section also had to be specified as an input to the Kornhauser model. In reality, the mixing section pressure is a function of ejector geometry and flow conditions. However, in using this model, the assumption was to set the mixing section pressure to a pressure that corresponded to a 5°C drop in saturation pressure from the suction nozzle inlet pressure. In order to analyze the effect of liquid and vapor separation on ejector cycle performance, liquid and vapor separation efficiencies had to be defined. These efficiencies are given in equations (4) and (5). The liquid and vapor separation efficiencies were varied from 0.6 to 1.0 for the simulation runs. Condensation and evaporation temperatures of 45 °C and 5 °C respectively were assumed. Superheat and subcooling were both assumed to be 5 °C. The compressor was assumed to have an isentropic efficiency of 0.75. The non-linear set of equations was evaluated with EES (2011). A similar model was also developed for a conventional vapor-compression refrigeration system for comparison with the ejector system.

\[
\eta_{mn} = \frac{h_{mn, in} - h_{mn, out}}{h_{mn, in} - h_{mn, out, isen}} \\
\eta_{sn} = \frac{h_{sn, in} - h_{sn, out}}{h_{sn, in} - h_{sn, out, isen}} \\
\eta_{diff} = \frac{h_{diff, in} - h_{diff, out}}{0.5 \cdot u_{mixing}^2} \\
\eta_{liquid} = \frac{\dot{m}_{liquid-at-liquid-port}}{\dot{m}_{diff, out} (1 - x_{diff, out})} \\
\eta_{vapor} = \frac{\dot{m}_{vapor-at-vapor-port}}{\dot{m}_{diff, out} x_{diff, out}}
\]

2.2 Model Results

Figure 2 shows the results of the thermodynamic simulation for R134a and R1234yf at perfect separation and at 80 % liquid and vapor separation (meaning liquid and vapor separation efficiencies are both set to 80 % during the same test). The condensation temperature is varied from 35 to 55°C to show the effect of the ejector at different operating temperatures. The performance of the ejector system with the liquid-vapor separator is compared to the performance of the conventional system with the COP ratio as defined in equation (6).

\[
COP_{ratio} = \frac{COP_{ejector}}{COP_{conventional}}
\]

It can be seen that the benefit of the ejector system is increased (higher COP ratio) at higher condensation temperatures. At higher condensations temperatures, the saturation pressure in the condenser will be higher so the motive inlet pressure will be higher. A higher motive pressure will yield a higher motive enthalpy (with constant subcooling) and the potential for higher work recovery. Thus, at higher condensation temperatures, the benefit of using the ejector will be higher. At lower condensation temperatures, the simulation shows that it is possible to have a COP ratio of less than unity, meaning the use of an ejector would hurt system performance. This result is a
consequence of the component isentropic efficiencies being less than unity. Because the mixing section pressure is less than the evaporation pressure, a poor mixing section and diffuser may not be able to increase the pressure above the evaporation pressure. In this case, the ejector would provide a pressure decrease and harm the system performance.

Liquid separation inefficiency decreases system performance for two reasons. First, less liquid going to the evaporator will decrease the cooling capacity. Second, more liquid flowing to the compressor will increase mass flow through the compressor and the compressor work. Although increased mass flow through the compressor will also mean increased motive mass flow, the net effect of inefficient liquid separation is a decrease in COP. It is important to note, however, that liquid sent to the compressor can be utilized for additional capacity if an internal heat exchanger is used.

Vapor separation inefficiency will also decrease the system performance. Vapor going to the evaporator will not increase the cooling capacity, except for the small sensible portion, and less mass going to the compressor will decrease compressor work. However, less mass to the compressor means less motive flow in the ejector, which will result in lower work recovery. The overall result will be a decrease in COP. Although not reflected in the simplified model, additional mass flow through the evaporator because of the vapor will increase pressure drop and further decrease COP.

The results show that R1234yf benefits more from the ejector than R134a does. At perfect separation, the COP ratio for R1234yf is about 1 to 3 % higher than that of R134a over the range of condensation temperatures considered. R134a and R1234yf do have similar properties, but R1234yf yields a slightly higher difference between isentropic and isenthalpic throttling than R134a does; this is why R1234yf benefits slightly more from the use of an ejector. At 80 % liquid and vapor separation efficiency, the COP ratio of R1234yf is only slightly higher than that of R134a (less than 1 % over the range of condensation temperatures considered). Thus, although R1234yf seems to gain higher benefit from an ejector, it is also impacted more severely by imperfect separation.

Figure 2 also demonstrates that the effect of liquid-vapor separation inefficiency on ejector system performance can be significant. A 20 % decrease in liquid and vapor separation efficiency will decrease the performance of the ejector cycle by about 5 to 10 % over the range of condensation temperatures investigated. While perfect separation allows the COP ratio to remain greater than one for almost the entire temperature range, 80 % liquid and vapor separation efficiency will yield a COP ratio of less than one for the majority of the temperature range; an imperfect separator can cause the ejector cycle to only be beneficial at high condensation temperatures.

Figure 2: Theoretical performance comparison of standard ejector cycle with liquid-vapor separator to conventional vapor-compression cycle; simulation parameters are listed in Table 1; condensation temperature is varied; results are plotted for R134a and R1234yf at perfect separation and liquid and vapor separation efficiencies of 80%
Figure 3 shows the effect of varying the liquid and vapor separation efficiencies individually and together with R134a at a fixed condensation temperature of 45°C. Vapor and liquid separation inefficiency have approximately the same effect (varied individually) for the first 20% inefficiency. However, at inefficiencies greater than 20%, the liquid separation efficiency has a greater impact on the system performance. When both efficiencies are varied together, the effect is more dramatic, as would be expected. Figure 3 also shows the effect of varying the liquid and vapor separation efficiencies individually and together with R1234yf at a fixed condensation temperature of 45°C. It can be seen that R1234yf shows similar behavior as R134a. For both fluids, a liquid and vapor separation inefficiency of about 15% or greater will produce a COP ratio of less than one. Thus, at 15% liquid and vapor separation inefficiency and fixed condensation temperature of 45°C, it is no longer beneficial to use an ejector. Depending on the type and size of the liquid-vapor separator, this can actually be a realistic amount of inefficiency.

![Figure 3](image.png)

**Figure 3:** Theoretical performance comparison of R134a and R1234yf standard ejector cycle with liquid-vapor separator to conventional vapor-compression cycle at a fixed condensation temperature of 45°C; liquid separation efficiency and vapor separation efficiency varied separately and together

### 2.3 Ejector Cycles without Liquid-Vapor Separator

Because of the potential for COP decrease associated with inefficient separators in standard ejector cycles, it is worth looking into alternate ejector cycles that do not require liquid-vapor separators. As mentioned previously, several alternate ejector cycles have been proposed, and two of these cycles will be considered here.

In the first alternate cycle (Oshitani *et al.*, 2010), the subcooled liquid at the condenser outlet is split; this configuration will be referred to as the ejector cycle with condenser outlet split (COS). One stream of liquid becomes the motive flow for the ejector. This stream combines with the suction stream and enters an evaporator after the diffuser outlet. The other stream is isenthalpically expanded, goes through a lower temperature evaporator, and becomes the suction flow in the ejector. Thus, the ejector is used to pump fluid through a second (lower temperature) evaporator in this system. Because of the pressure increase across the ejector, the evaporator before the suction nozzle will have a lower saturation temperature than the evaporator after the diffuser. Thus, this cycle yields two different evaporation temperatures. Because an evaporator is placed at the outlet of the diffuser, the liquid in the two-phase fluid will be evaporated before it is sent to the compressor, and liquid-vapor separation will not be necessary. A layout of the COS cycle configuration can be seen in Figure 4 (a) and a P-h diagram of the cycle in Figure 4 (b).

The second alternate cycle (Burk *et al.*, 2006) involves splitting the two-phase fluid at the diffuser outlet; this configuration will be referred to as the ejector cycle with diffuser outlet split (DOS). One stream after the split goes straight to an evaporator, to the compressor and condenser, and becomes the motive flow in the ejector. The second stream is throttled to a slightly lower pressure, goes through a lower temperature evaporator, and becomes the suction flow in the ejector. Thus, the ejector is again used to pump fluid through a second evaporator. Because of the slight throttling before the second evaporator, this evaporator will be at a lower saturation temperature than the other. Thus, this cycle also yields two different evaporation temperatures. Similar to the COS configuration, the DOS configuration does not require liquid-vapor separation because an evaporator is placed at the diffuser outlet,
and all liquid is evaporated before entering the compressor. A layout of the COS cycle can be seen in Figure 5 (a) and a P-h diagram in Figure 5 (b).

![Figure 4: (a) Ejector Cycle with Condenser Outlet Split (COS) Layout, (b) COS Ejector Cycle P-h Diagram](image)

![Figure 5: (a) Ejector Cycle with Diffuser Outlet Split (DOS) Layout, (b) DOS Ejector Cycle P-h Diagram](image)

### 3. EXPERIMENTAL FACILITY AND METHODS

The test facility was modified from an existing system used for previous ejector experiments in order to construct a COS ejector system. A full description of the previous ejector system is given by Elbel (2007). Two closed-loop wind tunnels house the microchannel condenser and the two microchannel evaporators. Variable speed blowers and electric heaters allow for control of the air flow rate and the air inlet temperature to the condenser and evaporator. A fixed displacement automotive scroll compressor suitable for use with low-pressure refrigerants connected to a variable frequency drive 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. Two independent energy balances (refrigerant- and air-side) were obtained for the evaporator and condenser. When both balances were available (meaning there was superheat at the outlet of both evaporators), the balances generally agreed to within 5%. A torque and speed sensor on the compressor motor shaft was used to find the compressor power. Uncertainty propagation was performed using EES (2011). The uncertainty of the instruments is given by Elbel and Hrnjak (2008). It was found that the COP had an uncertainty of ±0.03, the cooling capacity had an uncertainty of ±0.02 kW, and the ejector work recovery efficiency (defined below) had an uncertainty of ±0.002.

Data was collected with a conventional system without an ejector, and this data was used as an input to an ejector design routine. The ejector motive nozzle throat diameter was designed using an empirical flow correlation by Henry and Fauske (1971). Additional ejector dimensions were designed using the previously mentioned Kornhauser (1990) model. This model required the assumption of efficiencies of the ejector components and an assumed mixing section pressure; reasonable estimates were provided for these inputs.
Data was collected at steady-state conditions per SAE Standard J2765 OCT2008 (2008) on the COS ejector system. Air flow rate and inlet temperature (in both the condenser and evaporator) and the compressor speed were kept the same for all tests unless otherwise stated. The condenser air inlet temperature was 45°C, and the air flow rate was 400 L/s. The evaporator air inlet temperature was 25°C, and air flow rate was 300 L/s. The compressor speed was 1500 rpm. R134a was used as the working fluid. R1234yf would be the focus of future experimental investigations.

Several different motive nozzle geometries were used in the ejector system tests. Three nozzle throat diameters, 0.8 mm, 1.0 mm, and 1.3 mm, were tested. Each nozzle had a diverging angle of 2.3°. The nozzles were tested with a 5 mm diameter, 60 mm long constant area mixing section. The expansion valve, shown in Figure 5, was used to control mass flow rate through the low-temperature evaporator, and data was taken at various low-temperature evaporator mass flow rates. The flow through the low-temperature evaporator is also the suction flow of the ejector; thus, varying this flow rate will affect the performance of the ejector.

The COS ejector cycle was compared to a conventional two evaporator cycle without an ejector. This cycle was constructed by making minor modifications to the COS ejector cycle, keeping tubing lengths approximately the same between the two cycles. In order to fairly compare the two cycles, all air-side conditions were kept the same, and the refrigerant flow rates to the low-temperature evaporator and the compressor as well as inlet temperatures to the evaporators were matched between the two cycles. In some cases, this required variation of the compressor speed during the conventional cycle tests.

4. EXPERIMENTAL RESULTS AND DISCUSSION

Figure 6 shows the performance of the three ejector motive nozzle geometries described above. The pressure ratio, as defined in equation (7), is plotted as a function of the mass entrainment ratio, as defined by equation (8). High entrainment ratio and high pressure ratio are both desirable when working with ejectors. However, as shown in Figure 6, there is a trade-off between higher entrainment ratio and higher pressure ratio. Similar to the operation of a pump, an ejector cannot provide a higher pressure increase without reducing the amount of mass it entrains.

\[
\pi_s = \frac{P_{\text{diff, out}}}{P_{sn, in}} \quad (7)
\]

\[
\Phi_m = \frac{\dot{m}_{sn}}{\dot{m}_{mn}} \quad (8)
\]

It can be seen from Figure 6 that the motive nozzles with the 1.0 mm throat diameter and the 1.3 mm throat diameter provided similar combinations of pressure ratio and mass entrainment ratio. The motive nozzle with the 0.8 mm throat diameter had noticeably poorer performance. If the throat diameter is too small, as may be the case with the 0.8 mm throat diameter, the excessive throttling and choking of the flow will cause too much flow loss and limit the performance of the ejector. It should be noted that if the throat diameter is too large, the flow will not expand to a low enough pressure and high enough velocity to entrain a sufficient amount of suction flow, which would also limit the performance of the ejector.

A 3 mm diameter mixing section was also tested with the same motive nozzles. However, ejector performance with the 3 mm diameter mixing section was significantly lower than with the 5 mm diameter mixing section. If the mixing diameter is too small, there will not be enough area for both the motive and suction flows to expand and properly mix, and the suction flow will be reduced, as was observed experimentally with the smaller mixing section. A 4.0° motive nozzle diverging angle was tested; however, this also resulted in reduced performance compared to the 2.3° diverging angle. For a fixed nozzle outlet diameter, a larger diverging angle will result in less diverging length. If the diverging section is too short, the flow will not expand to a high enough velocity and low enough pressure, reducing ejector performance. It should be noted that if the diverging section is too long, the negative effect of increased frictional losses will outweigh the positive effect of further expansion.
COS ejector cycle performance data is presented in Figure 7 for the ejector with the 1.1 mm throat motive nozzle and the 5 mm mixing section. The results show that COP and cooling capacity are both maximized at an intermediate value of entrainment ratio. This is due to the operation of the ejector. It is desirable to obtain several degrees of temperature difference between the two evaporators. However, the temperature difference in the evaporators is determined by the pressure increase that the ejector can provide. For a larger temperature and pressure difference, the mass flow rate through the low-temperature evaporator will be reduced, resulting in underutilization of the low-temperature evaporator. For a larger low-temperature evaporator mass flow rate, the temperature difference between the evaporators will be reduced, again resulting in less effective heat transfer. Thus, there is an intermediate entrainment ratio that will optimize the heat transfer in the evaporators. This is different from a conventional two evaporator refrigeration cycle, which is not necessarily penalized with reduced low-temperature evaporator mass flow rate at a larger evaporator temperature difference.

Figure 6: Pressure ratio versus entrainment ratio for different ejector motive nozzle diameters with a 2.3° diverging angle and a 5 mm diameter, 60 mm long mixing section

Figure 7 also shows the ejector work recovery efficiency as defined by Elbel and Hrnjak (2008) in equation (9). The efficiency reaches a maximum at an intermediate value of entrainment ratio. This is because the efficiency takes into account the pressure increase provided by the ejector in addition to the amount of mass entrained by the ejector. At high entrainment ratio, the pressure increase is too small, resulting in lower ejector efficiency. At low entrainment ratio, the pressure increase is large, but the entrained mass flow is still too low, which also results in
lower ejector efficiency. Thus, there is a peak in ejector efficiency between these two extreme cases. It is important to note that the point of maximum ejector performance does not always correspond to optimum system performance because the ejector is just one of multiple components of the system.

\[ \eta_{\text{ejec}} = \frac{\dot{W}_{\text{rec}}}{\dot{W}_{\text{rec, max}}} \]  

(9)

Table 1 shows the comparison of COS ejector and conventional two evaporator cycles for several data points. In running the conventional cycle tests, the low evaporation temperature was about 2°C lower than that of the COS ejector cycle tests because the pressure drop in the high-temperature evaporator was too large for the two temperatures to be matched. The high evaporation temperature and both the flow rates were matched well between tests on the two systems. The COP has been calculated based on the air-side energy balance because several tests on both cycles resulted in a two-phase evaporator outlet. It can be seen that the COS ejector cycle yields a higher COP than the conventional cycle at all three sets of test conditions. The ejector cycle provides a slightly higher cooling capacity than the conventional cycle due to the expansion process in the ejector cycle being closer to isentropic than in the conventional cycle. The conventional cycle also requires increased compressor work because the flow through the high-temperature evaporator is throttled down to the pressure of the low-temperature evaporator. Conversely, in the COS ejector cycle, the pressure of the low-temperature evaporator flow is increased to that of the high-temperature evaporator by the ejector. This results in a higher compressor inlet pressure and a reduction in compressor work for the COS ejector cycle. The improvement in COP of the COS ejector cycle over the conventional cycle reaches a maximum of 10 % for the second data point. This value of COP improvement is actually higher than it should be because the low evaporation temperature of the conventional cycle is lower than that of the COS ejector cycle. This lowers the compressor inlet pressure and increases compressor work more than necessary in the conventional cycle. However, the results still show that ejectors can prove beneficial to systems even with low pressure refrigerants.

<table>
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<td>3.1</td>
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<td>2.38</td>
<td>2.74</td>
</tr>
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5. CONCLUSIONS

The results of a thermodynamic simulation model used to analyze the effect of separation efficiency on the performance of the standard ejector system have been presented. It was shown that both liquid and vapor separation inefficiency have a negative impact on the system. The COP ratio of R1234yf was 1 to 3 % higher than that of R134a over the condensation temperature range of 35 to 55°C, meaning that R1234yf gains slightly more benefit from the ejector. However, R1234yf was also found to be more severely affected by separator inefficiency. It was also found that at 15 % or greater liquid and vapor separation inefficiency, it is no longer beneficial to use an ejector with R134a or R1234yf.

Ejector cycles that do not require a liquid-vapor separator have been analyzed. A COS ejector cycle test facility has been constructed with a fixed displacement automotive compressor. Data from tests on the conventional system has
been used to design an ejector suitable for use with low-pressure fluids. The testing of different motive nozzle geometries has shown that larger throat diameters (1.0 mm to 1.3 mm) and smaller diverging angles (2.3°) yield better ejector performance. Larger mixing section diameters increase ejector performance as well. It was also observed that the COS ejector cycle can achieve COP improvements upwards of 10 % over a conventional cycle under operating conditions that slightly favor the ejector cycle.

NOMENCLATURE

Abbreviations: COP coefficient of performance COS ejector cycle with condenser outlet split DOS ejector cycle with diffuser outlet split LMTD logarithmic-mean temperature difference Subscripts: cond condensation cp compressor diff diffuser evap evaporation high high-temperature evaporator

Symbols: \( D \) motive nozzle throat diameter (mm) \( h \) specific enthalpy (kJ/kg) \( m \) mass flow rate (g/s) \( T \) temperature (ºC) \( W \) work (kW) \( x \) quality (-) \( \alpha \) motive nozzle diverging angle (*) \( \eta \) efficiency (-) \( \Pi_s \) suction pressure ratio (-) \( \Phi_m \) mass entrainment ratio (-)

Greek: \( \alpha \) motive nozzle diverging angle (*) \( \eta \) efficiency (-) \( \Pi_s \) suction pressure ratio (-) \( \Phi_m \) mass entrainment ratio (-)

REFERENCES

Elbel, S.W., 2007, Experimental and analytical investigation of a two-phase ejector used for expansion work recovery in a transcritical R744 air-conditioning system, Ph.D. Thesis, Urbana, IL.

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