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Investigation to Improve Efficiency of Transcritical R744 Two-Stage Vapor Compression Systems

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ABSTRACT

The objective of this paper is to discuss an experimental investigation of a two-stage compression and expansion system with the intent of showing system performance improvement over a two-stage compression and single expansion system for an air-conditioning application. For the mechanical subcooling system, the low, moderate, and high air-temperature conditions showed a 6 to 8% increase in COP over the baseline system. Additionally, the optimization of the high-side pressure is more critical because variations from the optimum pressure have a larger effect on the COP than adjusting the bypass mass flow rate. Further analysis into the bypass valve operation for the mechanical subcooling system was done using a correlation for a short tube with a constant area.

1. INTRODUCTION

The interest in R744 as a refrigerant has increased as there has been a push toward environmentally friendly refrigerants. Since R744 has an ozone depleting potential of zero and its global warming potential is negligible if it is recovered from industrial processes, it is seen as one possible replacement for traditional refrigerants such as R134a.(Baek et al., 2005; Cecchinota, et al., 2005)

Because of its critical pressure, 7.38MPa, and relatively low critical temperature, 31.06°C, the cycle operation is different from traditional vapor compression cycles, which can cause the refrigerant at the high-side pressure to be supercritical.(Cecchinota, et al., 2005) For supercritical operating conditions, the heat transfer in the gas cooler does not involve a phase change but only gas cooling, thus the condenser is replaced by a gas cooler. (Cecchinota, et al., 2005) Additionally, because of the supercritical nature of the cycle, the throttling process behaves differently and the effect of the throttling process is stronger because of increased loss of exergy when the temperature of the gas cooler is above the critical temperature. (Cavallini et al., 2005; Cecchinota, et al., 2005) This leads to a relatively high gas cooler refrigerant exit temperature and thus high vapor quality at the evaporator entrance, (Cavallini et al., 2005) especially in a system without a suction line heat exchanger.

One method to improve R744 system performance is to use two-stage compression, and two-stage expansion. The object of this investigation and paper is to experimentally show performance improvement of double expansion over single expansion in a two-stage compression system for an air-conditioning application. In addition, the feasibility of using a fixed orifice for the bypass valve was studied and the results are presented.

2. SYSTEM DESCRIPTIONS

2.1 Baseline System

The transcritical vapor compression R744 systems presented in this paper are denoted as the baseline system and the mechanical subcooling system. The baseline system consists of the following components: a two-stage hermetic compressor with microchannel intercooler, a microchannel gas cooler, a tube-in-fin evaporator, slab suction line heat exchanger (SLHX) and an electronic expansion valve. The ideal cycle is shown in the p-h diagram in Figure 1. The numbers correspond to the following ideal processes.
The refrigerant undergoes an isentropic compression (1-2) to an intermediate pressure, isobaric heat extraction in the intercooler (2-3), isentropic compression to the high-side pressure (3-4), isobaric heat extraction in the gas cooler (4-5), and isobaric heat exchange with the low pressure refrigerant in the SLHX (5-6). The refrigerant then is subject to an isenthalpic expansion in the expansion valve (6-7) and isobaric heat addition in the evaporator (7-8). After the evaporator, the refrigerant is superheated isobarically in the low-side of the SLHX before returning to the compressor (8-1).

2.2 Mechanical Subcooling System
The mechanical subcooling system includes the components of the baseline system with the addition of a slab internal heat exchanger (IHX) and an additional electronic expansion valve. The ideal mechanical subcooling cycle is shown in the p-h diagram in Figure 1b with the numbers corresponding to the following ideal processes.

Similar to the baseline ideal cycle, the refrigerant undergoes an isentropic compression (1-2) to an intermediate pressure, isobaric heat extraction in the intercooler (2-3), isentropic compression to the high-side pressure (3-4), and isobaric heat extraction in the gas cooler (4-5). After the gas cooler, part of the refrigerant flow is separated from the total flow, and expanded isenthalpically to the intermediate pressure (5-10). The total flow rate and the separated flow rate are denoted as \( m_r \) and \( m_b \), respectively in Figure 1b. The separated refrigerant is used to isobarically extract heat (10-3) from the remaining refrigerant exiting the gas cooler (5-6) in the IHX. This flow is recombined with the main flow at the inlet to second stage of the compressor, where it is ideally mixed with the refrigerant exiting the intercooler (point 3). After the isobaric heat exchange at the high-side IHX, the main refrigerant flow isobarically exchanges heat in the SLHX (6-7) with the low pressure side of the SLHX. It is then isenthalpically expanded in the main expansion valve (7-8) and undergoes isobaric heat addition in the evaporator (8-9). After the evaporator, the refrigerant is superheated isobarically in the low-side of the SLHX before returning to the compressor (9-1).

![Figure 1 Baseline (a) and mechanical subcooling (b) ideal cycle p-h diagrams](image)

3. EXPERIMENTAL FACILITY
The experimental facility used for all experiments consists of two environmental chambers. The outdoor chamber contained the gas cooler, intercooler, and compressor, while the evaporator was mounted in the indoor chamber. There are several welded Type-T thermocouples attached to each chamber surface. The heat transmission losses through the chamber surfaces during an experiment were determined from temperature differences between the interior and the exterior of the chambers in combination with the UA values determined from chamber calibration experiments. During a test, the ambient conditions inside the chambers were maintained at specified values. The power consumption of all electrical devices operated in the chambers was measured with Watt transducers. By combining the reading of the Watt transducer, the heat loss through the walls, ceiling, floors and the latent portion of the evaporator capacity, it was possible to determine the capacity on the evaporator. The capacity on the evaporator
determined by this method is called the chamber energy balance. The chamber energy balance on the gas cooler/intercooler could not be determined because the amount of heat added to the chamber from the compressor was not known.

All refrigerant–air heat exchangers were installed in open-loop wind tunnels housed inside the chambers. Before and after each heat exchanger, four by four thermocouple grids consisting of welded Type-T thermocouples were used to determine the dry-bulb air temperatures. Downstream of the air outlet temperature grids, flow nozzles were installed to determine the airflow rates in both ducts by measuring the pressure drops with differential pressure transducers.

For the air side energy balance, the latent load was calculated from two chilled mirror dew point sensors. Radial blowers were connected at the outlet of each duct in order to force air through the wind tunnels. The airflow rates through the ducts were controlled with variable frequency drives. The indoor and outdoor blowers exhausted the air directly into the chambers. An external R404A chiller system with evaporators mounted on the ceiling inside the outdoor chamber compensated for the heat rejected by the gas coolers intercooler and compressor. PID-controlled electric heaters were installed in both chambers to reheat the room to the specified test conditions. In order to maintain the moisture content in the indoor room at a specified condition, steam injection was used to create the latent load at the evaporator.

The refrigerant-side energy balance was accomplished from temperature and pressure measurements in combination with two coriolis-type mass flow meters in series. Absolute pressures were measured at the outlet of every component. In addition, differential pressure measurements were taken across the heat exchangers to determine the absolute pressures at the heat exchanger inlets. All refrigerant-side temperatures were measured with ungrounded Type-T immersion thermocouple probes.

Shake down tests were performed before running baseline tests. It was found that the refrigerant, chamber and air capacity measurements matched within 10%, most within 5%.

This investigation focused on three conditions based on the air temperatures at the gas cooler and evaporator entrances: 27.8 °C gas cooler, 26.5 °C evaporator, denoted as the low condition; 35 °C gas cooler, 27 °C evaporator, denoted as the moderate condition and 42 °C gas cooler, 32 °C evaporator, denoted as the high condition. All three conditions did not have dehumidification.

4. COP IMPROVEMENT

The experimental results from the mechanical subcooling system are shown in the following contour plots. All results are compared at the cooling capacity and the maximum COP of the baseline system. For each condition, it can be seen that the COP depends on both the high-side pressure and the bypass mass flow rate. The contours show how the COP varies from the maximum COP for each operation condition. Each contour plot is based on a minimum of twenty data points.

At the maximum COP for the low condition the mechanical subcooling system had an 8% increase over the baseline system COP and a bypass mass flow rate of 1%. This 1% bypass mass flow rate contributes to a refrigerant temperature difference in the high-side of the IHX and point (7) in Figure 1b moves to the left, which then translates into higher COP by looking at the entire system. The darkest area in Figure 2, where the maximum COP occurs, shows that the optimum bypass flow rate is between 0% and 15% of the total mass flow rate.
For the moderate condition, at the maximum COP, there is a 6% increase in COP over the baseline system and has a bypass mass flow rate of 21% of the total mass flow rate. In Figure 3, the second highest contour is at 98% of the maximum and has a bypass mass flow rate of 15% to 30%.

For the high condition the maximum COP was 8% higher than the baseline system and had a bypass mass flow rate of 22%. The highest contour at approximately the maximum COP has a bypass mass flow rate of 15% to 24% and is shown in Figure 4.
The contour plots show that the system is more sensitive to optimizing the high-side pressure than the bypass mass flow rate to achieve optimum COP. This is seen by the narrow width along the pressure axis and wider along the bypass mass flow rate axis of the maximum COP section in the plots. The low condition has the steepest decent from the maximum COP dropping at 5% per contour. The moderate condition is less steep dropping approximately 2% per contour and the high-condition has the shallowest decent, only dropping 1% per contour from the maximum. This shows that as the ambient air temperatures increases, the sensitivity of the optimum high-side pressure and bypass mass flow rate to find the maximum COP decreases. From these plots, it can also be seen that the optimization of the high-side pressure is more critical because variations from the optimum pressure have a larger effect on the COP than adjusting the bypass mass flow rate.

5. BYPASS VALVE OPERATION

This sectionexamsmore closely the operation of the bypass valve in the mechanical subcooling system with SLHX. The manufacturer’s manual for the stepper motor electric expansion valve that was used as the bypass valve in the experiments gave an equation relating the voltage fed to the expansion valve and the percent opening of the expansion valve. This equation was used to determine the valve opening for representative bypass mass flow rate data for each condition. The results were plotted and shown in Figure 5. The plot shows that each condition follows a different linear trend for the bypass mass flow rate percent. The points of maximum COP for each condition are also marked on the figure and do not correspond to the same point for all the conditions. The low condition is significantly different since the maximum is at 1% bypass. As seen by the contour curves in Figure 2 the bypass can vary from 0% to 16% at the optimum high-side pressure. At the high end of this range, at 15%, the opening of the bypass valve is the not at the maximum COP point for the moderate condition, though appears to have the same opening as the high conditions. The high and moderate conditions have about the same optimum bypass mass flow rate percent; however the valve opening is different and the nominal mass flow rate is different.
One option for the bypass valve is to use a fixed orifice instead of an electronic expansion device. This would decrease the amount of controls and extra power needed to operate the system, and also decrease the cost of the system components. The issue with a fixed orifice is that it is a fixed diameter and will not be able to be optimized for all conditions. The deviations from the optimum and maximum COP were investigated using a correlation for a short tube orifice and the results are presented below.

The single-phase orifice equation to predict the mass flow rate through a short tube was corrected for two-phase flow and R744 to be (Liu et al., 2004):

\[
m = A_s \sqrt{2\rho (P_{\text{high}} - P_f)}
\]  

where \( A_s \) is the cross sectional area in \( m^2 \), \( \rho \) is the density of the fluid entering the orifice in \( kg/m^3 \), and \( P_f \) is the adjusted downstream pressure in MPa. In this case, \( P_{\text{high}} \) is the pressure at the exit of the gas cooler, \( P_{\text{cro}} \) and \( P_f \) is given by the correlation (Liu, et al., 2004):

\[
P_f = P_c \left[ 0.77213 + 0.086505 \cdot D_{R}^{0.2405} - 0.40453 \cdot P_{R}^{-4.7025} \cdot T_{R}^{-0.12293} + 0.00006 \cdot (L/D)^{1.6326} \right]
\]

where \( D_{R} = D/0.03 \), \( P_{R} = P_{\text{high}}/P_c \), and \( T_{R} = (T_{\text{high}} - T_c)/T_c \). \( P_c \) and \( T_c \) are the critical pressure and temperature of R744, 7.38MPa and 31.03°C respectively. \( D \) and \( L \) are the dimensions of the short tube used to develop the correlation. In this case, \( T_{\text{high}} \) is the refrigerant temperature at the exit of the gas cooler, \( T_{\text{cro}} \).

Equations (1) and (2) were developed to predict the flow of R744 through a short tube with the following constraints. The high-side pressure ranged between 7.55 to 10.24 MPa, the low-side pressure ranged between 3.03 to 5.02 MPa and the high-side temperature ranging between 35.3 to 45.5°C. While the low-side pressure in the bypass valve in this investigation is outside this range, between 5.4 and 8.4 MPa, using this correlation can give an idea as to whether a fixed orifice will work for the bypass valve. The mass flow rates in the study were also significantly higher. The study experimented with many different lengths and diameters of the short tubes and it was shown that the effect of the length was minimal while the effect of the diameter strongly influences the mass flow rate(Liu et al., 2004). Therefore, in the following illustration the length was set as one of the lengths used in the study leaving only the diameter as an unknown.
Using the Tcro and Pcro from the maximum COP point for the moderate condition and the corresponding mass flow rate, the diameter needed was calculated to be 0.3664 mm. Using the Tcro and Pcro from the maximum COP point for the high condition and the corresponding mass flow rate, the diameter needed was calculated to be 0.3352 mm. The Tcro for the low condition at maximum COP was below the critical temperature at 30 °C ambient temperature; therefore it cannot be used in equation (2). For illustration purposes, the temperature was raised by 2 degrees to 32 °C. Using the Pcro and adjusted Tcro from the maximum COP point for the low condition and the corresponding mass flow rate, the diameter needed was calculated to be 0.2389 mm.

In Table 1, the results for each of the found optimized diameters is summarized. The table shows a comparison of the predicted bypass mass flow rate which was calculated from the equation (1). The actual mass flow rate value was the measured flow rate and is necessary to satisfy the balance between the flow rates (total and bypass) for maximum COP. For each condition, an optimal diameter was found using the actual bypass mass flow rate. Using this diameter, a mass flow rate was found for the other two conditions, denoted as predicted mass flow rate in the table, and compared to the actual bypass mass flow rate. The actual and the predicted mass flow rates differ whenever the diameter used in the calculations is not the optimized diameter for that condition.

<table>
<thead>
<tr>
<th>Condition diameter is optimized for</th>
<th>Condition</th>
<th>Diameter [mm]</th>
<th>Predicted mass flow rate [g/s]</th>
<th>Actual mass flow rate [g/s]</th>
<th>Predicted bypass mass flow rate percent</th>
<th>Actual bypass mass flow rate percent</th>
<th>COP difference from maximum based on change in percent of bypass mass flow rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low</td>
<td>Low</td>
<td>0.2389</td>
<td>2.0</td>
<td>2.0</td>
<td>9%</td>
<td>9%</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Moderate</td>
<td>0.2389</td>
<td>2.3</td>
<td>5.3</td>
<td>9%</td>
<td>21%</td>
<td>-5%</td>
</tr>
<tr>
<td></td>
<td>High</td>
<td>0.2389</td>
<td>2.9</td>
<td>5.8</td>
<td>11%</td>
<td>22%</td>
<td>-2%</td>
</tr>
<tr>
<td>Moderate</td>
<td>Low</td>
<td>0.3664</td>
<td>4.6</td>
<td>2.0</td>
<td>20%</td>
<td>9%</td>
<td>-5%</td>
</tr>
<tr>
<td></td>
<td>Moderate</td>
<td>0.3664</td>
<td>5.3</td>
<td>5.3</td>
<td>21%</td>
<td>21%</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>High</td>
<td>0.3664</td>
<td>6.8</td>
<td>5.8</td>
<td>25%</td>
<td>22%</td>
<td>-1%</td>
</tr>
<tr>
<td>High</td>
<td>Low</td>
<td>0.3352</td>
<td>3.9</td>
<td>2.0</td>
<td>10%</td>
<td>9%</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Moderate</td>
<td>0.3352</td>
<td>4.5</td>
<td>5.3</td>
<td>17%</td>
<td>21%</td>
<td>-2%</td>
</tr>
<tr>
<td></td>
<td>High</td>
<td>0.3352</td>
<td>5.8</td>
<td>5.8</td>
<td>22%</td>
<td>22%</td>
<td>-</td>
</tr>
</tbody>
</table>

The low condition diameter predicts a mass flow rate of 2.9 g/s for the high condition with Tcro and Pcro from the maximum COP point for the high condition. This corresponds to 11% bypass mass flow rate and would drop the COP by 2% for the same high-side pressure. The moderate condition diameter predicts a mass flow rate of 6.8 g/s for the high condition. This corresponds to about 25 % bypass mass flow rate percent and would drop the COP by 1% for the same high-side pressure.

The diameter from the low condition predicts a mass flow rate of 2.3 g/s for the moderate condition with Tcro and Pcro from maximum COP point for the moderate condition. This corresponds to about 9% bypass mass flow rate and drops the COP by 5% for the same high-side pressure. The diameter from the high condition predicts a mass flow rate of 4.5 g/s. This corresponds to about 17% bypass mass flow rate and would drop the COP by 2% for the same high-side pressure.

Using the optimum diameter found for the moderate condition, and the Pcro and modified Tcro from the low condition, the predicted mass flow rate is 4.6 g/s. This corresponds to about 20% bypass mass flow rate and would drop the COP by 5% for the same high-side pressure. Using the optimum diameter found for the high condition the predicted mass flow rate is 3.9 g/s. This corresponds to about 10% bypass mass flow rate and is within the maximum COP area in Figure 2 for the same high-side pressure.

The results show that the optimized diameter for maximum COP for one condition cannot predict the mass flow rate for maximum COP for the other conditions. This begs the question: for which condition should the system be optimized? One option is to optimize for the high condition because it has the smallest variations of the investigated operation conditions. This is based purely on a numbers standpoint. From an engineering standpoint, the actual usage of a system must be taken into account. Therefore, the system should be optimized for the most common condition to occur. Additionally, variations from the optimum may be acceptable in the final design and must also be considered.
6. CONCLUSIONS

The following conclusions are made based on the above analysis. The experiments showed 6%-8% increase in COP. For the mechanical subcooling system, the COP depends on high-side pressure and bypass mass flow rate. Also, as the ambient air temperatures increases, the sensitivity of the optimum high-side pressure and bypass mass flow rate to find the maximum COP decreases. The high-side pressure is more sensitive and can be optimized using a COP maximizing equation based on the gas cooler air exit temperature. The bypass electronic expansion valve may be replaced by a fixed orifice but can only be optimized for one condition. For the conditions and data presented, the maximum loss in COP is 5%.

NOMENCLATURE

| AFR | Air Flow Rate (m³/hr) |
| As | Cross sectional Area (m²) |
| COP | Coefficient of Performance (-) |
| D | Diameter (mm) |
| h | Enthalpy (kJ/kg) |
| IHX | Internal Heat Exchanger (-) |
| L | Length (mm) |
| \( \dot{m} \), \( m \) | mass flow rate (g/s) |
| p | Pressure (MPa) |
| Pcro | Gas Cooler Refrigerant Exit Pressure (MPa) |
| \( \rho \) | Density (kg/m³) |
| SLHX | Suction Line Heat Exchanger (-) |
| Tcro | Gas Cooler Refrigerant Exit Temperature (°C) |

Subscripts

- high
- high-side
- R
- reduced
- r
- total
- b
- separated

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