SEPARATION OF LIQUID-VAPOR TWO-PHASE FLOW IN HEADERS OF MICROCHANNEL CONDENSERS

BY

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THESIS

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Abstract

This work presents an experimental and numerical study of separation of liquid and vapor as a way to improve condenser efficiency and heat transfer performance of typically microchannel design. This thesis is composed of three parts in the following order. 1. Separation of vapor and liquid in condensers is evaluated via numerical study as a way to improve efficiency. 2. Effects of separation of vapor and liquid on condenser performance are experimentally investigated on a MAC system. 3. Experimental study of separation of vapor and liquid in a vertical header of MCHX with flow visualization is conducted to study the separation mechanisms.

The first part of the chapters evaluates the concept of separation of vapor and liquid in condenser as a way to improve efficiency. An experimentally validated microchannel condenser model indicates that separation of vapor and liquid in condenser is beneficial for performance, by either reducing the refrigerant exit temperature (enthalpy) or increasing the condensation mass flow rate at the same air side conditions. The magnitude is function of separation efficiency and optimization of the circuiting. When the sum of tube numbers of liquid and vapor pass strictly follow each pass in the baseline, the condenser refrigerant outlet temperature of the separation condenser is lower than the baseline by 0.7°C at the same refrigerant inlet state. In addition, condenser pass circuiting with different pre-assumed separation results in the header is investigated by the model.

In the second part, effects of separation of vapor and liquid on condenser performance are experimentally investigated by implementing the separation condenser into an R134a MAC system. In the heat exchanger-level test, compared to the baseline condenser with the identical geometry and operating at the same condition, the separation condenser generates round 7.4% more condensate. In the system-level test, an experimental comparison at matched capacity revealed that separation condenser provided a maximum COP improvement of 6.6%. The benefit is identified and discussed: increased refrigerant-side heat transfer coefficient induced by separation of two phases. Separation efficiency in the real application is investigated and potential of further improvement is shown if separation efficiency could be increased.

The last part of this work presents the experimental study of separation of two-phase flow in a vertical header of MCHX based on quantified visualization using fast camera. A header prototype is made that has an inlet in the longitudinal center part. Two sub-passes downstream are
designed, lower for liquid and upper vapor flow. The header for experiment is clear to provide visual access. R-134a is used as the fluid of interest and mass flux through the inlet microchannels is controlled between 55 kg/(m²s)-195 kg/(m²s). The experiment results indicate that ideal separation in that header can happen at low mass flux up to 70 kg/(m²s). Results are presented in function of liquid and vapor separation efficiencies ($\eta_l$, $\eta_v$). Flow patterns inside the header are identified and analyzed to study the mechanisms for liquid-vapor separation. The efficiency deteriorates dramatically when the recirculation region elevates up to the top of the header, with increasing inlet flow rate and/or quality. Potential design options to improve two-phase separation are discussed. The objective should be to avoid or at least delay the recirculation region from reaching the vapor exit by reducing the liquid upward momentum or the vapor upward velocity, and decreasing liquid and vapor force interaction.
To my parents
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NOMENCLATURE

Symbols

\( A \)    heat transfer area    \( (m^2) \)
\( A_c \)  cross sectional area  \( (m^2) \)
\( D \)    tube diameter         \( (mm) \)
\( D_h \)  microchannel size     \( (mm) \)
\( D_{hd} \) header diameter    \( (mm) \)
\( \Delta h_{fg} \) latent heat of vaporization \( (kJ \cdot kg^{-1}) \)
\( \Delta P \) pressure drop     \( (kPa) \)
\( F \)    frame rate            \( (\text{frame} \cdot \text{s}^{-1}) \)
\( F_c \)  compressor torque    \( (N \cdot m) \)
\( G \)    mass flux             \( (kg \cdot m^2 \cdot s^{-1}) \)
\( h \)    heat transfer coefficient \( (W \cdot m^2 \cdot K^{-1}) \)
\( hA \)   heat conductance      \( (W \cdot K^{-1}) \)
\( k \)    pressure loss coefficient \( (-) \)
\( L_t \)  total tube length     \( (m) \)
\( \dot{m} \) mass flow rate     \( (g \cdot s^{-1}) \)
\( P \)    pressure              \( (kPa) \)
\( q \)    heat flux             \( (kW \cdot m^2) \)
\( Q \)    cooling capacity      \( (kW) \)
\( t \)    Time                  \( (s) \)
\( T \)    temperature          \( (^oC) \)
\( U \) velocity \( (m \cdot s^{-1}) \)

\( V_C \) compressor speed \( (rpm) \)

\( W_C \) compressor power \( (kW) \)

\( WB \) wet-bulb temperature \( (^{\circ}C) \)

\( x \) refrigerant quality \( (-) \)

**Abbreviations**

\( COP \) coefficient of performance \( (-) \)

\( DB \) dry-bulb temperature \( (^{\circ}C) \)

\( EEV \) electric expansion valve

\( LMTD \) logarithm mean temperature difference \( (^{\circ}C) \)

\( MCHX \) microchannel heat exchanger

**Greek**

\( \eta \) separation efficiency \( (-) \)

\( \theta \) inclination angle of inlet tube of T-junction \( (^{\circ}) \)

\( \mu \) dynamic viscosity \( (Pa \cdot s) \)

\( \rho \) Refrigerant density \( (kg \cdot m^{-3}) \)

**Subscripts**

\( air/a \) indoor air side

\( cri \) condenser refrigerant inlet

\( cro \) condenser refrigerant outlet

\( ero \) evaporator exit

\( exp \) flow expansion

\( HX \) heat exchanger
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tr>
<td>in/i</td>
<td>Inlet</td>
</tr>
<tr>
<td>liq/l</td>
<td>liquid phase</td>
</tr>
<tr>
<td>out/o</td>
<td>Outlet</td>
</tr>
<tr>
<td>ref</td>
<td>refrigerant side</td>
</tr>
<tr>
<td>tot</td>
<td>total</td>
</tr>
<tr>
<td>vap/v</td>
<td>vapor phase</td>
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<td>xri</td>
<td>expansion valve refrigerant inlet</td>
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Chapter 1  INTRODUCTION

1.1 BACKGROUND

Microchannel heat exchangers (MCHXs) are widely used in Heating, Ventilation, Air-Conditioning & Refrigeration (HVAC&R) industry due to their high overall heat transfer coefficient, lower refrigerant inventory, compactness and lower weight. So they are gaining attention from numerous scholars and researchers.

Air side was known for having the major heat resistance \( (1/hA) \) in heat transfer for a MCHX. However, with fin design being renovated and the resultant larger air-side area, refrigerant-side thermal resistance becomes comparable to air-side thermal resistance on the same order of magnitude. This gives a promising prospect of enhancing the refrigerant-side overall heat transfer coefficient \( (h) \) or contact area \( (A) \), thus increasing the heat transfer performance of the whole heat exchanger. For a heat transfer process, liquid on the wall of condenser is detrimental while vapor on the wall of evaporator detrimental. Removing unwanted phase is one of the options to improve heat transfer and reduce pressure drop.

For a microchannel condenser, removing unwanted phase may refer to separating liquid from vapor during the condensation process and then reassigning the flow passages for separated vapor and liquid, respectively, or draining the generated liquid out of the condenser or to further downstream passes. Either separation or drainage may be realized via flow passage arrangement / smart circuiting without too much additional cost.
The theoretical rationale behind the advantages of a condenser with vapor-liquid separation is following. For a condensation process at the same mass flux, the high-quality ($x=0.8-1$) two-phase refrigerant has 10-15 times higher heat transfer coefficient than a low-quality ($x=0.2-0.3$) two-phase refrigerant, depending on the category of refrigerant. When condenser face area and total heat transfer area are fixed, effective separation and smart circuiting can potentially improve heat transfer performance by utilizing the high heat transfer coefficient of separated vapor and giving more heat transfer area (more microchannel tubes) to this vapor flow. If the mass flux of vapor can be kept the same with non-separated refrigerant, it should have a much higher heat transfer coefficient and will enhance the overall heat transfer of the condenser.

A valid question would then be how much separation really exists in a header of a microchannel condenser. In general, MCHXs have multiple microchannel tubes in parallel connected by two manifolds. Since two-phase refrigerant flow has very complicated flow behavior inside the manifolds, it is almost inevitable to feed non-homogeneous refrigerant flow into each microchannel tube in this type of parallel flow structure. Non-uniform distribution (maldistribution) has been observed in reality and extensively studied, mostly in evaporators, because it will cause capacity loss, frost formation, or even compressor failure due to flooding of liquid refrigerant coming out of the evaporator. In a microchannel condenser, refrigerant is condensed into two phase prior to entering the second header. It is inevitable to have maldistribution as along as two-phase refrigerant exists in the header.

While maldistribution exists as a problem, it provides a possibility for creating vapor-liquid separation. Meanwhile, two-phase distribution can also be improved by first
separating the liquid and vapor then sending each to separated passes, since single-phase flow after separation has more uniform properties than a two-phase flow. Questions that also need to be addressed include how much is the effect of separation on a condenser performance and on an air-conditioning system, and what the influential parameters are on phase separation in a header of a microchannel condenser.

The scientific merit of this research is to quantify the effect of liquid-vapor phase separation on condenser and system performance. Also, this research provides both a quantitative and visual understanding of two-phase refrigerant separation in headers and studies the corresponding separation mechanisms. Potential options to improve separation in headers are proposed.

1.2 STRUCTURE OF THE THESIS

In addition to the introductory chapter, this thesis is organized in 4 chapters. Chapter 2 presents a literature review about previous work carried out in separation application in heat exchangers and separation of two-phase flow in headers. Chapter 3 presents the description of the liquid-vapor refrigerant separation concept and numerical study of the effect of separation on a microchannel condenser. Chapter 4 presents experimental investigation of the effect of liquid-vapor refrigerant separation on a microchannel condenser and its effect on the performance of an air conditioning system. In Chapter 5, liquid-vapor refrigerant separation in a vertical microchannel is investigated experimentally. Flow visualization is taken for analysis of separation phenomena of liquid-vapor refrigerant. Separation mechanisms and design criteria are revealed for enhancing the separation in the vertical header.
Chapter 2  LITERATURE REVIEW

This chapter presents a state-of-the-art review of the most relevant literature related to the topics of this thesis. The literature review includes four sections: 1) separation of two-phase flow to improve condensers; 2) separation of two-phase flow in headers; 3) summary and conclusions.

2.1 SEPARATION OF TWO-PHASE FLOW TO IMPROVE CONDENSERS

Separation of two-phase flow in condensers started to be found in open literatures in recent years. The application can mainly be categorized into two types of configuration: extraction and separation. Mostly, the condensers are parallel-flow condensers. Functioning tubes are serpertine round tubes or microchannel tubes with fins on them. Schematics of these two different geometries are in Figure 2.1 and Figure 2.2.

Figure 2.1 represents an extraction condenser. The geometry differs from those normal parallel-flow condensers because the second baffle has a number of holes on it. As two-phase refrigerant enters the header, the holes provide the possibility for drainage of the liquid phase. Consequently, the extraction of liquid can improve the average refrigerant quality of the tube passes, thus enhancing the condensation heat transfer.
Figure 2.1 Schematic for extraction geometry

Figure 2.2 represents a separation condenser. The inlet of the condenser is located at the middle of the first header. The 2nd pass is divided into an upper pass and a lower pass. As the superheated refrigerant is condensed through the 1st pass into a two-phase flow entering the second header, the two-phase flow is designed to have vapor separate from liquid because of gravity effect. In this way, the upper pass has a higher heat transfer coefficient, because of higher quality, than the 2nd pass without separation. Consequently, with more tubes in the upper pass than the lower pass the overall heat transfer of the condenser is increased.

Figure 2.2 Schematic for separation geometry
In the open literature, the real number of microchannel tubes in each pass and the real number of passes in papers may vary to achieve a best performance for the condenser.

2.1.1 EXTRACTION CONDENSERS

Wu et al. (2010) presented their study on extraction condenser based on several of their earlier published patents (Peng et al., 2003; Peng et al., 2007). After fundamentally demonstrating the heat transfer enhancement from the entrance effect of condensation, their aim is to separate liquid from gas and make condensation always occur in droplet and unsteady thin film mode everywhere in the whole condenser, as shown schematically in Figure 2.3. Figure 2.4 shows the extraction design based on an original serpentine type condenser with refrigerating capacity of 2300 W. In the two vertical manifolds several liquid-vapor separators were included to ensure that pure vapor enters the next flow passage. Experiments were conducted to validate the high performance of extraction condenser. Both capillary length and refrigerant charge were optimized to obtain the highest COP and refrigerant capacity for extraction condenser and baseline condenser. With heat transfer area being 37% less than the baseline, the extraction condenser performed as good as the baseline condenser.

Figure 2.3 Schematic for separation geometry (Wu et al., 2010)
Ye et al. (2009) tried to extract vapor phase instead of liquid phase. Two holes were punched in the center of the first two baffles in the headers to introduce vapor from upstream to downstream passes while reducing the tube numbers in the de-superheat zone. Two-phase zone in extraction condenser (multiple parallel-pass, MPP) was claimed to be theoretically larger than that in the same-size benchmark condenser (parallel flow, PF), as shown in Figure 2.5. Heat transfer was therefore enhanced while pressure drop was reduced. Also, a more uniform refrigerant quality entering the next pass could be achieved because superheated vapor introduced through the holes merged and mixed vigorously with the two-phase refrigerant from the pass upstream. Experimental data showed under the same refrigerant inlet condition and outlet subcooling, this technique was able to improve cooling performance as high as 9.6% while the refrigerant mass flow was increased by 13.34%.
Chen et al. (2012) tested the performances of an original fin-and-tube coil condenser and a condenser with liquid-vapor separation baffles, as shown in Figure 2.6, on an R-22 air conditioner using capillary tube. At fixed indoor ($DB=27$, $WB=19$) and outdoor ($DB=35$, $WB=24$) temperature, the optimal refrigerant charge amount and capillary tube length were sought to get a maximum energy efficiency ratio. The results showed the liquid-vapor separation condenser (LSC) had equivalent cooling capacity and energy efficiency ratio as the original condenser while the heat transfer area of the separation condenser was only 63.1% of the baseline and the charge amount was 80.3%. The pressure drop of the LSC was 48.6%-54.5% less than that of the baseline. The author concluded this was caused by the arrangement of the multi-pass, which gave smaller mass flow rate, shorter route path and more even refrigerant distribution.
Figure 2.6 Schematics of the baseline condenser and the LSC (Chen et al., 2012)

The benefits of reducing pressure drop were further clarified by Zhong et al. (2014a), in which they tested three different condensers as shown in Figure 2.7. The LSC had the same total area of coil tubes with a parallel-flow condenser (PFC) and a serpentine condenser (SC). The pressure of the LSC was reduced remarkably, by 77.1-81.4%, compared with that of the SC, and by 57.5-64.6%, compared with that of the PFC, at a heat flux of 6.45 kW/m². Also, the authors found the pressure drop of LSC hardly varied as the heat flux and condensing temperature changed. Other than pressure drop for comparison, they also adopted the penalty factor (PF) proposed by Cavallini et al. (2001) to evaluate the combined thermal performance during the condensation in the two condensers. The PF’s of the LSC were lower by over 50% compared to those of the SC and the PFC.
Based on the experience with fin-and-tube condensers, Zhong et al. (2014b) moved on to microchannels to experimentally compare the performance of a liquid-vapor separation microchannel condenser (LSMC) with a common parallel flow microchannel condenser (PFMC). By having a pre-cooler upstream the condenser, they were able to control the average quality (average of the inlet and outlet) of R134a as a parameter to test condenser’s sensitivity. Mass flux, heat flux, and condensing temperature were also regulated. Results showed that the average heat transfer coefficient of the LSMC exceeds that of the PFMC when mass flux is more than 590 kg/(m²s) at condensing temperature of 45°C or when the average quality is more than 0.57. The pressure drop of the LSMC is
greatly reduced by 30.5%-52.6% of the PFMC. The combined thermodynamic performance of the LSMC was proven to be superior to that of the PFMC. The maximum reduction of the PF value between the two condensers is 52.7% and 55.2% at condensing temperature of 45°C and 50°C, respectively. In addition, the minimum entropy generation number defined by Saechan and Wongwises (2008) for the LSMC is 13.9%-30.6% less than that of the PFMC at 45°C and 17.8%-27.2% less at 50°C.

2.1.2 SEPARATION CONDENSERS

The configuration for separation condenser was first proposed by Oh et al. (2003) from Halla Co., shown by Figure 2.8. The authors claim that the gas-liquid separating condenser can enhance the sub-cooling rate in section pre-sub-cooling section dm4’ as well as in the total sections. It is also claimed the invention can have suitable designs according to calculated conditional expressions of relative dimensional ratios of the sections in condensation of refrigerant to realize the optimum condensing efficiency regardless of the overall size of the gas-liquid separating condenser. The vertical section between baffle 161 and baffle 164 denotes the dm4’. It has been investigated experimentally the subcooling sensitivity on the ratio \( A_{dm4'}/A_{TOTAL} \) of the passage area of the pre-sub-cooling section dm4’ to the total heat transfer area of the condenser. Subcooling is inversely proportional to the ratio \( A_{dm4'}/A_{TOTAL} \) and \( A_{dm4'}/A_{TOTAL} \) is suitable in a range of about 3% - 20%. The condenser can obtain suitable value of heat radiation in a range of 20% to 55% which corresponds to an expression of 0.20 < \( A_{dm5}/A_{dm1} \) < 0.55, wherein \( A_{dm5} \) is the area of the second sub-cooling section dm5 and \( A_{dm1} \) is the area of the super heat cooling/condensing section dm1.
Figure 2.8 Flow of refrigerant in the multi-stage gas and liquid phase separation condenser (Oh et al., 2003)

Won (2006) theoretically optimized the tube array (tube per pass) in the separation condenser shown as in Figure 2.8. Optimization was achieved on a condenser level by keeping refrigerant inlet and mass rate and air conditions as constants, while number of tube array was varied. Empirical modeling method was adopted. Through comparison, the best tube array was 14-6-3-5-3-4. That array gave 7% higher heat release rate, much less pressure drop, and less charging amount in comparison to the baseline array, 17-5-3-3-2-5.

Although Oh et al. and Won had experimental and modeling results to verify the supreme performance of separation condenser, they have uniformly assumed a perfect separation of gas from liquid in second header tank 150 defined by baffle 163 and baffle 164. Relatively active gaseous phase moves upward owing to buoyance based upon the density difference between gas and liquid, however, it is unlikely that pure gas enters section dm2 while pure liquid enters section dm4’. Usually upward moving gas flow will
entrain liquid droplets/ligaments in it and liquid flow will mix with gas-phase bubbles. Nevertheless, it has been proven to a certain degree by both experiments and simulation that separation condensers have shown their advantages over traditional multi-pass parallel flow microchannel condensers.

2.2 SEPARATION OF TWO-PHASE FLOW IN HEADERS

2.2.1 TWO-PHASE FLOW MALDISTRIBUTION IN VERTICAL HEADERS

The flow maldistribution is defined as the inherent non-uniform distribution of vapor and liquid flow to each tube in a header-parallel tube structure. It gains interest because it will deteriorate heat exchanger performance, cause dry out in evaporators, or even result in compressor failure in flooding situation. Figure 2.9 is showing a typical dividing header which feeds hundreds of parallel microchannel ports in a microchannel heat exchanger. For industry, header shape can be cylindrical or D-shape. Two-phase flow has really complicate fluid dynamics under different inlet mass flow, quality and fluid properties, on the other hand, air side may also have varied thermal load in real applications, consequently, the driving pressure differential experienced by the all branch tubes will not be the same and may be significantly different from each other in certain cases, which in turn will cause maldistributed refrigerant mass flow rate.
Separation of vapor out of liquid is essentially an extreme type of flow maldistribution in the intermediate headers of condensers. While two-phase flow maldistribution phenomena in headers has been extensively researched during the past decades, it is necessary to review what the key parameters are in the maldistribution situation.

As in a separation condenser in Figure 2.8, the main separation mechanism is the buoyance for the vapor phase based on the density difference between liquid and vapor. Considering the geometrical characteristics should be representative of the separation condenser, only vertical header is of interest of literature review. Table 2.1 presents a summary of previous studies on distribution in vertical type manifold structures.
Table 2.1 Summary of previous studies on distribution in vertical type manifold structures

<table>
<thead>
<tr>
<th>Ref.</th>
<th>Fluid pair</th>
<th>Operating condition ($\dot{m}^\prime$: kg/m$^2$s)</th>
<th>Developing length (mm)</th>
<th>Main tube orientation / Brach flow direction</th>
<th>Main tube cross section / $D_h / l_h$ (mm)</th>
<th>Branch tube cross section / $D_b / l_b$ (mm)</th>
<th>Branch tube pitch / intrusion depth (mm)</th>
<th>Flow pattern in main tube</th>
</tr>
</thead>
<tbody>
<tr>
<td>Watanabe et al. (1995)</td>
<td>R-11</td>
<td>$\dot{m}^\prime$ = 440, 620 $x = 0$–0.3</td>
<td>/</td>
<td>Vertical upward / Horizontal</td>
<td>C / 6 /</td>
<td>C / 6 /</td>
<td>/</td>
<td>Annular, froth / slug</td>
</tr>
<tr>
<td>Yoo et al. (2002)</td>
<td>Air-water</td>
<td>$\dot{m}^\prime$ = 39–393 $x = 0.03$–0.48</td>
<td>/</td>
<td>Vertical upward and downward / Horizontal</td>
<td>SQ / /</td>
<td>SQ / 1.586 / 317.5</td>
<td>9.9 /</td>
<td>Churn, free-stream, annular</td>
</tr>
<tr>
<td>Cho and Cho (2004)</td>
<td>R-22</td>
<td>$\dot{m}^\prime$ = 60 $x = 0.1$, 0.2, 0.3</td>
<td>/</td>
<td>Vertical in-line, cross and parallel / Horizontal</td>
<td>C / 19.4 / 148</td>
<td>SQ / 1.32 / 622</td>
<td>/</td>
<td>/</td>
</tr>
<tr>
<td>Lee and Lee (2004)</td>
<td>Air-water</td>
<td>$\dot{m}^\prime$ = 54–134 $x = 0.2$–0.5</td>
<td>1650</td>
<td>Vertical upward / Horizontal</td>
<td>SQ / 24 /</td>
<td>RC / 3.33 /</td>
<td>9.8 / 0, 6, 12</td>
<td>Annular, recirculation</td>
</tr>
<tr>
<td>Lee (2009a)</td>
<td>Air-water</td>
<td>$\dot{m}^\prime$ = 70–165 $x = 0.3$–0.7</td>
<td>1650</td>
<td>Vertical upward / Horizontal</td>
<td>SQ / 14 /</td>
<td>RC / /</td>
<td>11.2 /</td>
<td>/</td>
</tr>
<tr>
<td>Lee (2009b)</td>
<td>Air-water</td>
<td>$\dot{m}^\prime$ = 70–165 $x = 0.3$–0.7</td>
<td>1650</td>
<td>Vertical upward / Horizontal</td>
<td>SQ / 8 / 600</td>
<td>RC / 1.78 / 300</td>
<td>9-49 / 0</td>
<td>Annular</td>
</tr>
<tr>
<td>Lee (2010)</td>
<td>Air-water</td>
<td>$\dot{m}^\prime$ = 70–165 $x = 0.3$–0.7</td>
<td>1650</td>
<td>Vertical upward / Horizontal</td>
<td>SQ / 14 /</td>
<td>RC / 2.8 /</td>
<td>10, 21.6 / 0, 1.75, 3.5, 7</td>
<td>Annular</td>
</tr>
<tr>
<td>Ref.</td>
<td>Fluid pair</td>
<td>Operating condition ($\dot{m}''$: kg/m²s)</td>
<td>Developing length (mm)</td>
<td>Main tube orientation / Branch flow direction</td>
<td>Main tube cross section / $D_h / l_h$ (mm)</td>
<td>Branch tube cross section / $D_b / l_b$ (mm)</td>
<td>Branch tube pitch / intrusion depth (mm)</td>
<td>Flow pattern in main tube</td>
</tr>
<tr>
<td>------</td>
<td>-----------</td>
<td>------------------------------------------</td>
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<td>------------------------------------------</td>
<td>------------------------------------------</td>
<td>---------------------------------------------</td>
<td>--------------------------</td>
</tr>
<tr>
<td>Zou and Hrnjak (2013)</td>
<td>R134a</td>
<td>$\dot{m}''$= 92.3–276.8 $x$ = 0.2–0.8</td>
<td>/</td>
<td>Vertical upward/ Horizontal</td>
<td>$C / 15.0, 15.4 / 170$</td>
<td>$SQ / 0.5 /$</td>
<td>13 / 3.8, 7.5</td>
<td>Churn, separated</td>
</tr>
<tr>
<td>Zou and Hrnjak (2013)</td>
<td>R410A</td>
<td>$\dot{m}''$= 21.8–127.3 $x$ = 0.2–0.8</td>
<td>/</td>
<td>Vertical in-line, cross and parallel / Horizontal</td>
<td>$C / 15.0, 15.4 / 170, 300$</td>
<td>$SQ / 0.5 / 1200$</td>
<td>13 / 3.8, 7.5</td>
<td>Churn, separated</td>
</tr>
</tbody>
</table>

Note: $\dot{m}''$: mass flux $x =$: mass quality $D_h$: diameter of header $l_b$: length of branch tube C: circular SQ: square RC: rectangular
In literatures, many scholars have extensively studied two-phase flow distribution behavior in vertical headers under various experimental conditions. One objective was to find ways to avoid flow maldistribution, whereas vapor-liquid separation has been observed and seen as one of the reasons to cause maldistribution. The maldistribution in vertical headers is mainly caused by three forces. 1) Gravitational force intends to separate liquid and vapor phase, resulting in non-homogenous flow regime in the header. 2) The local pressure at each microchannel tube inlet, e.g. each header outlet, is different. From a dividing header to a combing header in a MCHX, pressure drops along headers and microchannels include friction, deceleration / acceleration pressure drop, and local contraction / extraction pressure loss due to the protrusion of tubes inward the header. These pressure drop inevitably result in the different local pressure at each microchannel tube inlet for a dividing header. 3) In addition to pressure difference, vapor and liquid refrigerant have different densities and therefore inertia force such that they have different tendencies to branch out of the header into parallel tubes among each other.

Watanabe et al. (1995) explored two-phase R11 upward flow in a vertical header with five horizontal round tubes. Based on the visualization, the flow regime in the vertical header was initially annular. Along the flow, the velocity became lower due to losing mass. The flow regime transited to froth or slug flow. If the subcooled liquid entered into the header, no liquid could reach the top three branches. Increasing inlet quality up to 0.3 resulted in the top branch tube had more liquid due to the higher velocity. This phenomenon was even more obvious if inlet mass flux was also increased.

Yoo et al. (2002) visualized the distribution of air/water mixture in the vertical header with microchannel tubes. The cross-sectional area was creatively controlled by the
thickness of a spacer plate. The observed flow regimes were churn, annular and free-stream flow. Two flow directions were visualized. The down flows inside the header correspond to annular and free-stream regimes: Lower mass flux, higher quality, and lower flux area produce annular while other orientations result in free-stream regimes. For the up flows, lower flux and high quality correlate with annular or free-stream annular but with higher flux and lower quality the regime shifts to churn.

Cho and Cho (2004) studied mass flow rate distribution and vapor-liquid phase separation of R-22 in multi-microchannel tubes under adiabatic condition. Among the test parameters, the orientation of the dividing header was found to have the most significant effect on the flow distribution and horizontal header showed better distribution characteristics than the vertical header. Three flow directions into the dividing header in Figure 2.10 were compared, namely, in-line flow, parallel flow, cross flow. When R22 was supplied from the bottom (in-line flow), the bottom branch tube had the largest amount of liquid. The liquid amount in the branch tube decreased gradually as the flow going to the top. The distribution was affected by the inlet flow directions but changing inlet quality from 0.1 to 0.3 did not have too much impact.

Figure 2.10 Flow direction into the inlet header (Cho and Cho, 2004)
Lee and Lee (2004) presented the protrusion effect on distribution of air/water mixture in a square vertical header with flat tubes. Annular flow was observed in the header when there was no protrusion. The branch-out liquid amount reduced gradually along the flow. Increasing tubes protrusion changed the flow regime and created the local recirculation, which caused the downstream tubes to have more liquid. The optimal protrusion, i.e. at 1=8 depth, was obtained when the flow in the header was homogeneous and the liquid distribution was almost uniform. The effects of inlet quality and mass flux were not significant when tubes protrusion was present.

Lee (2009a) conducted experiment in a square vertical header with air/water mixture, as shown in Figure 2.11. Fifteen flat tubes were not protruded into the header. Visualization results showed three regions were formed in the header with the annular inlet flow. In the first region, less and less liquid branched out through the tubes along the flow. In the second region, the liquid film separated from the wall at the highest location in this region, so the liquid branch-out increased along the upward flow direction. In the last region, liquid recirculation appeared, but the trend of liquid branch-out was similar to that in the first region. Both liquid film separation location and recirculation in the header affected the distribution profile.
Dshida and Hrnjak (2008) investigated the effect of refrigerant maldistribution in a multi-pass outdoor microchannel heat exchanger (MCHX) on the heating performance of a residential mini-split type A/C and heat pump system. Results showed that refrigerant maldistribution in the 4-pass MCHX tested caused up to about 25% COP reduction compared to the same system using the baseline round tube plain fin heat exchanger. An effort to improve refrigerant distribution was made such that the MCHX was modified into a single-pass design and refrigerant flow was divided into eight smaller flows with a conical distributor before entering the header. But it was not very successful and only 5% COP improvement was obtained upon the worst case.

Zou and Hrnjak (2010) for the first time reported and visualized R410A refrigerant distribution in a vertical intermediate header with feeding refrigerant flow upward. The major application was for a microchannel heat exchanger with a two-pass design used in
residential A/C systems operating in the reverse heat pump mode. The refrigerant quality investigated was up to 0.95. It showed that liquid distribution was in a strong function of total mass flux, inlet quality and the flow regimes.

Overall, two-phase refrigerant maldistribution due to the uneven quality at tube inlets is very complicated and to date there is no generic method or empirical correlations to predict the distribution of the two-phase mixtures. The above studies make it clear that many variables act together: 1) geometric factors (manifold orientation, manifold inner diameter, tube protrusion, location and direction of the evaporator inlet, etc. 2) operating conditions, such as refrigerant mass flux and quality in the manifold, air thermal load, evaporating temperature, etc. Considering the trial-and-error design procedure and a lack of in-depth understanding of two-phase flow behavior in modified headers, good distribution may not be achieved by one specific distribution device for a wide range of operating conditions.

However, it provides the opportunity to utilize the maldistribution and make phases separation possible in separation condensers. In fact, separation of vapor phase from liquid phase have been successfully visualized in recent years in open literatures. The following session would show the evidence of phase separation in dividing headers and/or intermediate headers.

2.2.2 VISUALIZATION OF TWO-PHASE SEPARATION IN HEADERS

In aforementioned Ye et al. (2009), a clear and stable vapor-liquid interface was observed as Figure 2.12 and reported by the authors in the intermediate header of the parallel flow baseline microchannel condenser. The interface was found right above the
baffle with a little vapor bubbles on it. When refrigerant exits this section of the header in the parallel flow condenser, liquid tends to enter the lower microchannel tubes while vapor flows into the upper tubes. As the authors explained, it was due to liquid-vapor separation induced by gravity.

Figure 2.12 Vapor-liquid interface from horizontal view near a baffle in the parallel flow condenser (Ye et al., 2009)

Byun and Kim (2011) investigated the distribution of R410A in a two-pass microchannel heat exchanger in Figure 2.13(a). The inlet quality to the first header was 0.33 and the average inlet quality to the second header was 0.65. The flow patterns are shown in Figure 2.13(b). For the inlet header, a pool was formed at the bottom, so the bottom tubes had more liquid than the top tubes. For the second pass header, a two-phase jet entered into the header and formed a liquid film. Because of the high axial momentum, only a little liquid branched out through the bottom tube. Most liquid exited through the
middle tubes, so there was no more liquid for the top tubes. When mass flux was increased, the distribution was changed: most liquid exited from the top tubes.

![Diagram](image)

(a)

![Flow pattern photos and sketches](image)

(b)

Figure 2.13 (a) Schematic drawing of the parallel flow heat exchanger considered; (b) Flow pattern photos and sketches at \( G = 60 \text{ kg/m}^2\text{s} \) with top inlet and bottom outlet (Byun and Kim, 2011)

Mo et al. (2014) studied a liquid–gas separation unit that can be used in a parallel-flow extraction condenser, and the phase-separation characteristics were determined under different conditions. The unit used air and water to simulate two-phase refrigerant and had an upper arm as the inlet. The lower arm was designed for air outlet whereas
water flowed through the hole at the bottom of the header. Figure 2.14 shows appropriate liquid level maintained in the header for a stratified flow inlet and an annular flow inlet. The liquid-separation efficiency varied with the inlet flow pattern. For an annular flow inlet, slug flow inlet at low liquid-inlet superficial velocities, and stratified flow inlet, the efficiencies exceeded 45%, exceeded 80%, and approached 100%, respectively. The drain limit at high gas-inlet superficial velocities and the flooding limit at high liquid-inlet superficial velocities were fitted analytically by dimensionless correlations, as well as the liquid level in the header between the two limits.

![Flow characteristics in the header for (a) a stratified flow inlet; (b) an annular flow inlet (Mo et al., 2014)](image).

**2.3 SUMMARY AND CONCLUSION**
Phase separation in both extraction condensers and separation condensers has been applied and proven to have potential to improve MCHX performance. Questions remain in terms of how much separation really exists in the vertical intermediate header configuration where buoyance based on density difference serves as the dominating separating mechanism. Two-phase maldistribution in header is complicated in natural, and may significantly reduce heat exchanger performance. Conventional methods have relied on structural modification of dividing headers by adding baffles, distributors, or other geometry-specific remedies. The underlying mechanism behind this idea is to create and maintain a homogenous two phase flow while preventing separated flow regime induced by gravitational and inertial forces. While most attempts focus on improving two-phase or uneven quality maldistribution in the dividing headers, this maldistribution provides the potential for phase separation and further for incorporation with condenser pass circuiting in order to achieve the best performance of a heat exchange.

In the following chapters, the advantage of phase separation condensers will be shown both numerically and experimentally. Numerical modeling will also help to investigate the how to circuit the condenser with pre-assumed separation performance in the header to optimize the performance. Lastly, flow visualization inside a header will serve to study the phase separation performance and separation mechanisms behind.
Chapter 3  SEPARATION IN CONDENSERS AS A WAY TO IMPROVE EFFICIENCY

3.1  INTRODUCTION

This chapter introduces the concept of separation of two-phase flow in condenser as a way to improve condenser efficiency. The benefits of vapor-liquid refrigerant separation and the reason why it will improve the condenser performance will be explained. Numerical studies are presented on the effects of separation on performance of an R134a microchannel condenser. A modeling comparison with a baseline parallel flow microchannel condenser demonstrates that the separation condenser can condense a bigger flow rate. The condenser refrigerant outlet temperature of the separation condenser is lower than the baseline by up to 0.7°C at the same refrigerant inlet state. In addition, condenser pass circuiting with pre-assumed separation results in the header is investigated by the model.

3.2  DESCRIPTION OF THE SEPARATION CONCEPT

Different from a traditional multi-pass condenser starting from the top ending at the bottom, the separation condenser is designed to have vapor-liquid phase separation in the vertical second header by placing the inlet at center of the condenser height. After de-superheating in the 1st pass, vapor flow will be condensed to a certain quality, depending on different operating conditions, at the end of the 1st pass, e.g. inlet of the second header
Relatively active vapor phase is expected to move upward owing to buoyancy based on the density difference between vapor phase and liquid phase. Meanwhile, liquid phase moves downward along gravity direction based on larger density and high viscosity. This separation between vapor and liquid is based purely on density difference, which will provide no additional cost for manufacturing.

Figure is showing several potential design options for condensers with separation circuiting. Upper pass is named vapor pass because it is expected to receive vapor-rich flow, while lower pass is named liquid pass because it is expected to receive liquid-rich flow. The most simplistic designing circuiting would be Figure (a), with only 2 passes. According to customs in manufacturing industries, a 3rd pass can be added after the receiver working as a designated subcooling pass, as shown in Figure (b). However, the drawback of the 2-pass design is the difficulty to keep mass flux of 2nd vapor pass at the same level with that in the traditional-condenser 2nd pass, so prototype separation condensers usually adopt a 3-pass design as shown in Figure (c). 2nd pass is further divided into 2 pass to increase the mass flux in vapor passes. A subcooling pass can also be added at the end after the integrated receiver as shown in Figure (d).

Figure 3.1 Geometries of separation condensers
Separation condensers can be either microchannel tube condensers or tube-and-fin condensers. The optimization of tube number in each pass needs further investigation considering the corresponding separation performance in the 2nd header. Even though separation happens in the aid of vertical configuration of the header and separation mechanism is based on gravitational force, it does not exclude the possibility that headers can have cheap special liquid-vapor separators installed inside for better separation performance after cost design analysis.

3.3 WHY AND HOW

The theoretical rationale behind the advantages of a separation condenser is following. For a condensation process of a refrigerant, heat transfer coefficient usually decreases monotonically with quality. Figure 3.2 is showing the trend of R134a and R410A. At the same mass flux, the high-quality (0.9-1) two-phase refrigerant has 5-7 times higher heat transfer coefficient than a low-quality (0-0.1) two-phase refrigerant, depending on the category of refrigerant. This provides the opportunity to increase the heat exchanger performance by separating vapor from the liquid and using the high heat transfer
coefficient of vapor, in the meanwhile, maintaining or even decreasing a bit the heat transfer coefficient of liquid after separation.

Figure 3.2 R134a and R410A HTC experimental data inside tubes of 1.4 mm hydraulic diameter, Cavallini et al. (2004)

When condenser face area and total heat transfer area are fixed, smart circuiting after separation can potentially improve heat transfer performance by utilizing the high heat
transfer coefficient of separated vapor and giving more heat transfer area (more microchannel tubes) to this vapor flow. If the mass flux of vapor can be kept the same with non-separated refrigerant, it should have a much higher heat transfer coefficient. Overall, it will enhance the heat transfer of the condenser.

Figure 3.3 presents the potential of heat transfer coefficient enhancement of the separation condenser. The heat transfer coefficient profile along the flow path for a constant mass flux in a traditional condenser is shown in Figure 3.3(a). When it is converted to a separation condenser in Figure 3.3(b), vapor pass is divided into the 2nd vapor pass and the 3rd vapor pass to maintain a close mass flux with that in the 2nd pass in Figure 3.3(a). While the total tube number remains constant, the heat transfer coefficient of the 2nd vapor pass and 3rd vapor pass is largely increased compared with that at the same location in Figure 3.3(a). This comparison shows the potential of usage of separation to increase heat transfer performance.

![Figure 3.3 Potential of heat transfer enhancement of the separation condenser](image)
Meanwhile, in downstream intermediate headers, two-phase distribution can also be improved by first separating the liquid and vapor because each phase after separation has more uniform properties than a two-phase flow. However, since flow distribution is less of an issue in condenser than in evaporator, the following discussion by numerical method does not take distribution into account.

3.4 MODEL DESCRIPTION

3.4.1 METHODOLOGY AND EMPIRICAL CORRELATIONS

In order to understand how phase separation affects the condenser performance, an empirical steady-state microchannel condenser model is developed based on $\varepsilon$-NTU method. Finite volume method is used as the discretization scheme, shown in Figure 3.4. This approach has been developed and applied in numerous studies on the modeling of...
MCHX. Litch and Hrnjak (1999) modeled a microchannel condenser with horizontal tubes and vertical headers using finite volume method. The condenser was divided into two zones and was imposed with a quality distribution profile on the two phase zone to obtain a wall temperature profile consistent with the infrared image taken. Kim and Bullard (2001) modeled the performance of an evaporator with CO$_2$ as the refrigerant, and attained a good prediction of cooling capacity and overall pressure drop. The authors considered the refrigerant to be uniformly distributed. Algirdas and Hrnjak (2011) proposed an evaporator model on the basis of the pseudo 2-D finite volume method that considered the refrigerant distribution issue. Each pressure loss and refrigerant distribution issue was considered in detail.

![Diagram of MCHX discretization](image)

Figure 3.4 Example of discretization for one pass in a MCHX
In this study, the approach was extended to evaluate the impact of separation on the performance of a multi-pass microchannel condenser with vertical headers. In the model for the baseline condenser as shown in Figure 3.5(a), finite volume method is conducted in a normal way that the outputs from the current element serve as the inputs into the next element. The model collects the outputs of each element and sums the capacity and pressure drop in each element to provide the total values for the condenser. In the model for the separation condenser as shown in Figure 3.5(b), the model reaches convergence when the upper flow path and the lower flow path between the header ($P_{\text{header}}$) and the receiver ($P_{\text{out}}$) have the equal pressure drop. This is met by varying mass flow rates through two separated flow path for certain separation performance in the second header.

Figure 3.5 Schematic of a two-pass microchannel condenser: (a) Baseline; (b) Separation
Figure 3.5 (cont.)

(b) Separation condenser

Figure 3.6 Zoom in of the separation header

Figure 3.6 shows nomenclature of the quantification phase separation in the separation header. $x_v$ and $x_l$ denote the quality at the vapor exit and the quality at the liquid exit, respectively. They are calculated as

$$x_v = \frac{\dot{m}_{v,v}}{\dot{m}_{v,v} + \dot{m}_{l,v}}$$  \hspace{1cm} (3-1)

$$x_l = \frac{\dot{m}_{v,l}}{\dot{m}_{v,l} + \dot{m}_{l,l}}$$  \hspace{1cm} (3-2)
where \( \dot{m}_{v,v} \) and \( \dot{m}_{l,v} \) are the vapor mass flow rate and liquid mass flow rate, respectively, at the vapor exit; \( \dot{m}_{v,l} \) and \( \dot{m}_{l,l} \) are the vapor mass flow rate and liquid mass flow rate, respectively, at the liquid exit.

\( x_v \) is first assumed for every simulation case with a certain inlet condition \((\dot{m}_{\text{in}}, x_{\text{in}})\) for the header. Different \( x_v \) represents different separation result in the header. When \( x_v > x_{\text{in}} \), phase separation happens.

Figure 3.6 also contains the variables needed to quantify separation in a separation condenser. Two separation efficiencies have been defined for liquid and for vapor, respectively. Vapor separation efficiency is evaluated as the ratio of the separated vapor which flows through the designated vapor outlet divided by the total amount of vapor entering the header, as shown by Eqn. (3-3). Liquid separation efficiency is calculated as the ratio of separated liquid which flows through the designated liquid outlet of the header divided by the amount of liquid supplied to the inlet, as shown by Eqn. (3-4).

\[
\eta_v = \frac{\dot{m}_{v,v}}{\dot{m}_{v,v} + \dot{m}_{v,l}} \quad (3-3)
\]

\[
\eta_l = \frac{\dot{m}_{l,l}}{\dot{m}_{l,l} + \dot{m}_{l,v}} \quad (3-4)
\]

Considering distribution has less of effect in condenser than in evaporator and our focus is on phase separation, distribution issue is not taken into account in the model. Following assumptions are made to simplify the model: (1) Refrigerant distribution is uniform among microchannel tubes in each pass; (2) at each port in the same tube, refrigerant mass flow rate is the same; (3) no heat is conducted along the tube or between...
tube and fins; (4) all headers are adiabatic; (5) incoming air has uniform temperature and velocity profile.

Classic correlations for heat transfer and pressure drop are adopted in the model. Selected correlations are listed in Table 3.1. The heat transfer correlation for the condensing superheated region can be referred to Kondo and Hrnjak (2011). In the model, the superheated routine outputs an inner wall temperature. This inner wall temperature is checked to see if it is below the fluid’s saturation temperature. If the inner wall temperature is below the saturation temperature while the bulk temperature is still above, the previous calculations for the element are disregarded and the inner wall temperature is used to iterate through correlations in the condensing superheated zone.

Table 3.1 Summary of selected heat transfer and pressure drop correlations

<table>
<thead>
<tr>
<th>Item</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Air side</strong></td>
<td></td>
</tr>
<tr>
<td>Heat transfer coefficient</td>
<td>Chang and Wang (1997)</td>
</tr>
<tr>
<td>Pressure drop</td>
<td>Chang and Wang (1996)</td>
</tr>
<tr>
<td><strong>Refrigerant side – Single phase region</strong></td>
<td></td>
</tr>
<tr>
<td>Heat transfer coefficient</td>
<td>Gnielinski (1976)</td>
</tr>
<tr>
<td>Frictional pressure drop</td>
<td>Churchill (1977)</td>
</tr>
<tr>
<td><strong>Refrigerant side – Two phase region</strong></td>
<td></td>
</tr>
<tr>
<td>Heat transfer coefficient</td>
<td>Cavallini et al. (2006)</td>
</tr>
<tr>
<td>Frictional pressure drop</td>
<td>Cavallini et al. (2006)</td>
</tr>
<tr>
<td>Deceleration pressure drop</td>
<td>Cavallini et al. (2009)</td>
</tr>
<tr>
<td><strong>Refrigerant side – Condensing superheated region</strong></td>
<td></td>
</tr>
<tr>
<td>Heat transfer coefficient</td>
<td>Kondo and Hrnjak (2011)</td>
</tr>
</tbody>
</table>
3.4.2 Model validation based on baseline condenser

Modeling results are compared to experimental results that are carried out on a mobile A/C system test facility, on which a 3-pass baseline condenser that had similar configuration with Figure 3.5(a) was used. The tested baseline condenser is shown in Figure 3.7(a) and the schematic circuiting with tube number for each pass is in Figure 3.7(b). A detailed description of the A/C system test facility will be introduced in Section 4.2.

Operating conditions of experiments for modeling validation were set for two air temperatures (35ºC and 45 ºC) per SAE Standard J2765 (2008). The refrigerant condensing pressure is range from 1175 to 1674 kPa, while the mass flow rate of R134a refrigerant 26.1g/s~34.4g/s. Subcooling at the condenser exit was controlled in the range of 0~20ºC.

The model took the refrigerant inlet and air inlet from experiment measurements as the inputs. Refrigerant outlet and air outlet are the outputs. Figure 3.8 shows the comparison of predicted and measured condenser heating capacity. All the data points are
predicted within +/- 2% deviation from the experimental results. Figure 3.9 compares the predicted and measured condenser pressure drop. About 94% of the data points are predicted within +/- 20% deviation from the experimental results. Overall, simulation results show fairly good agreement with experimental results.

Figure 3.8 Comparison of predicted and experimental heating capacity of the microchannel condenser
3.5 RESULTS AND DISCUSSIONS

3.5.1 IMPROVED PERFORMANCE OF SEPARATION CONDENSER

Two microchannel condenser prototypes have been compared and studied using the model presented in 3.4.1 and validated in 3.4.2. Figure 3.10 illustrates the pass arrangement of both condensers, with microchannel tube number shown for each pass. The sum of the tube number of vapor pass and the tube number of liquid pass in the separation condenser is the same with the corresponding pass in the baseline condenser. For example, the 2nd pass in baseline has 17 tubes which is equal to the sum of 9 tubes in 2nd vapor pass and 8 tubes in the 2nd liquid pass in the separation condenser. Besides this, both condensers are parallel-flow, single-slab condensers with louver fins.
Geometrical parameters for the two condensers are kept the same such as length, height, fin geometries, microchannels, etc. Table 3.2 presents the main geometry of the simulated condensers.

![Schematic of the condensers](image)

**Figure 3.10** Schematic of the condensers: (a) Baseline condenser; (b) Separation condenser

<table>
<thead>
<tr>
<th>Item</th>
<th>Number</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube length</td>
<td>680 mm</td>
</tr>
<tr>
<td>Tube pitch</td>
<td>6.8 mm</td>
</tr>
<tr>
<td>Total tube number</td>
<td>46</td>
</tr>
<tr>
<td>Flow depth</td>
<td>13.6 mm</td>
</tr>
<tr>
<td>Fin height</td>
<td>5.8 mm</td>
</tr>
<tr>
<td>Fin pitch</td>
<td>1.21 mm</td>
</tr>
<tr>
<td>Louver length</td>
<td>4 mm</td>
</tr>
<tr>
<td>Microchannel port $D_h$</td>
<td>0.65 mm</td>
</tr>
<tr>
<td>Microchannel ports</td>
<td>17</td>
</tr>
</tbody>
</table>

### 3.5.1.1 Model predicts lower exit temperature

The first comparison criterion is the refrigerant outlet temperature $T_{cro}$. While refrigerant inlet temperature $T_{cri}$ for baseline and separation condensers are kept the same, a lower outlet temperature $T_{cro}$ indicates a more efficient condenser. Meanwhile, refrigerant inlet pressure $P_{cri}$, mass flow rate $\dot{m}$, and air conditions are kept the same. The
inlet temperature and pressure are decided by experiment data for a real separation condenser under air condition per SAE Standard J2765 (2008). Mass flow rate is arbitrarily in order to change $T_{c_{\text{ro}}}$. The simulated operating conditions are shown in Table 3.3.

<table>
<thead>
<tr>
<th>Condition</th>
<th>I35a</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{c_{\text{ri}}}$ [kPa]</td>
<td>1408</td>
</tr>
<tr>
<td>$T_{c_{\text{ri}}}$ [$^\circ$C]</td>
<td>78.9</td>
</tr>
<tr>
<td>$T_{c_{\text{ai}}}$ [$^\circ$C]</td>
<td>35.1</td>
</tr>
<tr>
<td>$V_{c_{\text{ai}}}$ [m/s]</td>
<td>1.5</td>
</tr>
<tr>
<td>$RH_{c_{\text{ai}}}$ [-]</td>
<td>22.5%</td>
</tr>
</tbody>
</table>

Results for comparison of separation condenser with baseline condenser are in Figure 3.11. The exit temperature of separation condenser denoted by orange and red columns is lower than the baseline condenser denoted by blue columns. $\dot{m}$ and $x_v$ are marked for each mass flow case. $x_v$ varies from 0.59 to 0.63. $x_v$ is assumed to be bigger than $x_{\text{in}}$ so that separation happens in the header. From comparison of the results, it is evident that separation condenser has a lower $T_{c_{\text{ro}}}$ than the baseline, which means it has a better heat transfer performance. Subooling varies from 4.1$^\circ$C to 8.9$^\circ$C. The biggest difference of $T_{c_{\text{ro}}}$ between baseline condenser and separation condenser is 0.7$^\circ$C for simulated conditions. In addition, the difference on $T_{c_{\text{ro}}}$ changes with $x_v$. At the same mass flow rate, $T_{c_{\text{ro}}}$ for $x_v=0.9$ is lower than $T_{c_{\text{ro}}}$ for $x_v=0.7$. Therefore, better separation results in the header brings a higher heat transfer. Here separation efficiency increases with $x_v$; decreases with $\dot{m}$. The highest vapor separation efficiency $\eta_v$ and the highest liquid separation efficiency $\eta_l$ both happen at $\dot{m} = 23$ g/s and $x_v = 0.9$. $\eta_{v,\text{max}} = 62.6\%$ and $\eta_{l,\text{max}}$
= 90.0%. Table 3.4 shows the UA value for each condition of the separation condenser is higher than that of the baseline condenser.

![Graph showing comparison of T_{cro} at the same mass flow rate for different separation efficiencies.]

**Figure 3.11** Comparison of $T_{cro}$ at the same mass flow rate for different separation efficiencies

**Table 3.4 Comparison of the UA value from Figure 3.11**

<table>
<thead>
<tr>
<th>Condition</th>
<th>$\dot{m} = 25\text{g/s}$</th>
<th>$\dot{m} = 24\text{g/s}$</th>
<th>$\dot{m} = 23\text{g/s}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Baseline</td>
<td>490.8</td>
<td>464.7</td>
<td>437.9</td>
</tr>
<tr>
<td>Separation $x_v=0.7$</td>
<td>493.5</td>
<td>467.6</td>
<td>440.5</td>
</tr>
<tr>
<td>Separation $x_v=0.9$</td>
<td>492.9</td>
<td>468.3</td>
<td>442</td>
</tr>
</tbody>
</table>

**3.5.1.2 MODEL PREDICTS HIGHER CONDENSATION MASS FLOW RATE**

The second comparison is based on the fact that a more efficient condenser can condense more refrigerant. For the two condensers, air-side conditions are again kept the same with the first comparison. Meanwhile, refrigerant inlet conditions ($T_{cri}$, $P_{cri}$) and exit temperature ($T_{cro}$) are the same. The two condensers in Figure 3.10 are again simulated to
evaluate which one can condense more mass flow. The same operating conditions in Table 3.3 are used. While \( T_{cri} \) is kept as constant, various \( T_{cro} \) represents each comparison case in Figure 3.12. For each simulated case of \( T_{cri} \) and \( T_{cro} \), Figure 3.12 shows the separation condenser constantly condenses more refrigerant than the baseline, by up to 0.4 g/s. Subcooling for baseline is between 4.5°C~9.0°C, which is the same for the separation condenser. Pressure drop of the two condensers is within 3% difference. In addition, \( \dot{m}_{ref} \) for \( x_v=0.9 \) is bigger than \( \dot{m}_{ref} \) for \( x_v=0.7 \), which means improving separation inside the header is beneficial to the condenser performance. The heat conductance \( UA \) values are certainly higher in the separation condenser.

![Figure 3.12 \( \dot{m}_{ref} \) for the two condenser at the same inlet and exit temperatures for different separation efficiencies](image)

Figure 3.12 \( \dot{m}_{ref} \) for the two condenser at the same inlet and exit temperatures for different separation efficiencies
3.5.2 THEORETICAL ANALYSIS OF THE IMPROVEMENTS

To further analyze the observed improved $UA$ of the separation condenser, Figure 3.13 shows the detailed heat transfer coefficient profile of the refrigerant along the parallel flow microchannel tubes at $\dot{m}_{\text{ref}} = 23$ g/s and $x_v = 0.9$ with condition I35a shown in Table 3.3. It can be seen that HTC of in the 2nd vapor pass in the separation condenser is about higher than that in the 2nd pass in the baseline condenser, mainly because local refrigerant almost in the vapor phase ($x_v = 0.9$) flowing along the 2nd vapor pass with similar value of mass flux with that flowing in the 2nd pass of the baseline condenser. Besides, mass flux at the 2nd liquid pass in not significantly decreased compared with that in the 2nd pass in the baseline condenser. This is because refrigerant quality at the entrance of the 2nd liquid pass is 0.37, not too low compared with 0.59 at the header inlet. Meanwhile, mass flux is a little higher at 2nd liquid pass with 13.6 g/s flowing into the 8 tubes out of 17 tubes of the 2nd pass. After entering the 3rd pass, benefits of mass flux at the 3rd liquid pass show even more than the single phase region in the baseline condenser. Consequently, overall $UA$ of the separation condenser is higher than that of the baseline.
Figure 3.13 \( \dot{m} = 23 \text{ g/s} \): comparison of separation condenser \((x_v = 0.9)\) vs. baseline on refrigerant side heat transfer coefficient inside the condenser.

From the analysis of Figure 3.13, it is evident that the geometry of the separation condenser has an impact on the overall heat transfer by being related to the mass flux of the refrigerant in each pass. Also, the separation efficiency also has impact on performance because it determines \( x_v \) coming into the 2nd vapor pass. In order to make the benefits of high-quality two-phase flow maximize, mass flux at the 2nd vapor pass should not be too small. Whether the original designing benefits would exist or not is determined by the trade-off between flow quality and mass flux at each pass.

Figure 3.14 is depicting the relationship between \( \dot{m}_v \) and \( x_v \) for the same condition in Table 3.3. \( \dot{m}_v \) decreases with increasing \( x_v \). That is because with improved separation efficiency, more vapor goes into the upper flow, then frictional factor becomes larger. On the other hand, with more liquid in the lower flow, frictional factor becomes smaller. So mass flow rate in the vapor pass \( \dot{m}_v \) needs to reduce to balance the pressure drop in the
vapor passes and pressure drop in the liquid passes. It also can be noticed that with smaller mass flow rate, the sensitivity of $\dot{m}_v$ to $x_v$ is larger.

![Graph showing separation impact on mass flow rate in the vapor pass at $P_{cr1} = 1408$ kPa, $T_{cr1} = 78.9^\circ$C](image)

Figure 3.14 Separation impact on mass flow rate in the vapor pass at $P_{cr1} = 1408$ kPa, $T_{cr1} = 78.9^\circ$C

### 3.6 SUMMARY AND CONCLUSION

This section concludes that successful separation of liquid and vapor phase in the intermediate header of the microchannel condenser can benefit the condenser performance. Two criteria have been served as the comparison between separation condenser and baseline. With the same geometrical characteristics except the pass circuiting, the separation condenser can condense more refrigerant and/or condense the same mass flow rate of refrigerant to a lower outlet enthalpy.

High quality two-phase flow has a trade-off with mass flux in downstream passes of the separation header. In order to utilize the high heat transfer coefficient of high-quality
vapor-pass flow, mass flux in the separation condenser needs to be kept close to the baseline condenser. This can be achieved by dividing the vapor pass and liquid pass into multi-passes after the separation header.

In next chapter, two real condensers, one with separation circuiting and the other one with traditional circuiting as the baseline, will be tested and compared to confirm the separation effect on a real MAC system.
Chapter 4  EXPERIMENTAL INVESTIGATION OF THE EFFECT OF VAPOR-LIQUID REFRIGERANT SEPARATION ON A MICROCHANNEL CONDENSER

4.1 INTRODUCTION

This chapter presents experimental study of the effect of vapor-liquid refrigerant separation in a microchannel condenser on an automobile AC system. R134a is chosen as the working fluid. A separation condenser and a baseline condenser have been tested in comparison to evaluate the difference of their performance. Two categories of experiments have been conducted: heat exchanger-level test and system-level test. In the condenser-level test, the separation condenser condenses up to 7.4% more mass flow than the baseline at the same inlet and outlet enthalpy; the separation condenser condenses the same mass flow to the lower enthalpy than the baseline condenser does. In the AC system-level test, COP is compared under the same superheat, subcooling and cooling capacity. Separation condenser shows 2.3% to 6.6% higher COP than the baseline condenser.

4.2 EXPERIMENTAL APPARATUS AND METHODS

4.2.1 EXPERIMENTAL FACILITY
In this section, a mobile air-conditioning system test bench will be introduced to experimentally investigate the effect of separation condenser on the performance of a system. Figure 4.1 illustrates the diagram of the experimental setup for the mobile air-conditioning system, while Figure 4.2 shows a photograph of the facility. It was originally a reversible system switching between heating mode and cooling mode to simulate 2014 Nissan Leaf EV in Feng and Hrnjak (2015). In the present study, only the cooling mode was tested. The working component in the indoor chamber is the evaporator. Several modifications/replacements have been made due to experimental limitations. Detailed information for each component of the test bench can be found in Appendix A. The test loop is a basic AC system consisting of a compressor, an indoor microchannel condenser, an outdoor microchannel condenser, a manually-controlled electric expansion valve, a microchannel evaporator and a microchannel evaporator.

Figure 4.1 Schematic drawing of the A/C system test facility
The condenser of test interest was first mounted on a wood board with a rectangular hole made to fit it in. The wood board assembly was then assembled to the outdoor wind tunnel as a gasket. Weather strips were used for sealing the seams between the wood board gaskets and wind tunnel flanges. This gasket-like geometry made it very convenient when changing of heat exchanger is needed. Two 5×5 T type thermocouple grids were made and placed closely upstream and downstream the outdoor heat exchanger. Each row of thermocouples was attached to a fishing line, which was fixed at one end on one side of the wind tunnel inner wall, while the other end was hooked to a spring through a screw eye on the other side of the wind tunnel inner wall. This design allows to take off the thermocouple grid easily from the heat exchanger when it needs to be replaced.
4.2.2 BASELINE AND SEPARATION CONDENSERS DESCRIPTION

The microchannel condenser in the outdoor room is the only changing component in the system during the tests. Pictures and schematics of the baseline condenser and the separation condenser are shown in Figure 4.3 and Figure 4.4, respectively. Number of microchannel tubes is shown on each pass in Figure 4.4. The two are both one-slab, parallel-flow microchannel heat exchangers with the same size of tubes and headers and same tube numbers. Only the circuiting for passes is different. The baseline condenser is a conventional condenser with 3 passes from top to the bottom. Considering to keep the mass flux of vapor at a comparative level, cross sectional area needs to be reduced for the vapor path after separation. So for the separation condenser, after the 2nd pass is divided into vapor and liquid path, each path is divided into 2 passes, namely, 2nd vapor pass - 3rd vapor pass and 2nd liquid pass - 3rd liquid pass. Geometry of the two condensers was measured and summarized in Table 4.1.

![Figure 4.3 Microchannel condensers used in this study: (a) Baseline condenser; (b) Separation condenser](image)
Figure 4.4 Schematic of the condensers: (a) Baseline condenser; (b) Separation condenser

Table 4.1 Microchannel condenser geometry

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Width w/ headers [mm]</td>
<td>710</td>
</tr>
<tr>
<td>Width w/o headers [mm]</td>
<td>680</td>
</tr>
<tr>
<td>Height w/ side plates [mm]</td>
<td>390</td>
</tr>
<tr>
<td>Height w/o side plates [mm]</td>
<td>405</td>
</tr>
<tr>
<td>Depth [mm]</td>
<td>12</td>
</tr>
<tr>
<td>Inlet tube outer diameter [mm]</td>
<td>12.0</td>
</tr>
<tr>
<td>Outlet tube outer diameter [mm]</td>
<td>15.9</td>
</tr>
<tr>
<td>Microchannel tube thickness [mm]</td>
<td>1.01</td>
</tr>
<tr>
<td>Microchannel tube pitch [mm]</td>
<td>6.8</td>
</tr>
<tr>
<td>Fin thickness [mm]</td>
<td>0.064</td>
</tr>
<tr>
<td>Number of slabs</td>
<td>1</td>
</tr>
<tr>
<td>Total number of tubes</td>
<td>54</td>
</tr>
<tr>
<td>Number of side plates</td>
<td>2</td>
</tr>
<tr>
<td>Number of fins per row</td>
<td>560</td>
</tr>
</tbody>
</table>

4.3 DATA REDUCTION AND UNCERTAINTY

Compressor speed $V_C$ and torque $F_C$ applied on the compressor shaft are measured.

The compressor work is product of these two, calculated as:
Both a refrigerant-side energy balance \( (Q_r) \) and an air-side energy balance \( (Q_a) \) are used to calculate system capacities. The system capacity is based on an average value of the two energy balances:

\[
Q = \frac{Q_{ref} + Q_{air}}{2} \quad (4-2)
\]

On the air side \( Q_{air} \), for the evaporator, air-side capacity can be written as Eqn. (4-3).

\[
Q_{ea} = \dot{m}_{idn,air} (h_{eai} - h_{idn}) \quad (4-3)
\]

\[
\dot{m}_{idn,air} = \dot{V}_{idn,air} \rho_{idn,air} \quad (4-4)
\]

\[
\dot{V}_{idn,air} = C_{d,idn} A_{idn} \sqrt{\frac{2\Delta P_{idn}}{\rho_{idn,air}}} \quad (4-5)
\]

\[
A_{idn} = n \frac{\pi D_{idn}^2}{4} \quad (4-6)
\]

where \( h_{eai}, h_{idn} \) denote evaporator air inlet and indoor nozzle outlet enthalpy. Indoor nozzle enthalpy \( h_{idn} \) is the enthalpy of well mixed flow through each area of the evaporator so it is chosen to represent the evaporator outlet enthalpy to deduct capacity. Air flow rate \( \dot{m}_{idn,air} \) is calculated from Eqn. (4-4). Air enthalpy and air density at the nozzle \( h_{eai}, h_{idn}, \rho_{idn,air} \) are based on humid air, which are obtained from temperature, pressure and relatively humidity measurement at corresponding locations. The volumetric flow rate \( \dot{V}_{idn,air} \) is determined by (4-5). \( C_{d,idn} \) is a dimensionless constant calibrated from previous projects as 0.975. \( A_{idn} \) is the flow area of the nozzle area calculated through (4-6).
For the condenser, air-side capacity is calculated by Eqn. (4-7) in a similar way as the evaporator, where $m_{odn,air}$ denotes the outdoor nozzle air mass flow rate and $h_{cai}, h_{odn}$ denote condenser air inlet and outdoor nozzle outlet enthalpy. The detailed deductions remain the same with indoor nozzle. Since dew point temperature in outdoor chamber is not measured, and its effect on mass flow rate is small, the monitored dew point temperature from website of Department of Atmospheric Science is used for calculation of relative humidity.

$$Q_{cai} = m_{odn,air} (h_{cai} - h_{odn})$$  \hspace{1cm} (4-7)

$$m_{odn,air} = \dot{V}_{odn,air} \rho_{odn,air}$$  \hspace{1cm} (4-8)

$$\dot{V}_{odn,air} = C_d,odn A_{odn} \sqrt{\frac{2 \Delta P_{odn}}{\rho_{odn,air}}}$$  \hspace{1cm} (4-9)

$$A_{odn} = n \frac{\pi D_{odn}^2}{4}$$ \hspace{1cm} (4-10)

On the refrigerant side, refrigerant capacity equal the product of refrigerant mass flow rate and enthalpy difference, e.g. Eqn. (4-11) and (4-12). $h_{eri}, h_{ero}$ denote refrigerant enthalpy at the evaporator inlet and outlet enthalpy. Refrigerant enthalpy for subcooled liquid and superheated vapor are determined by refrigerant pressure and temperature. As no liquid is allowed to enter compressor through accumulator, for steady state operation, when there is no observable superheat at these locations, the vapor can be assumed to be saturated vapor, while ignoring the refrigerant that is dissolved in lubricant. For situations where no subcooling is observable at condenser outlet, the refrigerant mass flow rate cannot be measured by the mass flow meter, and heat exchanger exit enthalpy cannot be calculated with temperature and pressure inputs, therefore refrigerant side capacity is
unknown. In summary, the refrigerant enthalpies at different location of the system are determined by following the logic in Table 4.3.

\[ Q_{cr} = \dot{m}_r (h_{ero} - h_{eri}) \]  
\[ Q_{cr} = \dot{m}_r (h_{eri} - h_{cro}) \]  

(4-11) \hspace{1cm} (4-12)

Table 4.2 Refrigerant enthalpy at different location

<table>
<thead>
<tr>
<th>Location</th>
<th>Temperature</th>
<th>Pressure</th>
<th>Phase</th>
<th>Enthalpy</th>
</tr>
</thead>
<tbody>
<tr>
<td>cpri</td>
<td>( T_{cpri} )</td>
<td>( P_{cpri} )</td>
<td>SH/V\text{ sat}</td>
<td>( h(\text{Ref, T, P}) / h(\text{Ref, P, } x=1) )</td>
</tr>
<tr>
<td>cpro</td>
<td>( T_{cpro} )</td>
<td>( P_{cpro} )</td>
<td>SH</td>
<td>( h(\text{Ref, T, P}) )</td>
</tr>
<tr>
<td>ohri</td>
<td>( T_{ohri} )</td>
<td>( P_{ohri} )</td>
<td>SH</td>
<td>( h(\text{Ref, T, P}) )</td>
</tr>
<tr>
<td>ohro</td>
<td>( T_{ohro} )</td>
<td>( P_{ohro} )</td>
<td>SC/TP</td>
<td>( h(\text{Ref, T, P}) / \text{NA} )</td>
</tr>
<tr>
<td>xri</td>
<td>( T_{xri} )</td>
<td>( P_{xri} )</td>
<td>SC/TP</td>
<td>( h(\text{Ref, T, P}) / \text{NA} )</td>
</tr>
<tr>
<td>eri</td>
<td>( T_{eri} )</td>
<td>( P_{eri} )</td>
<td>TP</td>
<td>( h_{xri} )</td>
</tr>
<tr>
<td>ero</td>
<td>( T_{ero} )</td>
<td>( P_{ero} )</td>
<td>SH/V\text{ sat}</td>
<td>( h(\text{Ref, T, P}) / h(\text{Ref, P, } x=1) )</td>
</tr>
</tbody>
</table>

The coefficient of performance (\( COP \)) is then as:

\[ COP = \frac{Q_c}{W_c} \]  

(4-13)

\( Q_c \) is calculated by Eqn. (4-2) except for the cases where refrigerant side capacity is not available. The uncertainties for the calculated system parameters, namely the cooling capacity, compressor power, and \( COP \) are calculated using the method given by Moffat (1988). According to this method, the function \( U \) is assumed to be calculated from a set of \( N \) measurements (independent variables) represented by

\[ U = U(X_1, X_2, X_3, ..., X_N) \]  

(4-14)

Then the uncertainty of the result \( U \) can be determined by combining the uncertainties of the individual terms using a root-sum-square method, i.e.
\[
\delta U = \sqrt{\sum_{i=1}^{N} \left( \frac{\partial U}{\partial X_i} \delta X_i \right)^2}
\]  

(4-15)

Using uncertainties for measured variables presented in Table 4.3, the uncertainties of the calculated parameters are determined. The total uncertainties of \( Q \) and COP are estimated to be 3% and 3.5%, respectively. For the system operating in both modes, independently obtained air and refrigerant side energy balances were available and agreed within 3%.

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Unit</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant pressure</td>
<td>kPa</td>
<td>±3.56</td>
</tr>
<tr>
<td>Refrigerant pressure drop</td>
<td>kPa</td>
<td>±1.52</td>
</tr>
<tr>
<td>Nozzle pressure drop</td>
<td>Pa</td>
<td>±6.5</td>
</tr>
<tr>
<td>Temperature</td>
<td>°C</td>
<td>±0.5</td>
</tr>
<tr>
<td>Refrigerant mass flow rate</td>
<td>g/s</td>
<td>±0.5%</td>
</tr>
<tr>
<td>Compressor speed</td>
<td>rpm</td>
<td>±5</td>
</tr>
<tr>
<td>Compressor torque</td>
<td>N·m</td>
<td>±0.05</td>
</tr>
</tbody>
</table>

### 4.4 SEPARATION EFFECT ON CONDENSER AS A HEAT EXCHANGER

The separation condenser and the baseline condenser were first tested on the experimental system in a way that focuses on condenser only: parameters only regarding to condenser were taken into account for comparison. Two criteria have been used to evaluate the condensing performance: 1) exit temperature; 2) condensate flow rate. Table
lists the operating conditions, which includes three compressor speed settings (low, medium and high).

Table 4.4 Operating conditions

<table>
<thead>
<tr>
<th>Compressor speed setting</th>
<th>(T_{\text{indoor}})</th>
<th>(T_{\text{outdoor}})</th>
<th>(\dot{M}_{\text{indoor}})</th>
<th>(v_{\text{outdoor}})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low (I45 / I35a)</td>
<td>35</td>
<td>45 / 35</td>
<td>9.5</td>
<td>1.5</td>
</tr>
<tr>
<td>Medium (L45 / L35a)</td>
<td>35</td>
<td>45 / 35</td>
<td>9.5</td>
<td>2</td>
</tr>
<tr>
<td>High (M45 / M35a)</td>
<td>35</td>
<td>45 / 35</td>
<td>9.5</td>
<td>3</td>
</tr>
</tbody>
</table>

4.4.1 **FIRST BENEFIT AS A HEAT EXCHANGER: REDUCED EXIT TEMPERATURE**

The first criterion for evaluating a condenser is the exit temperature. A more efficient condenser should condense the same flow rate of refrigerant to a lower temperature at the same air conditions. In experiments, inlet temperature and refrigerant mass flow rate were also controlled the same. EEV opening and compressor speed were adjusted together to control the inlet temperature and mass flow rate for refrigerant the same. Charge was also adjusted to control the condensing temperature the same in order to keep the same refrigerant-air temperature difference.

Subcooling was always ensured since if exit is in two-phase region this criterion loses its validity and refrigerant exit temperature would only be a function of pressure drop.
Figure 4.5 Comparison of refrigerant exit temperatures in baseline and separation condensers at the same air conditions
Figure 4.5 shows separation condenser constantly has a lower exit temperature than the baseline condenser for the same air side flow rate and temperature. 45 in Figure 4.5(a) denotes the outdoor temperature of 45°C and 35 in Figure 4.5(b) denotes the outdoor temperature of 35°C. Shaded columns represent exit temperature for each case while top of the dotted column represents condensing temperature. Condensing temperature was within ±0.2°C in separation condenser and baseline condenser for each case. The length of dotted area represents the degree of subcooling. Except for L35a in Figure 4.6(b), a bigger subcooling is also achieved in the separation condenser. The reason for the exception in L35a is probably because: 1) separation has the drawback of bigger pressure drop; 2) refrigerant-air temperature difference for the separation condenser is lower than the baseline.

As shown in Figure 4.6, the condensation process for test case M35a is plotted on a R134a T-h diagram. Separation condenser condenses the refrigerant to a lower exit temperature. However, air exit temperature and thus capacity keep almost the same reading from the dotted air temperature lines. The reason for that is probably due to big subcooling (10°C) and bigger pressure drop of the separation condenser. Refrigerant exit temperature approaches the air temperature which results in “pinched” area at the end of condenser. With lower exit temperature of the separation condenser, condenser performance would be even more pinched. Figure 4.7 is showing the infrared image of the separation condenser surface. The 4th pass, 3rd liquid pass and end of the 3rd vapor pass are full of subcooled liquid. The portion of two phase region decreased as subcooling region increased. Bigger pressure drop in the separation condenser made the
LMTD after two-phase zone become smaller, thus reducing the heat transfer. Future work should compare the two condensers at reduced degree of subcooling.

So for next criterion, subcooled area in both condensers would be kept the same by having the same inlet and outlet refrigerant temperature while mass flow rate served to be the comparison variable. Compared to the first criteria of exit temperature, performance enhancement due to mass flow rate should be more obvious.

![Figure 4.6 T-h diagram for M35a: compare refrigerant exit temperature](image)

Figure 4.6 T-h diagram for M35a: compare refrigerant exit temperature
4.4.2 SECOND BENEFIT AS A HEAT EXCHANGER: INCREASED CONDENSATION FLOW RATE

The second criterion for evaluating a condenser is the condensate flow rate. A more efficient condenser should have more condensate at the same air conditions. In experiments, inlet temperature and exit temperature of refrigerant were also controlled the same. The controlling method is same with the first one, EEV opening, compressor speed and refrigerant charge to the system were adjusted simultaneously to control the inlet temperature, exit temperature and condensing temperature the same for fair comparison. Subcooling was again always ensured.
Figure 4.8 Comparison of condensate mass flow rate in baseline and separation condensers at the same air conditions
Figure 4.8 shows separation condenser constantly has a larger condensate mass flow rate than the baseline condenser for the same air side flow rate and temperature. Again, 45 in Figure 4.8 (a) denotes the outdoor temperature of 45°C and 35 in Figure 4.8 (b) denotes the outdoor temperature of 35°C. Inlet pressure and exit pressure are marked with each column representing each condition. The average pressure is again kept around the same which gives condensing temperatures within ± 0.2°C in separation condenser and baseline condenser. The same inlet temperature and exit temperature for separation and baseline are also marked with each comparing case. The mass flow rate improvement varies from 1.4% to 7.4% in the six conditions.

Figure 4.9 shows the condensation process for test case M35a plotted on a R134a T-h diagram. With 7.4% more condensate flow rate, air exit temperature was higher and capacity was successfully increased by 5.1% compared to Figure 4.7 of the first criteria. The 7.4% increment of mass flow rate elevated the heat transfer coefficient. At the same time, larger mass flow rate made pressure drop higher which resulted in a lower average condensing temperature, which can be read from Figure 4.9, and a smaller refrigerant-air LMTD. But the benefits of higher heat transfer coefficient dominated the decrease of LMTD by the ending with a 5.1% increase on capacity is achieved. Thus, the effect of separation made the condenser receive and condense more refrigerant and thus having a higher capacity.
4.5 **MAXIMUM COP ENHANCEMENT AT MATCHED COOLING CAPACITY**

To have the only one measure of performance improvement (here *COP*) the cooling capacity (*Q_c*) is maintained constant by varying the compressor speed in system with the baseline condenser and in system with the separation condenser. Compressor speed was first set constant at 1625 rpm for the baseline system. Charge was then gradually increased to achieve different subcooling for the system. In the system with separation condenser, the compressor speed of the separation system was adjusted accordingly to match the capacity to the baseline. Table 4.5 shows the operating conditions for this system test.
Table 4.5 Operating conditions for system experiments at matched capacity

<table>
<thead>
<tr>
<th>Items</th>
<th>Unit</th>
<th>Condenser side</th>
<th>Evaporator side</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air inlet temperature</td>
<td></td>
<td>35</td>
<td>35</td>
</tr>
<tr>
<td>Air flow</td>
<td>m³/s</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Relative humidity</td>
<td></td>
<td>-</td>
<td>Dry condition</td>
</tr>
<tr>
<td>Evaporator exit superheat</td>
<td>ºC</td>
<td></td>
<td>10</td>
</tr>
<tr>
<td>Condenser subcooling</td>
<td>ºC</td>
<td>5, 7, 10, 13,16</td>
<td></td>
</tr>
</tbody>
</table>

As shown in Figure 4.10, among the tested subcoolings, subcooling of 5ºC gives the maximum COP enhancement.

![COP vs. Subcooling](image)

Figure 4.10 COP comparison for refrigeration cycles with baseline condenser and separation condenser at superheat of 10ºC and at the matched capacity.

As shown in Figure 4.11, corresponding T-h diagram and p-h diagram for superheat of 15ºC and subcooling of 5ºC are shown to identify primary source of COP
improvement. Capacities are maintained around 4.76 kW with a deviation less than ±1%. As shown in Figure 4.11, separation system results in up to 6.6% higher COP increment than baseline system. The compressor speed in the separation system is reduced approximately 155 rpm (roughly 9%) below that of the baseline system to provide the same capacity. In general, lower speed can result in higher compressor efficiencies. For current experimental result isentropic efficiency in the separation system is 2.9% higher than in the baseline system. Thus, improved isentropic efficiency does not significantly affect the system performance.

In the separation system, mass flow rate in the separation system is smaller because of lower compressor speed, so the difference of specific enthalpy between the evaporator outlet and inlet is larger. The condenser exit pressure is reduced 46 kPa primarily due to improved condenser heat transfer coefficient. The condensing temperature in the separation system is 39.6°C, lower than the condensing temperature of 41.9°C in the baseline system. The reduction of condensing temperature decreases the compressor discharge pressure and therefore reduces the compressor work. Both of these effects reduce the compressor work, resulting in a greater COP. At the matched capacity, COP is higher in the separation system than in the baseline system by 6.6%.
Figure 4.11 Refrigeration cycles for baseline and separation for a superheat of 15°C and at the matched capacity
4.6 SUMMARY AND CONCLUSIONS

Flow separation is implemented in the microchannel condenser of an R134a MAC system. An experimental comparison to a conventional condenser was conducted using the same components and under the same operating conditions.

Separation effect was first tested only in a heat exchanger. At the same refrigerant inlet temperature and exit temperature, a maximum improvement of 7.4% on the refrigerant mass flow rate was achieved in the separation condenser while the heating capacity increased by 5.1% over the conventional condenser. Having the same mass flow rate, the separation condenser was able to condense the refrigerant to a lower temperature.

Second, the effect of separation was tested in the AC system measuring the COP enhancement at matched cooling capacity. When the cooling capacity is matched by adjusting the compressor speed, the COP improvement was 1.7% to 6.6% higher compared to the system with the conventional condenser.

This section concludes that the separation circuiting can benefit the condenser performance. Because it is very difficult to measure directly the mass flow in the separated passes inside the condenser, a valid question would then be how much separation really exists in the intermediate header. To solve this problem, a component-level experiment facility is built up with visualization access to study the separation of two-phase flow in a vertical header as part of a MCHE.
Chapter 5  SEPARATION OF LIQUID-VAPOR TWO-PHASE FLOW IN A VERTICAL HEADER

5.1 INTRODUCTION

This chapter presents the experimental study of separation of two-phase flow in a vertical header of MCHX based on quantified visualization using fast camera. A header prototype is made that has an inlet in the longitudinal center part. Two sub-passes downstream are designed, lower for liquid and upper vapor flow. The header for experiment is clear to provide visual access. R-134a is used as the fluid of interest and mass flux through the inlet microchannels is controlled between 55 kg/(m²s)-195 kg/(m²s). The experiment results indicate that ideal separation in that header can happen at low mass flux up to 70 kg/(m²s). Results are presented in function of liquid and vapor separation efficiencies (\(\eta_l, \eta_v\)). Flow patterns inside the header are identified and analyzed to study the mechanisms for liquid-vapor separation. The efficiency deteriorates dramatically when the recirculation regions elevates up to the vapor exit, with increasing inlet flow rate and/or quality. Potential design options to improve two-phase separation are discussed. The objective should be to avoid or at least delay the recirculation region from reaching the vapor exit by reducing the liquid upward momentum or the vapor upward velocity, and decreasing liquid and vapor force interaction.
5.2 EXPERIMENT SETUP AND MEASUREMENTS

5.2.1 FACILITY

The schematic drawing of the test loop for gas-liquid separation in the vertical header is shown in Figure 5-1. A diaphragm pump is used to supply the desired mass flow rate of the working refrigerant with a certain discharge pressure. Following the flow direction, the refrigerant flow rate in the subcooled state is measured in the pump discharge line by a Coriolis-type mass flow meter MF_t. An electric pre-heater controls the refrigerant to be heated to saturated liquid prior to entering the test section. System pressure and temperature are measured by a pressure transducer (P) and a T-type thermocouple (T).

Test flow is separated into two streams in the test section and Figure 5-2 is a photograph of it before installing electrical heating tapes and insulation wraps. When both of the 3-way valves are switched to the left, system is in the test mode. Each of two streams goes into a flash tank, named as liquid exit and vapor exit, respectively. The flash tanks collect liquid and remove vapor phase that arrives at the flash tanks. Over a given amount of time, the mass of liquid phase collected in each of the two tanks is measured, yielding a time averaged liquid mass flow rate. Meanwhile, vapor flow rate from each of the collection tank is measured by flow meters MF_v and MF_l, respectively.

A metering valve is installed on the flow path of liquid exit with intention to simulate various flow resistance downstream of the separation header in a real condenser. Between point 1 and 2 a differential pressure transducer (∆P) is installed to measure the downstream pressure drop along the liquid exit flow path. After the testing apparatuses,
refrigerant will flow through a plate condenser, a refrigerant receiver, and a tube-tube heat exchanger as a subcooler and finally come back to the liquid pump.

A high-speed CCD camera was used to visualize the flow patterns in the test header normally at a recording speed range of 1600-2230 fps (frames per second).

![Figure 5.1 Schematic of the experimental facility](image-url)
A detailed schematic of the test section is shown in Figure 5.3. The flow passes of the test section are made by 36 (21 in 1st pass, 11 in 2nd-vapor pass and 4 in 2nd-liquid pass) aluminum microchannel tubes and the header was made by circular PVC transparent tube for visualization. The tube numbers are selected based on a real separation condenser. Inlet quality is controlled by 42 electric heating tapes (2 for each inlet microchannel tube), which output the calculated amount of heat to achieve the required quality in the entrance of the test headers. A summary of the geometry of the visualization header is found in Table 5.1.
Figure 5.3 Detailed schematic of the test section

Table 5.1 Geometries of the visualization header

<table>
<thead>
<tr>
<th>Item</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet microchannel tube number</td>
<td>21</td>
</tr>
<tr>
<td>Vapor outlet microchannel number</td>
<td>11</td>
</tr>
<tr>
<td>Liquid outlet microchannel number</td>
<td>4</td>
</tr>
<tr>
<td>Header length</td>
<td>281 mm</td>
</tr>
<tr>
<td>Header inner diameter</td>
<td>15.8 mm</td>
</tr>
<tr>
<td>Microchannel tube pitch</td>
<td>7 mm</td>
</tr>
</tbody>
</table>

In order to simulate the case for a real condenser, sizes of the header and microchannel tubes for test are selected based on the same real condenser mentioned above. The cross-section dimensions for both are shown in Figure 5.4. The PVC tube dimensions and the microchannel dimensions are chosen to be close to the real header dimensions within 10%. The horizontal microchannel tube for test has a width of 13.6 mm and a thickness of 1.01 mm. There are 17 ports with an equivalent hydraulic diameter of 0.65 mm as shown in Figure 5.5.
5.2.2 DATA REDUCTION AND UNCERTAINTY ANALYSIS

Considering the possible heat loss at the heating part of the test section, the inlet refrigerant quality of the test header $x_{in}$ is not deducted from the electric power measurement. As shown in Figure 3.6 and Figure 5.3, $x_{in}$ is calculated based on the vapor and liquid mass flow rates at the two exits, as following:

$$x_{in} = \frac{\dot{m}_{v,v} + \dot{m}_{v,l}}{\dot{m}_{in}}$$  \hspace{1cm} (5-1)

where $\dot{m}_{in}$ is the inlet mass flow rate, $\dot{m}_{v,v}$ and $\dot{m}_{v,l}$ are the vapor mass flow rate at the vapor exit and the vapor mass flow rate at the liquid exit, respectively.
Separation efficiencies $\eta_l$, $\eta_v$ are defined the same as Equation (3-3) and (3-4), the qualities at the two exits are to quantify the condition at the two exits. They can be calculated as

$$\eta_v = \frac{\dot{m}_{v,v}}{\dot{m}_{v,v} + \dot{m}_{v,l}}$$  \hspace{1cm} (5-2)

$$\eta_l = \frac{\dot{m}_{l,l}}{\dot{m}_{l,l} + \dot{m}_{l,v}}$$  \hspace{1cm} (5-3)

$$x_v = \frac{\dot{m}_{v,v}}{\dot{m}_{v,v} + \dot{m}_{l,v}}$$  \hspace{1cm} (5-4)

$$x_l = \frac{\dot{m}_{v,l}}{\dot{m}_{v,l} + \dot{m}_{l,l}}$$  \hspace{1cm} (5-5)

where $x_v$ and $x_l$ are the quality at the vapor exit and the quality at the liquid exit, respectively; $\dot{m}_{l,v}$ and $\dot{m}_{l,l}$ are the liquid mass flow rate at the vapor exit and the liquid mass flow rate at the liquid exit, respectively.

The overall measurement uncertainty in quality and the phase separation efficiency are calculated using the same root-sum-square combination as in Chapter 4.2.2. If a function $U$ is assumed to be calculated from a set of $N$ measurements (independent variables) represented by

$$U = U\left(X_1, X_2, X_3, ..., X_N\right)$$  \hspace{1cm} (5-4)

then the uncertainty of the result $U$ can be determined by combining the uncertainties of the individual terms using a root-sum-square method, i.e.

$$\delta U = \frac{{\sum_{i=1}^{N} \left(\frac{\partial U}{\partial X_i} \delta X_i\right)^2}}{\sqrt{N}}$$  \hspace{1cm} (5-5)
Using the accuracies for the measured variables presented in Table 4.3, the maximum values of measurement uncertainty for separation efficiency (\(\eta\)) and quality (\(x\)) are ±1.7% and ±2.5%, respectively. The values of zero stability for mass flow meters \(MF_t\), \(MF_v\), and \(MF_l\) can be found in Appendix B.

### Table 5.2 Measured uncertainties

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Unit</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet liquid mass flow rate (MF_t)</td>
<td>g/s</td>
<td>±0.10% ± ([(\text{zero stability} / \text{flow rate}) \times 100])%</td>
</tr>
<tr>
<td>Vapor mass flow rate (MF_v / MF_l)</td>
<td>g/s</td>
<td>±0.50% ± ([(\text{zero stability} / \text{flow rate}) \times 100])%</td>
</tr>
<tr>
<td>Heat Input</td>
<td>W</td>
<td>±0.2%</td>
</tr>
<tr>
<td>Liquid Level in Flash Tanks</td>
<td>mm</td>
<td>±2</td>
</tr>
<tr>
<td>Time</td>
<td>s</td>
<td>±2</td>
</tr>
<tr>
<td>Pressure</td>
<td>kPa</td>
<td>±0.5</td>
</tr>
<tr>
<td>Pressure drop</td>
<td>kPa</td>
<td>±0.2</td>
</tr>
<tr>
<td>Temperature</td>
<td>ºC</td>
<td>±0.5</td>
</tr>
</tbody>
</table>

#### 5.2.3 OPERATING CONDITIONS

In a separation condenser, the upper vapor path and the lower liquid path would finally mix in a combining header [Figure 5.6(a)] or a receiver to get out of the condenser. Usually after the receiver there is a subcooling pass. As has been shown in 3.5.2 downstream geometry and air load all affect the downstream flow resistance and give different pressure drop, thus altering the flow separation phenomenon in the separation header.
In the experiment, this effect can be simulated by adding two metering valves downstream the two flash tanks as shown in Figure 5.6(b). Between 1 and 2, the pressure drop along the vapor path and liquid path is equal to each other. So different separation phenomenon resulting from different the flow resistance can be achieved by adjusting the metering valves. In the following sessions, a series of experiments are first conducted when downstream flow resistance for the header are kept to a minimum by having the metering valves at maximum open degree. Then, the downstream flow resistance is varied by closing the valves to see the effect on separation efficiencies.

![Figure 5.6](image)

Figure 5.6 (a) Downstream pressure drop $\Delta P$ in a real condenser; (b) Downstream pressure drop $\Delta P$ in experiment system

As to initial experiments, the two metering valves are left with the maximum opening, which gives the test header minimum downstream flow resistance.

Total mass flow rate is varied from 8.4 g/s to 30 g/s, which corresponds to mass flux of 54 kg/m$^2$·s - 193 kg/m$^2$·s through the microchannel tubes in the first pass of the condenser. The developing length for the saturated liquid prior to entering the first header
is 1.2 m. Inlet quality to the header is controlled in a range from 0.05 to 0.25, which is on the lower end of the quality in a real condenser. However, it is where separation phenomenon and mechanism change, so this range is of interest to study phase separation as a starter.

5.3 SEPARATION EFFICIENCY WITH MINIMUM DOWNSTREAM FLOW RESISTANCE

5.3.1 EFFECT OF SEPARATION LOAD

Figure 5.7(a) and Figure 5.7(b) present the results of separation efficiencies ($\eta_l$ and $\eta_v$) with different separation loads (liquid mass flow rate) for R134a. Inlet quality gradually increases for 4 typical mass flow rates: 10 g/s, 16 g/s, 20 g/s, and 25 g/s. At low mass flow rates, inlet quality shows strong impact on liquid separation. At 10 g/s with $x=0.06$, $\eta_l$ is 0.38, which means more than of the liquid is sending out from the vapor exit. However, as $x$ increases to 0.14, $\eta_l$ increases dramatically to 1, which means no liquid is escaping from the vapor exit.

However, as mass flow rate becomes higher up to 20 g/s, first, $\eta_l$ becomes smaller; second, plots become flatter meaning it does not change with inlet quality as much as it does at low mass flow rates. The effect of quality in this range of mass flow rate results in a maximum value of $\eta_l$ at a certain quality.

As the mass flow rate increases further to 25 g/s, for different $x$, $\eta_l$ is changing within 5%. Thus, inlet quality is losing the impact on liquid separation.
For vapor separation efficiency $\eta_v$ in Figure 5.7(b), except for 10 g/s under 0.15 inlet quality, $\eta_v$ keeps dropping with the increase of inlet quality. The dropping amount varies from 20% to 14% as mass flow rate increases. It is apparent that both high mass flow rate and high inlet quality are detrimental to the separation of vapor. It is extrapolated that with higher amount of vapor entering the header, more vapor would highly mix the flow inside the header and make the flow approach to homogeneity, thus, more vapor is getting out through the liquid exit.

Figure 5.7 (a) Liquid separation efficiency $\eta_l$ vs. Inlet quality $x$; (b) Vapor separation efficiency $\eta_v$ vs. Inlet quality $x$
To study the separation results better, visualization is introduced as a way to reveal the physics behind. Figure 5.8 presents results of separation efficiencies ($\eta_l$ and $\eta_v$) at low mass flow rate for different inlet qualities while Figure 5.9 shows it for higher mass flow rate.

At low mass flow rates of 10 g/s, inlet quality shows strong impact on separation. Separation efficiency $\eta_l$ changes from 0.38 to 1 while vapor separation efficiency $\eta_v$ drops from 0.83 to 0.66. At low qualities and low mass flow rates, for example 10 g/s at $x=0.06$ in Figure 5.8, the header is almost filled with liquid. It is a consequence of difficulty to send liquid through the 2nd pass out. In these operating conditions pressure in the second header is low. Liquid flow through the second liquid pass is provided mostly by hydrostatic head ($\rho gh$), so that liquid is being accumulated in the header to increase flow rate. Consequently, vapor separation is poor and much liquid leaves through the vapor side resulting in a $\eta_l$ as low as 0.38.
Figure 5.8 Separation phenomenon at 10 g/s

When the quality is increased to $x_{in}=0.14$ at the same flow rate of 10 g/s, $\eta_l$ increases dramatically to 1. Quality at vapor exit is $x_v=1$. Pressure in the second header had increased and mass flow rate of liquid is reduced resulting in good drainage of the liquid providing possibility for a good separation in the second header. Liquid coming from microchannels falls relatively unobstructed to the bottom of the header while vapor goes
up passing falling liquid easy thanks to low vapor flow and thus velocity. With further increase of inlet quality to 0.25, $\eta_l$ is still 1 while $\eta_v$ drops to 0.664. Visualization shows that vapor extends to the liquid exit. Quality at liquid exit $x_l$ is 0.096.

At high mass flow rate 30 g/s, Figure 5.9 shows: 1) both liquid and vapor separation efficiencies drop compared with the same quality at 10 g/s; 2) liquid or vapor separation
efficiency does not change with changing inlet quality as dramatically as it does at low mass flow rate. Two-phase flow coming out of the inlet microchannel tubes first splashes on the inner wall of the header, then is divided almost equally into two streams: upward and downward. For the three cases of inlet quality to the second header \( x_{in} \) (0.06, 0.13, and 0.18), average quality at vapor exit (inlet to the microchannels of the 2nd-vapor pass) \( x_v \) (0.07, 0.20, and 0.31) is close to respective inlet quality.

For all the tested conditions, quality at liquid exit is always at a lower value than its corresponding inlet quality, which indicates separation happens. As shown in Figure 5.10(a), the dash line is the equivalent-value line and all the data points are under it. The smallest difference between \( x_{in} \) and \( x_l \) is 29% at 25 g/s and inlet quality of 0.21. A more liquid-rich flow is obtainable at the liquid exit.

From mass conservation of vapor phase and liquid, quality at the vapor exit should always be higher than \( x_{in} \) for the same conditions, which is shown in Figure 5.10(b). Similar with liquid separation efficiency, quality at vapor exit is high at low mass flow rate (10 g/s) and high inlet quality (\( x_{in} > 0.15 \)). But at higher mass flow rate, due to the tendency of flow homogeneity, quality at vapor exit could drop dramatically, as shown in Figure 5.10(b).

It is not hard to conclude the same trend of liquid separation efficiency with quality at vapor exit. In fact, high liquid separation efficiency means little liquid going up, which gives a high quality at the vapor exit, vice versa. On the other hand, low quality at the liquid exit means little vapor going down, in which case the vapor separation efficiency is high. It is possible that quality at the vapor exit would become smaller and have less
dependence on inlet quality were at an even higher mass flow rate beyond the range of current test conditions.

Figure 5.10 (a) Liquid exit quality $x_l$ vs. Inlet quality $x_{in}$; (b) Vapor exit quality $x_v$ vs. Inlet quality $x_{in}$
5.3.2 Flow patterns inside the header

In an intermediate header, flow coming out of microchannels experiences sudden expansion, stagnation splashing onto the inner wall, direction conversion and reaction and mixing with adjacent flows, etc. It is highly complicate developing two-phase flow. From visualization, flows out of the microchannels are in forms of slug/intermittent flow or highly agitated two-phase flow jets. Two factors afterwards are making liquid to move upward: 1) splashing of the header wall gives to liquid initial upward velocity; 2) vapor going upward will entrain some of the liquid in forms of droplets or ligaments. This causes the generation of a recirculation region when the trajectory of liquid phase is not high enough to reach the top of the header.

Figure 5.11 shows flow regimes at different inlet mass flow rate. A more detailed summary of flow patterns inside the intermediate header can be found in Appendix C. At intermediate mass flow rate (16 g/s) of the tested conditions, recirculation region is a steady zone where the bulk of liquid is having a force balance to be steady at a certain height in the header. Ligaments/droplets with uncertain moving directions are spinning around in this region. For an intermediate/low mass flow rate, two-phase flow inside the header can be divided into three regions by visualization: mist, recirculation, and churn region. Mist region is on top of the recirculation region where only droplets exist as liquid phase. Some of the droplets go downward whereas the others go out through the vapor exit. Churn region is at the bottom where liquid jets from the inlet tubes are acting with each other after splashing onto the inner wall and the vapor jet impinges into the liquid surface at the liquid exit.
For a high load condition (30 g/s, \(x_{in}=0.25\)), liquid moving upward splashes on the top while vapor moving downward impinges into the bottom liquid pool. Recirculation region moves to the top and is even compressed because of high upward velocity of the liquid. Relatively big amount of liquid and vapor phase both go out through undesired exits, which makes separation efficiencies drop. Gas cavities become obvious because of the high momentum of the liquid jets coming out of the microchannels. Violent liquid interaction is happening on the “splashing side” of the wall.

![Diagram showing different flow regimes at various separation loads](image)

**Figure 5.11** Different flow regimes at various separation loads

As a summary from above, we conclude that there are two conditions are required to the form a recirculation zone. 1) Momentum after impinging onto the wall drives the
liquid go through the blockage from other liquid jet above it and continue move upward.

2) The drag force from the upward vapor is sufficient to hold up the liquid. This drag force could result from the exchange of the momentum between vapor and liquid, which is resisting the liquid from dropping down. Particularly, the first one may exacerbate the second as the churny mixture layer will block the flow area for the upward vapor flow, and thus increase the local velocity and the phase interaction.

5.4 SEPARATION EFFICIENCY WITH CHANGING DOWNSTREAM FLOW RESISTANCE

5.4.1 TRADE-OFF BETWEEN VAPOR SEPARATION EFFICIENCY AND LIQUID SEPARATION EFFICIENCY

When separation header is adopted in a real condenser, separation results for two-phase flow will change with different downstream tubes numbers and/or different heating loads for downstream passes resulting from various air conditions. These scenarios will result in different downstream flow resistance for the header, which will change the boundary condition for separation flow field thus changing separation results.

After initial tests with minimum downstream flow resistance, tests are conducted with varying flow resistance. In initial tests, a big portion of vapor is going down to the liquid pass. $\eta_v$ is as low as 57% at 25 g/s, $x_{in}$=0.21. In a real separation condenser, more vapor – the condensing phase – is favored for going into the upper vapor passes. Therefore, flow resistance on the path of liquid exit is adjusted by gradually closing the metering valve downstream the collecting tank for the liquid exit. Since vapor has higher friction factor
than liquid at the same level of mass flow rate, a larger portion of the impact should be on vapor than liquid. In other words, vapor separation efficiency could have a larger improvement than the detriment to the liquid separation efficiency. Downstream pressure drop ($\Delta P$) is measured between points 1 and 2 in Figure 5.6(b). The increment of $\Delta P$ should be proportional to flow resistance.

Figure 5.12 Separation efficiency vs. $\Delta P$

From Figure 5.12, it can be seen the inevitable drop for $\eta_l$ with the increase of $\eta_v$. However, the improvement of $\eta_v$ is percentage-wise larger than the decrease of $\eta_l$. For example, at 16 g/s, $\eta_v$ increases from 0.598 to 0.776, by 29.8% while $\eta_l$ decreases from 0.745 to 0.617, by 17.2%. It also can be drawn from Figure 5-10 the impact of mass flow rate on this methodology. For mass flow rate of 16 g/s, the improvement for separation is better. At the largest $\Delta P$ 95.1 kPa, $\eta_v$ (0.776) exceeds the initial value of $\eta_l$ (0.745) while $\eta_l$ (0.617) does not drop lower than the initial value of $\eta_v$ (0.598). On the other hand, for
mass flow rate of 30 g/s, at the largest \( \Delta P \) 169.4 kPa, \( \eta_v \) (0.751) exceeds the initial value of \( \eta_l \) (0.627) while \( \eta_l \) drops to 0.481, lower than the initial value 0.578 for \( \eta_v \).

As for an improved design in future work, the number of microchannel tubes at the top passes may be increased while that at bottom passes may be reduced, since the idea behind this design is that vapor may preferentially flow through the top pass with increasing the flow resistance at the lower passes relative to the top pass.

**5.4.2 Transition of Flow Pattern on Separation Efficiency**

When the header is used as a two-phase separator in a separation condenser, only vapor refrigerant from the top (vapor) exit should be supplied into the 2nd vapor pass since the high heat transfer coefficient of vapor phase is the core benefit of a separation condenser. At the same time no vapor should be carried out by the downward liquid flow to the 2nd liquid pass. Imperfect liquid separation (liquid entrainment in upward vapor flow) would reduce heating capacity and subcooling. For this reason, in experiments presented in this session the vapor separation efficiency was always maintained higher than 60% by adjusting the downstream valve. In this way, downstream flow resistance is changed and the liquid separation efficiency \( \eta_l \) is used as only index to evaluate the separator performance.

A churny and highly agitated two-phase mixture is formed which strongly recirculates right above the two-phase region. In recirculation region the entrained liquid is mainly in the form of bulk, churny ligament like structure. Recirculation region is higher with higher vapor upward velocity. Figure 5.13 shows the elevation of recirculation region when inlet mass flow rate changes from 10 g/s to 20 g/s at \( x_{in}=0.15 \).
Figure 5.13 Flow separation regime at fixed inlet quality 15% and varied mass flow rates

Figure 5.14 shows the separation of liquid when vapor is well separated. $\eta_v$ is kept high by adjusting the valve downstream the liquid exit tank in Figure 5.6. The dash line for $x_{in}=0.15$ can illustrate the effect of flow patterns on separation. At $\dot{m}=10$ g/s, recirculation region starts to be generated. Up until $\dot{m}=14$ g/s, where recirculation region is elevated right below the vapor exit, $\eta_l$ is maintained at a high value. The liquid separation efficiency depends on the quantity and size of liquid droplets present at the junction. Although the smaller droplets would be easier to be entrained, they carry insignificant liquid mass, and therefore the liquid separation efficiency is still relatively high and insensitive to inlet quality. As $\dot{m}$ is increased to 16 g/s, recirculation region blocks part of the vapor exit, $\eta_l$ starts to drop. As $\dot{m}$ is increased further to 20 g/s, under this operating condition, the recirculation zone reaches the top, sustained liquid in the
recirculation zone has nowhere to go but the vapor exit, which makes more liquid flow through the vapor exit and $\eta_l$ drops even further.

Figure 5.14 Liquid separation efficiency $\eta_l$ when $\eta_v > 60\%$

As it is shown that separation efficiency is very sensitive in transition from three regions to one region. For future header design, to avoid the transition of flow pattern and maintain relative good separation performance, the design options of header should provide conditions to keep the recirculation region below the vapor exit. In order to achieve this, enlarging the header inner diameter or elongating the header height may be a good option.

5.5 Effect of Upward Vapor Velocity

5.5.1 Deduction of the velocity inside the header
Vapor momentum is a crucial factor that influences the phase separation in header. After incoming liquid jet impinges on the vertical tube, it intends to divert in both upward and downward directions. On one hand, at lower zones of the header vapor coming out from microchannels impinges the liquid surface. Higher downward vapor momentum that is created at splashing makes $\eta_v$ lower. On the other hand, higher upward vapor momentum would have fierce momentum exchange with liquid to lift it up, thus lowering $\eta_l$.

In order to reveal the physics behind phase separation in the header, it is important to calculate the phase velocity inside. Typically, higher velocity causes much stronger liquid inertial force and the vapor phase drag force, both of which contributes to the liquid entrainment to the vapor exit as both of them are opposite to the downward gravity force. The liquid inertial momentum and the vapor phase drag force essentially determines the eventual quantity of liquid entrained to the vapor exit.

The deduction of velocity inside the header is based on the measurements of mass flow rate at each of the exits. Figure 5.15 shows the numbering of the microchannel tubes and the discretization of the header volume to calculate the velocity for each phase along the longitudinal direction of the header. Tube number is named starting from the bottom tube of the 21 inlet tubes to the top tube of the 11 vapor exit tubes. Figure 5.15(b) shows the discretization taking the top of the inlet microchannels as an example. Each element is the space between every two adjacent microchannels. In a single element, one phase, either liquid or vapor, is assumed to have a uniform velocity.
After discretization, mass flow rates of both phases in each element is calculated based the mass flow rate data at the two exits. Figure 5.15 shows the schematic of the calculation of mass flow rate for each phase inside the header. It is assumed that the mass flow rate of each phase at each exit comes uniformly from each inlet microchannel tube of that phase. For example, the vapor mass flow rate at the vapor exit comes uniformly
from each inlet microchannel tube. Upward direction is taken as positive direction for vapor mass flow rate while downward direction for liquid mass flow rate. The local mass flow rate for vapor is the accumulative of the upward flow rate deducted from $\dot{m}_{v,v}$ minus the accumulative of the downward flow rate based on $\dot{m}_{v,l}$, as shown in Figure 5.16. After getting mass flow rate, velocity will be calculated by dividing the mass flow rate by density and flow cross sectional area. Void fraction of cross sectional area is calculated from Chisholm (1973).

![Figure 5.16 Vapor uniformly coming out of each microchannel tube](image)

**5.5.2 VELOCITY PROFILE INSIDE THE HEADER UNDER DIFFERENT CONDITIONS**

The maximum upward vapor velocity inside a header exists at location 21 in Figure 5.16, since it is where the vapor flow from each channel accumulates together. An overview of the impact of maximum upward vapor velocity on separation efficiency is
first shown in Figure 5.17. With increasing upward vapor velocity, both $\eta_l$ and $\eta_v$ become smaller. Large upward velocity lifts up liquid and large downward vapor velocity deepens the bubble impingement into liquid. These two effects both mix phases and make separation efficiencies drop.

![Graph](image)

**Figure 5.17 Maximum upward vapor velocity inside the header vs. separation efficiency**

To depict the two-phase phenomenon more clearly, three representative locations in Figure 5.15(a) along the vertical direction have been chosen to describe the flow structure in header based on visualization. They are the boundary between inlet and liquid exit, the middle of the whole header, and the boundary between inlet and the vapor exit. From the bottom tube of the inlet to the top tube of the vapor exit, the 32 microchannel tubes are named by number. Driving forces for two-phase flow motion are analyzed based on different flow structures.

Separation is better when downward liquid and upward vapor flows can pass each other. Figure 5.18 shows the flow structures at low mass flow rate 10 g/s with changing
quality. At 10 g/s $x=0.05$, there is a liquid pool in the header. But just because only hydrostatic head exists on the liquid exit, $\eta_l$ is not desirable, while $\eta_v$ can be kept at a high value. When the inlet quality is tripled around to 0.14, the situation changes. Vapor amount (on volume) strongly increases, which adds pressure head on the liquid exit. On the other hand, liquid and vapor are still able to pass by. That is because the momentum that liquid flow carries after impinging onto the wall is relatively small and gravity is able to easily drag the denser liquid down. Liquid level is right above the liquid exit, ending up with both very high $\eta_v$ and $\eta_v$.

Figure 5.18 Flow structures and velocity profile at low mass flow rate 10 g/s
Figure 5.19 Velocity profile inside the header at $\dot{m}_\text{in}=10$ g/s, $x_\text{in}=0.14$

Figure 5.19 is showing the vapor velocity profiles in the header for the corresponding operating conditions in Figure 5.18. Vapor velocity has the positive direction of upward while liquid velocity has the positive direction of downward. The liquid mass flow rate at the vapor exit $\dot{m}_\text{lv}=0$, so the local liquid velocity is equal to zero. Local upward vapor velocity in this part is calculated from mass flow rate by assuming $\dot{m}_\text{lv}$ is uniformly...
distributed into each microchannel tube of the vapor exit. The highest upward vapor velocity occurs at tube 21 for each velocity profile. That is the boundary between vapor exit and the inlet, where all the vapor mass flow from 21 inlet tubes add up. Negative values mean that the vapor changes direction to exit through the liquid exit.

Figure 5.20 shows the video snap for the lower part of tubes. Liquid jet with a higher velocity (1.3 m/s) impinges the inner wall of the vertical tube. Some portion of the incoming liquid is initially diverted upward against gravity, but later it falls down again, which creates a liquid film recirculation region right above the junction. This is because the local upward vapor flow (0.12 m/s) is insufficient to carry over such liquid further towards the top outlet, against downward gravity force. Thus, in these cases the liquid separation is still relatively high, and especially insensitive to change of the refrigerant inlet mass flow rate and quality, as can be seen in Figure 5.14.
When mass flow rate is increased to 16 g/s while the inlet quality is kept about the same, the difference with $\dot{m}_{in}=10$ g/s is apparent. Some liquid starts to exit from the vapor exit, which makes $\eta$ drop.

The magnitude of velocity becomes bigger compared with $\dot{m}_{in}=10$ g/s. It can be seen from Figure 5.21 that the recirculation region elevates and it is the region where the liquid velocity changes direction. This also explains the bulk flow characteristic of the recirculation region. Because the force balance applied to the bulk liquid, the liquid velocity is not certain. From the visualization and calculation, the recirculation region is from tube 16 to tube 23, which is the boundary region between inlet the vapor exit.
Figure 5.21 Velocity profile and flow structures inside the header at $\dot{m}_{in}=16$ g/s, $x_{in}=0.15$

The presented model can predict the vertical location of recirculation region well based on mass flow rate data at the two exits and uniformly distribution assumption in the header. The uniform distribution assumption is more valid here than in combining header or dividing header because the separation header is essentially an intermediate header. 21 microchannel tubes keep a good uniformity of flow as the inlet because it has superheated
vapor coming in real application at the first header. The model can be used in the future to control the recirculation region under the vapor exit, thus keeping liquid separation efficiency high, by controlling the mass flow rate at the vapor exit and liquid exit, respectively.

5.6 SUMMARY AND CONCLUSIONS

This chapter presents experimental study of vapor-liquid refrigerant separation in a vertical second header for microchannel condenser. R134a is chosen as testing refrigerant, and inlet mass flow rate and quality are varied approximately from 8.4 g/s to 30 g/s and from 5% to 25% with intention to simulate operation in air-conditioning systems with cooling capacities in a range of about to 4-5 kW. A few conclusions can be drawn as follows:

Separation efficiency is analyzed by both quantitative and visual methods. For a typical MAC condenser, ideal separation can happen at low mass flux up to 70 kg/(m²s) and separation effect in the second header (D=15.8 mm) is small when mass flow rate is over 20 g/s, if not designed well. Separation is better when downward liquid and upward vapor flows can pass each other. Two factors make it harder for liquid to separate: 1) splashing of the header wall gives to liquid some upward momentum; 2) vapor going upward can add to liquid momentum. As mass flow increases velocities increase, especially when combined with increasing quality. Large upward velocity lifts up liquid and large downward vapor velocity deepens the bubble impingement into liquid. These two effects both mix phases and make separation efficiencies drop.
The upward momentum of liquid and the drag force from vapor to liquid work together to generate a recirculation region of liquid. The recirculation region is higher with higher vapor upward velocity. When it reaches the top end, flow becomes one region and separation deteriorates. A model with uniform distribution assumptions can predict the location of recirculation well, and thus may work as a guidance to control mass flow to achieve higher $\eta_l$.

Efforts need to be made either to reduce vapor and liquid interaction or to reduce the vapor momentum. Enlarging the header inner diameter or elongating the header height may be good options to improve separation. However, it may not be good for maintaining the mass flux at the top pass from the heat transfer point of view as has been illustrated in Chapter 4. A trade-off between good separation and high overall heat transfer exists. Thus, consideration must be taken comprehensively when designing the separation condenser.
REFERENCES


Cavallini, A., G. Censi, D. Del Col, L. Doretti, G. A. Longo, and L. Rossetto, Experimental investigation on condensation heat transfer and pressure drop of new HFC refrigerants (R134a,


APPENDIX A: EXPERIMENT SETUP OF THE MAC SYSTEM

In Appendix A, a detailed description of the other components that are used in the MAC system of Chapter 4 will be presented.

For the experimental study, two environmental chambers, wind tunnels, a compressor stand, an experimental mobile heat pump air conditioning direct system, and measuring instrumentations were used. In each chamber, there is a variable power electric heater on the floor, and an evaporator core underneath the ceiling which is connected to a chiller next to outdoor chamber. Temperature in each chamber can be manually controlled by adjusting heater input power and chiller operating parameters. Relative humidity can be controlled by injecting vapor through building vapor supply line. In the current study, effect of relative humidity on system operation is not a priority, and all the experiments in this thesis are carried out at dry condition. Air flow through heat exchangers are drawn by frequency controlled metal blade blowers assembled at the exit of the wind tunnels. Air flow rate is obtained by measuring temperature of air at the nozzle outlet and pressure drop across the nozzle. Chilled mirror dew point sensors are installed to measure dew point temperature at air inlet of evaporator and at the nozzle. With measurements from the wind tunnel, air side capacity of heat exchangers can be obtained.

An ACDelco shaft-driven semi-hermetic compressor as shown in Figure 4.3 is used as the compressor. The compressor consists of six cylinders, each of which has a displacement of about 25 cm³. The compressor is driven by an electric motor located
below the compressor stand. The motor is controlled by a variable frequency driver. A torque transducer and a tachometer are used to measure the torque on the shaft and compressor speed. The inner condenser has two air doors at the front and back sides of it, which are used to block air flowing in. Thus, in real tests the inner condenser provides nothing but 20-30 kPa pressure drop.

![ACDelco semi-hermetic compressor](from the internet)

Figure A.1 ACDelco semi-hermetic compressor (from the internet)

A Muller 1/2” refrigeration ball valve is used to bypass the outdoor EEV in cooling mode and is manually switched. A Parker model SER-A EEV is used to mimic the original orifice tubes as expansion devices. A picture of the electronic expansion valve is shown in Figure A.2. To setup the EEV opening size, the Parker Interface Board (IB) model IQ2 is used. As IB controller needs a 0~10 V input signal to represent EEV opening size of 0% to 100%, a voltage division circuit with output voltage from 1 to 10 V was designed and made for each IB controller. The voltage signal can be easily adjusted by a manual potentiometer.
Figure A.2 Electronic expansion valve (from the internet)

The photo and detailed geometries of the microchannel evaporator are shown in Figure A.3 and Table A.1, respectively. The tested evaporator consists of totally 58 parallel tubes with the face area of about 0.094 m² and total air-side surface area of about 2.1 m²; each tube is 196.9 mm long. All the inlet, intermittent, and outlet headers are rectangular in shape. The evaporator was placed near the inlet of the wind tunnel, with an angle of 7° leaning toward downstream of the air flow to help condensate drainage. A 4×4 thermocouple grid was made by spacing welded type T thermocouples evenly across the face area of the evaporator to measure the average evaporator air inlet temperature. Each row of thermocouples is attached to a fishing line, which was fixed at one end on one side of the wind tunnel inner wall, while the other end was hooked to a spring through a screw eye on the other side of the wind tunnel inner wall. This design allows to take off the thermocouple grid easily when the heat exchanger needs to be moved or replaced. The evaporator air outlet temperature is measured by a 3×3 thermocouple grid.
Table A.1 Microchannel evaporator geometry

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<td>Number of tubes of fourth pass</td>
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The transparent accumulator shown in Figure A.4 works as a liquid vapor separator before compressor suction. The transparent wall allows easy observation of change in system active refrigerant charge amount. A ruler is attached to outside of the accumulator for liquid level measurement.
Figure A.4 Transparent accumulator (Feng and Hrnjak, 2015)
APPENDIX B: ZERO STABILITY OF MASS FLOW METERS USED IN CHAPTER 5

The mass flow rate meters used in Chapter 5 are Micro Motion ELITE series. The RFT9739 transmitter works with Micro Motion sensors to provide precision fluid measurement in a wide variety of fluid applications. The RFT9739 has modular, microprocessor-based electronics, incorporating ASIC digital technology with a choice of digital communication protocols. Combined with a Micro Motion sensor, the RFT9739 provides accurate mass flow, density, temperature, and volumetric measurements of process fluids.

The model and zero stability value used in Chapter 5 is boxed out in Figure B.1.
Zero stabilities

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<sup>(1)</sup> Based on standard temperature and pressure conditions of water at 20 to 25 °C and 1 to 2 bar

Figure B.1 Zero stability of mass flow rate meters used in the tests
APPENDIX C: FLOW PATTERN MAP FOR TESTS WITH MINIMUM DOWNSTREAM FLOW RESISTANCE

In Appendix C, flow structures are visualized with minimum downstream flow resistance (downstream valves of the separation header are fully open) in order to study the flow patterns. Results under various test conditions are shown in a test matrix categorized by using inlet mass flow rate as columns and inlet quality as lines. Flow photographs from high speed camera are shown together with separation efficiency $\eta_\text{l}$ and $\eta_\text{v}$. Accumulative bulk vapor velocity $v_\text{v}$ and bulk liquid velocity $v_\text{l}$ are also present with each operating condition. $v_\text{v}$ is at the upper boundary between inlet tubes and vapor-exit tubes. $v_\text{l}$ is at the lower boundary between inlet tubes and liquid-exit tubes. $v_\text{v}$ has the positive direction of upward while $v_\text{l}$ downward.

Generally, from visualization and measurements, good separation is happening in the area of low mass flow rate (under 14 g/s) and high quality (over 15%) in the test matrix. At mass flow rate over 20 g/s, separation deteriorates regardless of the inlet quality, because the liquid upward velocity and vapor upward velocity have been both increased.
Figure C.1 Flow pattern map for tests with minimum downstream flow resistance

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<th>m=10 g/s</th>
<th>m=14 g/s</th>
<th>m=16 g/s</th>
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<td>η_l=86.1%</td>
<td>η_r=84.3%</td>
<td>η_l=75.9%</td>
<td>η_r=65.1%</td>
<td>η_l=60.4%</td>
<td>η_r=60.4%</td>
</tr>
<tr>
<td>η_s=83.5%</td>
<td>η_r=82.4%</td>
<td>η_l=76.0%</td>
<td>η_r=73.6%</td>
<td>η_l=66.9%</td>
<td>η_r=60.8%</td>
<td>η_l=59.1%</td>
<td>η_r=59.1%</td>
</tr>
<tr>
<td>v_l=0.32 m/s</td>
<td>v_r=0.37 m/s</td>
<td>v_l=0.45 m/s</td>
<td>v_r=0.52 m/s</td>
<td>v_l=0.58 m/s</td>
<td>v_r=0.72 m/s</td>
<td>v_l=0.84 m/s</td>
<td>v_r=0.84 m/s</td>
</tr>
<tr>
<td>v_1=0.08 m/s</td>
<td>v_2=0.11 m/s</td>
<td>v_1=0.22 m/s</td>
<td>v_2=0.27 m/s</td>
<td>v_1=0.37 m/s</td>
<td>v_2=0.45 m/s</td>
<td>v_1=0.53 m/s</td>
<td>v_2=0.53 m/s</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>x=0.15</th>
<th>m=8.4 g/s</th>
<th>m=10 g/s</th>
<th>m=14 g/s</th>
<th>m=16 g/s</th>
<th>m=20 g/s</th>
<th>m=25 g/s</th>
<th>m=30 g/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>η_l=94.3%</td>
<td>η_r=100%</td>
<td>η_l=94.5%</td>
<td>η_r=84.8%</td>
<td>η_l=70.1%</td>
<td>η_r=67.5%</td>
<td>η_l=61.3%</td>
<td>η_r=61.3%</td>
</tr>
<tr>
<td>η_s=78.8%</td>
<td>η_r=83.1%</td>
<td>η_l=70.2%</td>
<td>η_r=64.5%</td>
<td>η_l=60.9%</td>
<td>η_r=60.3%</td>
<td>η_l=58.1%</td>
<td>η_r=58.1%</td>
</tr>
<tr>
<td>v_l=0.40 m/s</td>
<td>v_r=0.44 m/s</td>
<td>v_l=0.59 m/s</td>
<td>v_r=0.65 m/s</td>
<td>v_l=0.83 m/s</td>
<td>v_r=0.99 m/s</td>
<td>v_l=1.14 m/s</td>
<td>v_r=1.14 m/s</td>
</tr>
<tr>
<td>v_1=0.16 m/s</td>
<td>v_2=0.15 m/s</td>
<td>v_1=0.33 m/s</td>
<td>v_2=0.37 m/s</td>
<td>v_1=0.44 m/s</td>
<td>v_2=0.54 m/s</td>
<td>v_1=0.62 m/s</td>
<td>v_2=0.62 m/s</td>
</tr>
</tbody>
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