INVESTIGATIONS ON TWO-PHASE HEAT EXCHANGER FOR MIXED REFRIGERANT JOULE-THOMSON CRYOCOOLER

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ABSTRACT

The design of the recuperative heat exchanger used to pre-cool the refrigerant mixture prior to J-T expansion is crucial for the efficient operation of the mixed refrigerant Joule-Thomson (MR J-T) cryocooler. The multi-component non-azeotropic refrigerant mixture undergoes boiling and condensation heat transfer simultaneously in the heat exchanger. Therefore, it is important to analyze the performance of the heat exchanger in terms of temperature distribution with respect to the mixture of gases used.

In the present work, temperature measurements are carried out at the ends of the heat exchanger for high pressure stream, while eight sensors are installed at equal distance along the length of heat exchanger to measure temperature of low pressure stream. The paper reports variation in heat transfer coefficient along the length of the heat exchanger. The variation is discussed with respect to temperature distribution across the length and changes in thermo-physical properties of the gas mixture.

KEYWORDS: Cryocooler, Mixed gases, Recuperative heat exchanger.

INTRODUCTION

Joule-Thomson cryocoolers using mixed refrigerants show a lot of promise in producing low temperatures due to their efficiency, simple construction, operational reliability, fast cool-down time and no electromagnetic interference. By using specific mixtures, these cryocoolers can efficiently operate in the cooling temperature range from 80-230 K in many applications, such as cooling infrared sensors, gas chiller or liquefaction,
cryosurgery, cryo-preservation, etc. However, overall performance of the cryocooler is
governed by the selection of the mixture composition and the heat exchanger used.

Based on the studies carried out on the MR J-T cryocooler development at IIT
Bombay [1], it has been found that heat exchanger plays a very crucial role in determining
the performance of cryocooler. Experiments are being carried out to study the heat
exchanger performance with respect to mixture composition and operating parameters.
Many experimental/numerical studies that have been carried out by various workers on MR
J-T cryocooler are mainly related to the optimization of mixtures used and the
thermodynamic performance of overall refrigeration system [1-3]. The gas mixture in such
a system undergoes boiling and condensation heat transfer in the heat exchanger, which is
responsible for the high efficiency of J-T coolers. The design of the recuperative heat
exchanger used to pre-cool the refrigerant mixture prior to J-T expansion is most crucial in
the efficient operation of the cryocooler. However, little has been published about the heat
transfer characteristics of the tubes-in-tube helical coil heat exchanger operating with
mixed refrigerants. It may be due to the insufficient knowledge about the hydrodynamics
and heat transfer, as applied to liquid-vapor flows of multi-component working fluids.
Recently, Nellis G. et al. [4] described an experimental test facility and procedure that has
been used for the measurements of horizontal, flow boiling heat transfer coefficient for
optimal nitrogen-hydrocarbon mixtures over a range of compositions, temperatures, mass
flow rates, and pressures that are applicable to small scale J-T cryocoolers. Gong M.Q., et
al. [5] presented experimental results for pressure drop, temperature distributions at
different operating conditions and heat transfer characteristics of the heat exchangers.

The objective of the present study is to measure the temperature distribution in the
heat exchanger and to analyze its performance with respect to the mixture of gases used in
the system.

**EXPERIMENTAL SET-UP**

The experimental set-up consists of compressor, after-cooler, oil filters, heat
exchanger, capillary tube and evaporator as shown in Figure 1. The heat exchanger used is
the multi-tubes-in tube type helical coil heat exchanger. The dimensions of the heat
exchanger are as given in Table 1. The total length of the heat exchanger is kept as 6.75 m.
Temperatures and pressures of the working fluid are measured at suction and discharge line
of the compressor. In order to measure the temperatures along the length of the heat
exchanger, the heat exchanger may be divided into segments as reported by Gong et al. [5]
and temperature sensors may be installed at the joints of these segments. However, this
may lead to disturbing the flow pattern and associated two-phase heat transfer
characteristics in the heat exchanger.

In the present work a total of eight sensors (T1 to T8) starting from inlet to outlet of
cold fluid are installed on the outside of the outer tube through which low pressure
refrigerant mixture is circulated. Temperature measurement of high pressure stream
circulating through the inner tubes of the heat exchanger is challenging due to difficulty in
integrating the temperature sensors with the flow stream. Therefore, the hot stream
temperatures are measured at the inlet (T9) and outlet (T10) to heat exchanger only and
intermediate temperatures on high pressure side are computed using energy balance.
Temperatures at all the locations are measured with the help of PT 100 sensors calibrated
at liquid nitrogen temperature. Temperature data is recorded using the data logging system,
Data taker-800. A rotameter is installed on the suction line to measure the volume flow rate
of refrigerant at steady state condition. Pressure of the low and high pressure stream in the
Figure 1. The schematic of experimental set-up

TABLE 1. Specifications of the heat exchanger

<table>
<thead>
<tr>
<th>Tube</th>
<th>ID (mm)</th>
<th>OD (mm)</th>
<th>No. of tubes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer</td>
<td>10.7</td>
<td>12.5</td>
<td>01</td>
</tr>
<tr>
<td>Inner</td>
<td>02</td>
<td>03</td>
<td>07</td>
</tr>
</tbody>
</table>

heat exchanger is measured with the help of pressure gauges installed at the inlet and the outlet of heat exchanger. All the thermodynamic properties are calculated with Peng-Robinson equation of state [6] using aspenONE [7]. The mass flow rate of refrigerant mixture is calculated using density of the mixture in circulation at pressure and temperature to the inlet of rotameter. The composition of mixture in circulation is measured for each of the mixtures using gas chromatograph (Make: PerkinElmer - Clarus 500).

THEORETICAL ANALYSIS

In order to investigate the influence of mixture composition on the temperature distribution in the counter flow heat exchanger, the variation of two-phase specific heats of high pressure and low pressure stream is studied. The thermodynamic properties of the mixture depend on the pressure of the fluid. Therefore, the heat capacities of the two streams flowing at different pressures through the counter-flow heat exchanger do not match. The two-phase specific heat capacity of low pressure flow of non-azeotropic mixtures may be larger than that of high pressure flow because of contribution due to latent heat.

The enthalpy in two-phase state, $h_{tp}$, can be calculated using Equation 1 [8].

$$h_{tp} = x h_g + (1-x) h_l$$  \hspace{1cm} (1)

where $x$ is vapor fraction; $h_g$ and $h_l$ are gas and liquid phase enthalpy respectively.

Two-phase specific heat, $C_{p,tp}$, is given by Equation 2.

$$C_{p,tp} = \left( \frac{\partial h_{tp}}{\partial T} \right)_p$$  \hspace{1cm} (2)
Therefore,

\[ C_{p,p} = \left( \frac{\partial h}{\partial T} \right)_{p} h_e + xC_{p,g} + (1-x)C_{p,l} \]  

where \( C_{p,l} = \left( \frac{\partial h}{\partial T} \right)_{p} \), \( C_{p,g} = \left( \frac{\partial h}{\partial T} \right)_{p} \), \( h_e = (h_g - h_l) \)

To analyze the relative variation in specific heat capacity of the two fluids in the heat exchanger, a ratio \( c \) of specific heat capacity rates is defined as given in Equation 4.

\[ c = \frac{\dot{m}C_{p,p}^e}{\dot{m}C_{p,p}^c} \]  

where \( \dot{m} \) is the mass flow rate of the refrigerant mixture circulating through system.

The heat exchanger is divided into seven parts to install sensors on outside surface of the outer tube for temperature measurements of the cold fluid and to determine temperatures of the hot fluid theoretically. Each part between the two successive locations of the temperature measurement is treated as one segment of the heat exchanger. In order to find out the temperature distribution for the high pressure stream flowing through the inner tubes, energy balance for each segment is done. Knowing the measured value of the temperature of hot fluid at the inlet or outlet, the intermediate temperatures of the hot fluid are calculated at the corresponding locations where corresponding temperatures of the cold fluid are measured. The following assumptions are made in the analysis carried out:

1. The axial conduction and radiation losses in the heat exchanger are neglected.
2. There is negligible pressure drop in both high pressure and low pressure streams of the heat exchanger.
3. The mixture composition in circulation at steady state is same throughout the heat exchanger.

The overall heat transfer coefficient (HTC) for each segment of the heat exchanger is calculated using apparent log mean temperature difference (ALMTD) and heat transferred \( q \) in each segment. The average LMTD for the heat exchanger is defined as given in Equation 5.

\[ LMTD_{avg} = \frac{\sum q_i}{\sum \left( \frac{q_i}{ALMTD_i} \right)} = \frac{hc_2 - hc_1}{\sum \left( \frac{q_i}{ALMTD_i} \right)} \]  

where \( hc_2 \) and \( hc_1 \) is the enthalpy of the cold fluid at the outlet and inlet to the heat exchanger. The overall HTC for whole heat exchanger of surface area A is calculated with the help of Equation 6.

\[ HTC_{overall} = \frac{Q}{A(LMTD)_{avg}} = \frac{hc_2 - hc_1}{A(LMTD)_{avg}} \]
RESULTS AND DISCUSSIONS

Experiments are conducted with three different mixtures to study the effect on the performance of the heat exchanger. The different mixtures and the operating conditions like mass flow rates, pressure of the cold and hot fluid, are given in Table 2. The Mix#1[5] is such that it has lower percentage of low boiling point components (N₂/CH₄) and higher composition of middle boiling point component (i.e. C₂H₆) than that of Mix#2 and Mix#3. Mix#2 and Mix#3 are selected such as these consist of nearly the same composition of middle boiling point component, while Mix#2 is optimized to get lower refrigeration temperature. The no load refrigeration temperature, \( T_{\text{low}} \) obtained is 125.45 K and 110.95 K for Mix#1 and Mix#3 respectively whereas it is lowest for Mix#2 i.e. 98 K.

Figure 2 shows the temperature measurements along the length of heat exchanger for Mix#2 with respect to time. It is observed from the figure that the time required for attaining steady state temperature increases from locations at the cold end towards locations at the hot end of the heat exchanger. The cool-down time to reach the lowest temperature for the cryocooler is about 100 minutes, however, the time required for steady state behavior of the whole heat exchanger is more than 125 minutes. It is also observed that the rate of change of temperature at the locations nearer to the hot end is less than that at the cold end. This is attributed to the single phase heat transfer towards hot end of the heat exchanger.

The steady state temperature profiles for the hot and cold fluid for the Mix#1 are shown in Figure 3. The temperature difference between the two streams decreases towards the cold end of the heat exchanger for Mix#1. The maximum temperature difference between the two streams is 21.7 K. The high pressure stream enters the heat exchanger in gas state and leaves it in the liquid state since the bubble point and dew point temperatures

| Table 2. Mixture specifications and operating conditions |
|------------------|-----------------|-----------------|-----------------|-----------------|
| Mixture Components Charged/circulation Composition (%) | Mass flow rate (g/s) | \( P_L/P_H \) (bar) | \( T_{\text{low}} \) (K) |
| Mix#1 N₂/CH₄/C₂H₆/C₃H₈/iC₄H₁₀ | 5.01/38.19/27.31/13.97/15.52 | 4.41 | 4.6/13.6 | 125.4 |
| Mix#2 N₂/CH₄/C₂H₆/C₃H₈/iC₄H₁₀ | 25.5/21.3/14.2/20.5/18.5 | 3.81 | 3.8/14.2 | 98.0 |
| 40.105/18.43/13.10/17.86/10.49 | | | | |
| Mix#3 N₂/CH₄/C₂H₆/C₃H₈/iC₄H₁₀ | 15/34/14.5/19.5/17 | 4.02 | 3.9/13.5 | 110.9 |
| 18.80/33.44/15.39/18.14/14.23 | | | | |

FIGURE 2. Steady state behavior of the heat exchanger for Mix#2
for the hot fluid are 136.7 K and 280 K respectively. The low pressure stream changes its state from two-phase mixture to gas state since its bubble point and dew point temperatures are 108.48 K and 251.68 K respectively.

Figure 4 shows the variation in overall heat transfer coefficient, heat transfer rate and ALMTD for the segments along the length of heat exchanger. It is observed from Figure 4 that the overall HTC is maximum, i.e., 1095.22 W/m²K, in the two-phase region of the heat exchanger. On the other hand, the overall HTC is minimum at the hot end because both the streams are in gas state at the hot end.

Figure 5 and Figure 6 show temperature distributions and variation of HTC, heat transfer rate & ALMTD in the heat exchanger for Mix#2 respectively. It can be noted from the figure that the temperature difference between the two streams is maximum near the hot end of the heat exchanger and then decreases towards the cold end. The dew point and bubble temperatures for the hot stream are 282.6 K and 104.48 K respectively whereas for the cold stream, it is 248.72 K and 86.59 K respectively. The hot stream changes its phase from gas to two-phase while return cold stream changes from two-phase to gas state.

It is seen from Figures 5 and 6 that the temperature difference between the two streams for Mix#2 below 200 K remains nearly uniform up to the cold end of the heat exchanger. It is also observed from Figure 6 that the overall HTC near the hot end for the Mix#2 is less because of the low values of inside and outside HTC associated with the gas phase similar to that of Mix#1. However, the overall HTC towards cold region of the heat exchanger remains uniform compared to that for Mix#1 since both the streams are in two-phase up to 5 m length of the heat exchanger for Mix#2.
The temperature distribution for the Mix#3 is shown in Figure 7. The temperature difference for Mix#3 is more in the region near the hot end as compared to that for other mixtures. This is because the composition of high boiling point components is more in circulation for Mix#3 compared to that of Mix#1 and Mix#2. Mix#3 is such that the dew point temperature for hot and cold stream is 288 K and 254.56 K respectively whereas bubble point temperature for hot and cold stream is 114.4 K and 92.17 K respectively. In the region near the hot end both the streams are in gaseous phase while towards cold end there is two-phase region for Mix#3 similar to that for Mix#2. The temperature of the hot stream at the outlet is 120 K as compared to the computed value of 118.6 K. This indicates that the approach of applying energy balance in each segment can be used to predict temperature profile of the hot fluid because of the high effectiveness of the heat exchanger.

The variation in overall HTC along with the ALMTD and heat transfer rate for the Mix#3 is shown in Figure 8. It is found that the overall HTC does not remain uniform in the cold section of the heat exchanger for Mix#3 similar to Mix#2 since the heat transfer rate towards cold end is decreasing as against Mix#2. The average overall HTC, LMTD and heat transfer rate in the heat exchanger for the mixtures studied is compared in Table 3. It is clear that the overall HTC for Mix#1 is 718.79 W/m²K which is greater than that for Mix#2 and Mix#3 because of high rates of heat transfer in spite of higher value of LMTD for Mix#1. It is also seen from Table 3 that the average temperature difference for the Mix#2 is lowest as compared to the other two mixtures. From thermodynamic point of view, it is desirable to have minimum temperature difference between two fluids in the heat exchanger. Therefore, the no load refrigeration temperature for the Mix#2 is the lowest compared to two other mixtures.

From the above results, it is clear that the temperature distributions in the heat exchanger are strongly dependent on the operating conditions and the mixture composition. The overall heat transfer coefficient is also sensitive to the mixture and its composition used. This is due to the significant variation in thermo-physical properties of the mixture with respect to its composition. The variation of specific heat capacity rate ratio with respect to temperature for the three mixtures is plotted as shown in Figure 9. It is noted that the ratio of specific heat capacity of the two streams for all the mixtures is nearly equal.

**TABLE 3.** Overall heat transfer coefficients and LMTD

<table>
<thead>
<tr>
<th>Mixture</th>
<th>Overall HTC (W/m²K)</th>
<th>Average LMTD (K)</th>
<th>Q (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mix#1[5]</td>
<td>718.79</td>
<td>15.01</td>
<td>3128.76</td>
</tr>
<tr>
<td>Mix#2</td>
<td>660.05</td>
<td>11.88</td>
<td>2273.74</td>
</tr>
<tr>
<td>Mix#3</td>
<td>690.47</td>
<td>13.64</td>
<td>2731.69</td>
</tr>
</tbody>
</table>

**FIGURE 7.** Temperature distribution in the heat exchanger for mix#3.

**FIGURE 8.** Variation of HTC, Heat transfer rate and LMTD for mix#3.
to 1 in the hot section of the heat exchanger up to approximately dew point temperature of the hot stream. The ratio decreases from the maximum value greater than one up to the dew point temperature of the cold stream and then suddenly falls to a minimum value lower than one. This indicates that the hot stream is in two-phase while cold stream is still in gas phase in the heat exchanger. In the section of heat exchanger where both the streams are in two phase, the specific heat capacity of cold stream is greater than that of hot stream. In the cold section of the heat exchanger for both Mix#2 and Mix#3, specific heat of the hot stream is more than that of cold stream at the bubble point temperature of hot stream. Thus, it is clear that the specific heat capacity of cold stream may be greater or lower than that of hot stream depending on the state of these streams in the heat exchanger.

CONCLUSIONS

The two-phase heat transfer characteristics in the recuperative heat exchanger used in the MR J-T cryocooler are investigated. The effect of the mixture’s state and its composition on the steady state performance behavior of the heat exchanger is analyzed. The overall heat transfer coefficient for the Mix#1 is more compared to that for Mix#2 and Mix#3 even though the average LMTD is more for Mix#1. However the Mix#2 can produce the refrigeration temperature lower than that Mix#1 and Mix#3. It is observed that the overall heat transfer coefficients and temperature distributions are strongly influenced by the mixture composition, operating conditions and thereby, the mixture properties.

REFERENCES

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