STUDY OF BEHAVIOR IN THE HEAT EXCHANGER OF A MIXED GAS JOULE-THOMSON COOLER

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ABSTRACT

The object of the investigation is a mixed gas Joule-Thomson (J-T) cooler. A computational model was developed, which makes it possible to investigate the steady state behavior of the refrigerant in the heat exchanger of a mixed gas J-T system. The calculations show that the temperature distribution as well as the pressure distribution in the heat exchanger channels depends to a large degree on the pinch point locations in the process, which is strongly influenced by the mixture composition. This new understanding gives valuable information for the practical and optimal design of the heat exchanger as well as for the choice of the mixture composition.

INTRODUCTION

A mixed gas Joule-Thomson cooler has distinct advantages over other types of cryogenic refrigerators. The system has acceptable efficiency and the design is flexible for the interface with the cooled object. Based on the use of oil lubricated hermetic compressors this cooler combines such important benefits as reliability and “low cost”. It can replace presently used systems (for example Gifford-McMahon-refrigerators) and be used for old and new applications in the temperature range between 70 and 150 K.

The thermodynamics associated with the mixed gas J-T cycle are discussed in the literature\textsuperscript{1-3}. A more detailed description is given by Alexeev\textsuperscript{4}. Figures 1 and 2 show the flow diagram and T-h diagrams of J-T cycles with a typical nitrogen-hydrocarbon mixture and with pure nitrogen for typical working pressures. The following details are worth noting: the heat exchanger duty in a mixed gas cycle is approximately three times larger than in the nitrogen cycle; the average temperature difference in a mixed gas cycle is smaller than in the nitrogen cycle and the NTU is more than 50. Because the heat transfer coefficient in a mixed gas
exchanger is not higher than in a J-T system with pure nitrogen, a considerably larger heat exchange surface is necessary to realize a comparable mixed gas system.

The special degree of freedom of the mixed gas systems is that it is possible to influence the temperature profiles in the heat exchanger passages by a variation of the mixture composition. This is an important option, which is not available in systems with a single refrigerant.

In the development of an effective mixed gas system a solution for the following well-known contradiction has to be found: from a thermodynamic point of view an optimal mixture leads to minimal temperature differences in the heat exchanger. From a heat exchanger point of view, to transfer the heat at small temperature differences requires a larger surface and a higher pressure drop. This will lead to additional losses and to a degradation in cooler performance. Consequently an optimization of cycle parameters is necessary to realize the maximum potential of the mixed gas J-T-technology in a real system.

A number of optimization methods are available in the literature\(^1\)\(^-\)\(^3\). These methods allow one to optimize the mixture composition and the working pressures by assuming minimal temperature differences and pressure drops in the heat exchanger and theoretical compressor volumetric and power efficiencies. The value of the calculated results depends mostly on the accuracy of the assumed input data. Because the compressor performances are normally known (or can be evaluated relatively simply), the main problem is a correct assumption of the heat exchanger parameters: minimal temperature difference and pressure drops as well as temperature and pressure distribution.

In principle if these values could be predicted, a numerical model of the heat exchanger could extend prior optimization methods. However, the problem is that generalized correlations for heat transfer by condensation and vaporization of the multi-component mixtures, as well as predictions of the pressure drop for two-phase flow of multi-component mixtures in narrow channels with the necessary accuracy, are not available at the present time.

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**Figure 1.** Simple mixed gas Joule-Thomson cycle.

**Figure 2.** T-h diagrams of J-T cycles (\(T_{\text{amb}}=297\) K, \(\Delta T_{\text{min}}=3\) K, \(T_o=80\) K, low pressure \(LP=0.1\) MPa) with a mixture of \(N_2\) (30 mol), \(CH_4\) (23 mol), \(C_2H_6\) (17 mol), \(C_3H_8\) (10 mol), \(i-C_4H_{10}\) (20 mol), high pressure \(HP=1.8\) MPa and with pure nitrogen, high pressure \(HP=20\) MPa.
The goal of the present work was to obtain a deeper insight into the behavior of the mixed gas heat exchanger and to make further work in this field more efficient.

BACKGROUND

Very little information concerning the optimization of mixed gas JT-systems including simulation of the heat exchanger is available: Landa et al reported such a method\textsuperscript{5} and another contribution was made by Boiarski et al\textsuperscript{6}.

In the paper by Landa an optimization of a mixed gas system based on an oil-free multi-stage compressor with a Hampson-type heat exchanger is described. The parameter optimized was the cool-down time. It was assumed that the heat transfer is limited by the heat transfer on the low-pressure side and the liquid fraction in the low-pressure stream is small; therefore, the main heat transfer mechanism is not vaporization, but convection. Consequently, Landa could use the known correlation for forced convection to calculate the heat transfer. This model did not include any correlations for pressure drop in the heat exchanger. It was not needed because of the special properties of the chosen system, namely the pressure drop on the low pressure side is minimal and the pressure after the compressor is relatively high, therefore the pressure drop on the high pressure side does not significantly affect the performance of the system.

In the paper by Boiarski the first attempt was made to develop a complete optimization method for small mixed gas J-T coolers based on a single stage compressor. Some results of this optimization were presented. For such systems the pressure drop on the low-pressure side as well as on the high-pressure side plays an essential role. Therefore the correlation for pressure drops in the heat exchanger is required. These equations, as well as equations for heat transfer used are semi-empirical correlations, based on the information obtained from the testing of real coolers. Because the test set up allowed the measurement of temperature and pressure values at the ends of the heat exchanger, only average data could be evaluated. The use of average data simplifies the calculation. But on the other hand, the use of average coefficients has some disadvantages; for example, it does not give any information concerning the temperature and pressure drop distribution. It is not possible to generalize these data and to use them for other kinds and sizes of heat exchangers. Therefore the authors of this publication tried to combine these average data with simple theoretical models (based on the assumption of a homogeneous structure of the flow) for predictions of local heat transfer coefficients and pressure drops. Although, in general, this methodology is questionable, because such simplified models cannot describe correctly the complexity of the two-phase heat exchange and fluid mechanics, this procedure can nevertheless be useful for some problems with limited complexity.

SIMULATION MODEL

Several types of the heat exchangers can be used in mixed gas Joule-Thomson coolers. In the present work the double pipe heat exchanger and the multi-tube heat exchanger are discussed. A double pipe heat exchanger consists of two concentric tubes connected by end closures. The high-pressure flow (warm side) is through the inner pipe, and the low-pressure flow (cold side) is through the annulus formed by the inner and outer pipe. The outer tube is designated as the shell. The multi-tube heat exchangers are similar in construction to the double pipe heat exchangers, except that the inner pipe is replaced with a bundle of tubes. Although these heat exchangers are not compact and have a relatively high-pressure drop on the shell side, they are often used in mixed gas systems, because their manufacture is simple. Further advantages are low longitudinal heat conduction and a uniform flow distribution on the shell side.

A model for the simulation of the behavior of a heat exchanger in a mixed gas J-T system has been developed. It is based on the following assumptions:
• Steady-state operation
• Pressure losses in the lines between the compressor and the J-T-stage are neglected
• Pressure drop in the evaporator is neglected
• Mixture composition does not change in the cycle
• Uniform flow distribution on the tube, as well as on the shell side of heat exchanger
• No heat leak from ambient
• Longitudinal heat conductivity is neglected

Some of these assumptions are actually incompatible. For example, for the thermodynamic calculation we assume that the mixture composition does not change in the cycle, although the calculation method for the pressure drop in two-phase flow assumes slip, and consequently a change of the local mixture composition in the heat exchanger. In the present work this effect is neglected. Later study of this effect is needed.

The input parameters are the following:
• Ambient temperature (corresponds to the temperature of the warm end of the heat exchanger, high pressure stream)
• Pressures before and after the compressor
• Data for compressor volumetric and power efficiency
• Mixture composition
• Geometric configuration of the heat exchanger, including shell inside diameter, tube outside diameter, tube inside diameter, number of tubes in the bundle and heat exchanger length

The output parameters are the following:
• Mixture flow rate
• Cooling capacity of the cooler
• Cooling temperature
• Minimal temperature difference and temperature distribution in the heat exchanger
• Pressure drops and pressure distribution in the heat exchanger
• Distribution of the heat transfer coefficients in the heat exchanger

For a prediction of the thermodynamic properties of mixtures a Peng-Robinson equation of state was used. We used the TRAPP-method to predict viscosity and thermal conductivity for the mixtures. The surface tension for hydrocarbon liquids is estimated with the procedure 10 A3.1 described by API. The Bell-Delaware method was used to compute the pressure drops. The method was adjusted by introducing correction factors estimated from available experimental data. A modified method developed by Chen was used to calculate the heat transfer coefficients. This method includes correlations for forced convection as well as for vaporization and condensation of mixtures. The software program HEXTRAN was used for the calculation.

The calculation results were not compared with experimental data, except for the pressure drop at the shell side. The verification of calculations will be the next step in this investigation. However, the analysis of the calculated data does not show essential discrepancies in the behavior of the real systems and calculated values. This is the reason why it is plausible to use these data to study basic relationships between the heat exchanger parameters in a qualitative manner.

**DISCUSSION OF RESULTS**

A series of calculations for different mixtures were made. Each sequence included the calculation of the cooling capacity and optimal cooling temperature of the cooler for a set of fixed input parameters for different heat exchanger lengths. The combination of sequences allows one to compare the calculation results for different mixtures.
The simulation shows that the behavior in the heat exchanger initially depends on the mixture composition. For some mixtures this behavior is very similar, so one can use this fact to sort all mixtures into groups according to the kind of temperature distribution in the heat exchanger.

The following examples illustrate this consideration. They were evaluated for the following three nitrogen-hydrocarbon mixtures:

<table>
<thead>
<tr>
<th>components</th>
<th>mixture # 1</th>
<th>mixture # 2</th>
<th>mixture # 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>nitrogen:</td>
<td>30 mol %</td>
<td>27.27 mol %</td>
<td>33 mol %</td>
</tr>
<tr>
<td>methane:</td>
<td>23 mol %</td>
<td>20.91 mol %</td>
<td>18 mol %</td>
</tr>
<tr>
<td>ethane:</td>
<td>17 mol %</td>
<td>15.45 mol %</td>
<td>12 mol %</td>
</tr>
<tr>
<td>propane:</td>
<td>10 mol %</td>
<td>13.64 mol %</td>
<td>12 mol %</td>
</tr>
<tr>
<td>i-butane:</td>
<td>20 mol %</td>
<td>22.73 mol %</td>
<td>25 mol %</td>
</tr>
</tbody>
</table>

for following parameters
- Ambient temperature: 297 K
- Pressure after the compressor: 1.5 MPa
- Pressure before the compressor: 0.15 MPa
- Mixture flow rate: 2.1 standard m$^3$/h
- Shell inside diameter: 10 mm
- Tube outside diameter: 3 mm
- Tube inside diameter: 2 mm
- Number of tubes in bundle: 7

Figure 3 shows the different temperature distribution curves for a system with a heat exchanger length of 6 m.

The temperature distribution in the heat exchanger depends on the location of the pinch points in the thermodynamic process. For example, mixture # 1 contains a reduced fraction of high-boiling components (propane and butane), the pinch point is located at the warm end of the heat exchanger, and the temperature curve is very flat at the warm end. Since mixture # 2 consists of an increased fraction of high-boiling components, the pinch point is located at the cold end of the heat exchanger, and the flat part of the curve also locates at the cold end. Mixture # 3 consists of an increased fraction of high-boiling as well as low-boiling components (nitrogen and methane), the pinch point is located in the middle, and results in a flatness of the curve in the middle of the heat exchanger.

From a thermodynamic point of view, this pinch point effect is positive, because the heat exchange at the pinch point is less irreversible. On the other hand, the heat transfer with the small temperature difference demands a large surface, and the heat exchanger surface is used ineffectively. This takes place, for example, in the area from 4.5 m to 6 m for mixture # 1, in the area from 0 to 2.5 m for mixture # 2 and in the area from 3.5 m to 4.5 m for mixture # 3.

The influence of the pinch points is thus very important for the understanding of the behavior of a mixed gas heat exchanger. One can subdivide all mixtures in the groups based on the “pinch-point” criterion: how many pinch points does a mixture have, and at which place in the heat exchanger do these pinch points occur. Such an understanding considerably simplifies further analysis.

The data for the pressure distribution on the shell side of the heat exchanger are presented in Figure 4. The pressure distribution depends on the location of the pinch-point in the heat exchanger similar to the temperature distribution. The pressure drop for mixture # 1 (with a pinch point on the warm end of the heat exchanger) is greater than that for mixture # 2 (with a pinch point at the cold end).

It is a well-known fact, that for a heat exchanger operating with high-density fluids, the pressure drop expenditure relative to the heat transfer rate is generally smaller than for low-
density fluids. So the ratio of pressure drop to heat transfer coefficient for mixtures at lower
temperatures (80-100 K) is smaller than that at higher temperatures (250-300 K).

Figure 4 illustrates this effect: in the heat exchanger working with mixture # 1 approxi-
mately 25 % of the surface operates at the warm temperature (pinch point area), therefore the
heat exchange in this area is associated with a relatively high pressure drop (practically all
losses are concentrated in the pinch area). On the other hand, in the heat exchanger working
with mixture # 2, the fraction of the surface operating at the high temperature is relatively
small. This results in a smaller pressure drop for the shell side and consequently a lower
cooling temperature can be obtained.

Moreover, this understanding helps to explain the curves for the cooling capacities and
cooling temperatures of mixed gas J-T coolers. Figure 5 shows these for mixtures # 1 and # 2 for
different heat exchanger lengths. One can observe the following:

1. For both mixtures, the minimal length is approximately 2 m. If the heat exchanger is
shorter than this value, a cooling temperature lower than 85 K can not be reached.
2. With a heat exchanger length of 3 m an acceptable cooling capacity (10 W @ 86 K for mixture #1 and 14 W @ 84 K for mixture #2) can be achieved.

3. The maximum cooling capacity of 15-16 W can be produced with mixture #1 at 88-90 K, if the heat exchanger is longer than 5 m.

4. Approximately the same heat exchanger length is required to realize the potential of mixture #2; however, with mixture #2 a higher cooling capacity (18 W) at a lower temperature (below 85 K) can be produced.

5. The cooling temperature increases with increased heat exchanger length. But for mixture #1 the rise is higher than for mixture #2.

6. The use of mixture #2 is more favorable since a lower optimal cooling temperature and a larger cooling capacity of the cooler can be reached.

From this point of view, the use of mixtures with pinch points in the colder part of the heat exchanger is more favorable for a mixed gas J-T-system.

From a thermodynamic point of view, each system has an optimal heat exchanger length. If the heat exchanger is shorter, the cooling capacity of the system decreases. If the heat exchanger is longer, it leads to a higher pressure drop and to an increased cooling temperature and decreased cooling capacity. The optimal minimal temperature difference between the high and low pressure streams in the heat exchanger for the 80-90 K systems amounts to less than 0.5 K. However, the choice of the best heat exchanger length is more than a thermodynamic problem. In many cases it is reasonable to make the heat exchanger shorter than optimal. Although the cooling capacity decreases, it can bring some other advantages. For example, a smaller

Figure 5: Cooling capacity and cooling temperature for different heat exchanger lengths
refrigerant hold-up results and this allows simpler compressor start-up, because of the lower stand-still pressure in the system. This is important for safety, particularly if flammable mixture components are used. Another advantage of a shorter heat exchanger is a lower mass and volume of the cold head, which can result in a faster cool down.

Analysis of the temperature distribution in real systems can give some useful information for adjusting the mixture composition, i.e., a flat part of the curve indicates the temperature range of a component, whose fraction in the mixture is smaller than optimal.

**CONCLUSION**

A model of a mixed gas J-T system was developed, which allows one to investigate the steady state behavior in the heat exchanger of a mixed gas J-T-system.

The calculations show that the temperature distribution as well as the pressure distribution in the heat exchanger depends primarily on the pinch point locations in the heat exchanger. This understanding considerably simplifies the analysis. The study of the temperature and pressure profiles allows one to obtain some important data for the optimization of the mixture.

To use this model for the design of real systems, a check needs to be made of the computed results and adjustment made of its parameters, if necessary.

**REFERENCES**

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