Development of an Internal Heat Exchanger (IHX)

Shuji Kumamoto*  Hiroyuki Ohno*

Abstract

The current automotive air conditioning refrigerant "R134" will be prohibited as a measure against global warming and replaced with a lower GWP (Global Warming Potential) refrigerant such as R1234yf. If they are just replaced, the cooler performance degrades due to different physical properties. For a solution, we have been studying on adoption of an Internal Heat Exchanger (IHX). This report introduces our invention of the IHX structured with a unique spiral fin inside a double tube and demonstrates its potential with CAE analysis and validation test results.

Key Words : Air conditioning / Refrigerant / IHX / Spiral Fin

1. Introduction

With the worldwide problem of global warming, recently it was determined that a refrigerant (R134a/GWP: 1,340) with a high global warming potential (GWP) which is currently used for automotive air conditioners is also going to be regulated. For this reason, the refrigerant used as an automotive air conditioning refrigerant will be changed to a new refrigerant (R1234yf/GWP: 4) with a low GWP. However, R1234yf has been proven to have a reduced cooling capacity than R134a due to the influence of its thermodynamic & thermophysical properties. One useful method to remedy this decrease in capacity is to use an internal heat exchanger (IHX). This IHX is also an AC credit item in North America and demand for the IHX is expected to increase in the future. Therefore, we have devised and developed our own IHX structure.

2. Characteristics of IHX

2.1. Capability of IHX

Fig. 1 shows a cooling cycle with an IHX installed, and Fig. 2 shows the outline of the heat exchange mechanism of the IHX. By exchanging heat between the high-temperature and high-pressure liquid refrigerant of the condenser outlet and the low-temperature and low-pressure gas refrigerant of the evaporator outlet, the IHX can improve AC system cooling capacity.

Table 1 shows ratios of the enthalpy difference between the inlet and outlet of the evaporator, the refrigerant flow rate, and the cooling capacity of both R1234yf itself and R1234yf applied with IHX compared with those of R134. Fig. 3 shows the Mollier diagram at this time. As indicated in Fig. 3, as the gas-liquid saturation enthalpy difference of R1234yf is smaller than that of R134a, the
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enthalpy difference between the inlet and outlet of the evaporator is decreased by approx. 25% by replacing R134a with R1234yf. In addition, as the refrigerant density of compressor suction gas of R1234yf is greater than that of R134a, its flow rate is increased by approx. 20%, and its cooling capacity, which is the product of the enthalpy difference by the refrigerant flow rate, is decreased by approx. 10%.

Table 1 Ratio relative to R134a

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<thead>
<tr>
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<th>R134a</th>
<th>R1234yf</th>
<th>R1234yf +IHX</th>
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<tbody>
<tr>
<td>Enthalpy difference</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Evaporator</td>
<td>1</td>
<td>0.75</td>
<td>0.90</td>
</tr>
<tr>
<td>IHX</td>
<td>-</td>
<td>-</td>
<td>0.15</td>
</tr>
<tr>
<td>Refrigerant flow rate</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>1.20</td>
<td>1.11</td>
<td></td>
</tr>
<tr>
<td>Heat exchange amount</td>
<td></td>
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</tr>
<tr>
<td>Evaporator</td>
<td>1</td>
<td>0.90</td>
<td>1.00</td>
</tr>
<tr>
<td>IHX</td>
<td>-</td>
<td>-</td>
<td>0.17</td>
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<table>
<thead>
<tr>
<th></th>
<th>R1234yf p-h diagram</th>
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<tbody>
<tr>
<td>R134a</td>
<td>Saturation line of R134a</td>
</tr>
<tr>
<td>R1234yf</td>
<td>Saturation line of R1234yf</td>
</tr>
<tr>
<td>R1234yf+IHX</td>
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Fig. 3 Mollier diagram

When remeasuring this decrease in cooling capacity by specification changes of the current components, a lot of issues to be resolved will arise. For example, premising that the size of the condenser is not changed, when increasing the ratio of the subcooling condenser and reducing the enthalpy of the evaporator inlet, the area of the main condenser becomes smaller, and therefore, compressor discharge pressure increases. Along with this, because the performance of the subcooling condenser depends on the conditions on the air side, there is a possibility of some negative effects such as failing to obtain a stable heat exchange amount. When the refrigerant flow rate is increased by increasing the speed of the compressor, compressor power is also increased.

On the other hand, IHX has no impact to discharge pressure, and it has no influence of outside air temperature since refrigerant from a condenser and from an evaporator exchange heat each other within same cycle. It also makes compressor power lower due to decrease of refrigerant flow rate. As an adverse effect on cycle, we can think of the rise of compressor discharge temperature caused by increase in compressor suction superheat. However, when the IHX is applied to R1234yf cycles, the compressor discharge temperature has difficulty rising to the guaranteed temperature of the compressor due to the property differences (such as a compressor compression ratio and gradient of an isentropic curve) from R134a. Based on the above, it is effective to use the IHX as a measure to improve cooling capacity.

2.2. Heat exchange amount required by IHX

Because the IHX can reduce the enthalpy before the evaporator by exchanging heat between the high-temperature and high-pressure liquid refrigerant before the expansion valve and the low-temperature and low-pressure gas refrigerant of the evaporator outlet, it can increase the enthalpy difference between the inlet and outlet of the evaporator. On the other hand, because the IHX exchanges heat between the refrigerants in the same cycle, the compressor suction enthalpy becomes larger by the decreased amount of the enthalpy before the evaporator. As this makes the density of the compressor suction refrigerant smaller, the refrigerant flow rate is also decreased. As indicated in Table 1, when taking the evaporator enthalpy difference and the refrigerant flow rate of R134a as 1 and taking the IHX enthalpy difference of R1234yf cycle as 0.15, the evaporator enthalpy difference becomes 0.90 and the refrigerant flow rate becomes 1.11, and consequently the cooling capacity of the evaporator becomes equivalent to that of R134a. The heat exchange amount of the IHX in this case becomes 0.17, which is larger than the improved amount of cooling capacity 0.10.

3. IHX structure

3.1. Basic structure of IHX

Our development product is a double tube type in which low-pressure gas refrigerant flows in the inner tube and...
high-pressure liquid refrigerant flows between the inner tube and outer tube (Fig. 4). This structure has an advantage making it easy to change current systems by replacing the high pressure piping and low pressure piping with the IHX. However, the length of the heat exchange area might be restricted because the IHX is handled as piping as well. For this reason, it is necessary to improve its heat exchange performance and reduce the length for the required capacity.

3.2. Performance improvement measure
The heat exchange amount of a double tube is expressed by the following equation:

\[
Q = \frac{\Delta T}{\frac{1}{\alpha_{lo}A_{lo}} + \frac{\ln(A_{hi}/A_{lo})}{2\pi\lambda L} + \frac{1}{\alpha_{hi}A_{hi}}}
\]

- \(Q\): Heat exchange amount [W]
- \(\Delta T\): Temperature difference between high pressure refrigerant and low pressure refrigerant [K]
- \(\alpha_{lo}\): Heat transfer coefficient of low pressure gas refrigerant [W/m²K]
- \(A_{lo}\): Heat transfer area on the low pressure side [m²]
- \(\alpha_{hi}\): Heat transfer coefficient of high pressure liquid refrigerant [W/m²K]
- \(A_{hi}\): Heat transfer area on the high pressure side [m²]
- \(\lambda\): Heat conductivity of inner tube [W/mK]
- \(L\): IHX length [m]

In the heat exchange of the double tube, because the heat transfer coefficient of low pressure gas refrigerant is extremely low compared to that of high pressure liquid refrigerant, the former acts as a dominant factor. Therefore, we can say that improving the heat transfer coefficient of the low-pressure gas refrigerant is the most effective way to improve the heat exchange amount of the IHX.

Focusing on the two points: (1) improving the heat transfer coefficient by agitating the flow on the low pressure gas refrigerant side as much as possible, (2) making it possible to manufacture easily and anywhere and to reduce costs, we brought up ideas for this development product (Fig. 4) and devised a simple structure consisting of two element tubes with different diameters and a fin produced by forming a plate material into a spiral shape.

We decided the fin pitch based on performance, pressure loss, and manufacturing requirements, but we are planning to provide flexibility to the manufacturing side assuming the possibility of pitch change according to the requirement specifications (such as performance, pressure loss, and length) for the IHX.

3.3. Performance analysis

Fig. 5 shows the results of flow analysis on the low pressure side of the spiral fin structure. The faster the flow rate is represented by red color, and the slower flow rate by blue color. As can be seen from the Figure, the stream line turns based on the spiral fin design. Fig. 6 shows the analysis results of the heat transfer coefficients of the high pressure liquid refrigerant and low pressure gas refrigerant each for the spiral fin and the extruded tube (previous structure) and the heat exchange amount of the IHX. The previous structure of the extruded tube used for the comparison has multiple holes in the high pressure flow path and no special structure in the low pressure flow path (Fig. 7).

Compared to the extruded tube without a structure in its inner tube, the heat transfer coefficient of the low pressure gas of this spiral fin type is improved by 45%.
On the other hand, as for the heat transfer coefficient of the high pressure liquid, that of the extruded tube type with a multi-hole structure in its high pressure flow path is higher by 20% compared to the spiral fin type, but the total heat exchange amount of the spiral fin type is 30% higher.

### 4. Experimental results

As we have confirmed the usefulness of the spiral fin from the analysis, we conducted a benchtop demonstration experiment. We confirmed that the spiral fin can improve the IHX heat exchange amount by 30% compared to the extruded tube (Fig. 8), just like the results obtained by the analysis. Based on this result, when the heat exchange amounts are equivalent, the spiral fin can reduce the length of the IHX by approx. 30% compared to the extruded tube.

![Fig. 6 Analysis results of heat transfer coefficient](image)

**Fig. 6 Analysis results of heat transfer coefficient**

![Fig. 7 Extruded tube](image)

**Fig. 7 Extruded tube**

![Fig. 8 Experimental result of heat exchange amount](image)

**Fig. 8 Experimental result of heat exchange amount**

### 5. Conclusion

By these analysis and experiment conducted, we confirmed the improvement effect of the heat transfer coefficient and heat exchange amount on the low pressure side by the spiral fin structure. With this higher performance and the flexible layout associated with it, this development product can contribute to power saving as well as fuel saving of air conditioners.