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"DEVELOPMENT OF SMALL HERMETIC COMPRESSORS FOR R600a"

By
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ABSTRACT

Recently the possibility of using R600a has gained popularity among producers of household refrigerators in Europe, especially in Germany. This paper deals with the development of small hermetic compressors for this refrigerant. Part one is describing the thermodynamical properties that are influencing the compressor performance. Measurements and compressor simulations are presented and the necessary changes in the design are discussed. In part two the reliability of compressors for R600a is analyzed, including chemical stability, wear and lifetime and the compatibility of the involved materials.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>T</td>
<td>Temperature</td>
<td>°C</td>
</tr>
<tr>
<td>P</td>
<td>Pressure</td>
<td>bar</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient Of Performance</td>
<td>W/W</td>
</tr>
<tr>
<td>c</td>
<td>Specific heat capacity</td>
<td>J/kg K</td>
</tr>
<tr>
<td>ODP</td>
<td>Ozone depletion potential</td>
<td>-</td>
</tr>
<tr>
<td>POE</td>
<td>Polyolester oil</td>
<td>-</td>
</tr>
<tr>
<td>Greek</td>
<td>Density</td>
<td>kg/m³</td>
</tr>
<tr>
<td>p</td>
<td>Adiabatic exponent: c_p/c_v</td>
<td>-</td>
</tr>
<tr>
<td>k</td>
<td>Adiabatic exponent: c_p/c_v</td>
<td>-</td>
</tr>
</tbody>
</table>

Subscript:
- e: Evaporator
- c: Condenser
- a: Compressor condition without sub-cooling (CECOMAF)
- b: Compressor condition with sub-cooling (ASHRAE)
- p: Isobaric value
- v: Isochoric value

INTRODUCTION

The transition from R12 to R134a in domestic refrigeration is almost finished in Europe and well under way elsewhere. About 1988 the choice of R134a as alternative was made because it fulfills a list of key requirements: Zero ODP, thermodynamically similar to R12, non toxic, inflammable. If, however, it is possible to disregard the requirement of inflammability hydrocarbons could be a valid alternative, e.g. R600a, R600, R290. A non-azeotropic blend of R600a/R290 may be formulated with almost the same pressure characteristic as R12 but due to the non-azeotropic properties a standard refrigerator will normally require some redesign and blends are therefore not seen as universal alternatives. This paper deals solely with pure R600a.

THEORETICAL COMPARISONS - R134a/R600a

Thermomechanical analyses of the properties of refrigerant give theoretical information of how the compressors are going to operate in different situations, i.e. different evaporator temperature, subcooling or overheating environment, with different ambient temperatures and pressures etc. The thermomechanical behaviour also gives the first and general design criteria. Theoretical calculations are here presented and compared to R134a. The analysis should be seen qualitatively rather than quantitatively. This is because the theoretical comparisons disregard the actual compressor design, e.g. clearance volume, valve design, flow restrictions etc. These elements have a decisive impact on the final performance of the compressor, and of equal importance as the thermodynamical properties.
Table 1. Comparisons between R134a and R600a (Condenser temperature 55 °C).

Vapour pressures.
Table 1, columns 1, 2, 8 and 9, shows the saturated vapour pressure of R134a and R600a at different evaporating temperatures and at a condensing temperature of 55 °C. It is seen that the vapour pressure for isobutane is much lower.

Pressure difference.
The pressure differences which are caused from the compression from $P_c$ to $P_v$ are shown in columns 3 and 10 respectively. It is seen that the pressure difference of R600a is approx. 51% lower than for R134a which means that the bearings are not so heavily loaded when using R600a.

Compression ratio.
The compression ratios are shown in table 1 columns 4 and 11. The compression ratios of the two refrigerants are deviating just 6% at -40°C/55°C and 3% at 10°C/55°C (R600a being a little smaller).
The compression ratio affects the volumetric efficiency of the compressor. The smaller the compression ratio, the better volumetric efficiency. A factor of importance in the calculation of the volumetric efficiency is the effect of the reexpansion of clearance volume. This reexpansion is shown in figure 1. It is seen that R600a is more sensitive to clearance volume although the pressure ratio is smaller. This is due to the influence of the adiabatic exponent $\kappa = c_p/c_v$. $\kappa$ is not constant during the compression. In the above calculations an average of $\kappa = 1.1$ for R600a and $\kappa = 1.13$ for R134a is used.

End compression temperatures.
The temperatures at the end of isentropic compression are shown in table 2 columns 5 and 12. It is seen that the end temperatures are lower for R600a than for R134a. This calculation must be taken only as a guideline because the end temperature of the gas is affected by the total energy balance of the compressor. The calculations in table 1 are based on a fixed temperature of 32 °C at the beginning of a polytrophic process.
compression with $\kappa=1.1$ for R600a and $\kappa=1.13$ for R134a.

**Rankine cycle efficiency - COP.**
The Rankine cycle gives the maximum ratio of heat consumption in the evaporator to the power consumption with isentropic compression and no compressor losses. Table 2, columns 6 and 13, shows the COP for a Rankine cycle without subcooling and 32°C superheating. It is seen that the ideal COP is from 26% to 16% higher for R600a (higher for lower $T_e$). The COP with 32°C subcooling is 24% to 13% better for R600a. Again these values must be taken as guidelines because the real compressor efficiency is affected by thermal losses, flow losses and internal superheating.

**Compressor volume flow.**
The volume flow through the compressor depends on the required evaporator yield, the latent and specific heat capacity of the refrigerant and the refrigerant density at cylinder intake.
In figure 2 the calculated volume flows at different operating conditions are shown. The volume flows are normalised to R134a at condition -40°C/55°C/32°C. It is seen that the required volume flow to obtain the same evaporator yield is approx. 70% higher for R600a due to its high specific volume. The specific volume for R600a is approx. 3.2 times as high as the specific volume of R134a with the same operation conditions.
The calculations are made with the assumptions of 32°C superheating.

**System mass flow.**
The mass flow through the system is dependent on the refrigerant and the required evaporator yield. The required mass flow for R600a to obtain a certain evaporator yield is approx. 50% of the required mass flow for R134a.

From the above comparisons it is seen that the two refrigerants R600a and R134a, from a thermodynamic point of view, are very different. Changing from R134a to R600a has thus a large impact on both the compressor design and the design of the full refrigeration system.

**COMPRESSOR MEASUREMENTS AND SIMULATION**
Danfoss Compressors GmbH has during the last year developed a range of compressors for R600a with stroke volumes from 4 to 15 cm³. In this section a comparison between a compressor designed for R600a is made to a compressor designed for R134a. The compressors are of approx. the same capacity (see table 2).

It is seen from table 2 that the compressor using R600a has a COP only 5-6% better than R134a, i.e. the COP is considerably less than the theoretical 20% better Rankine cycle COP. Measurements and transient compressor simulations indicate that this difference may be explained by the increased flow losses due to higher volume flow. Another important source to the loss of efficiency might be the lower volumetric heat capacity for R600a compared...
to R134a (the volumetric heat capacity is by evaporating temperature of -25°C and 30°C overheating approx. 40% lower for R600a than for R134a) resulting in an increased internal superheating and consequently in high gas temperatures by inlet to the compression chamber.

<table>
<thead>
<tr>
<th>Compressor</th>
<th>Capacity [W]</th>
<th>Consumption [W]</th>
<th>COP$^*$ (CECOMAF)</th>
</tr>
</thead>
<tbody>
<tr>
<td>TLES7K</td>
<td>-25</td>
<td>-23.3</td>
<td>-10</td>
</tr>
<tr>
<td>TLES4F</td>
<td>69</td>
<td>84</td>
<td>153</td>
</tr>
</tbody>
</table>

Table 2. Measured performance comparisons between two Danfoss compressors with similar capacity. Both compressors are of the optimized type with suction muffler and optimized motor. The TLES7K compression volume is 6.49cm$^3$ and the TLES4F compression volume is 3.86cm$^3$.

**COMPRESSOR TESTS.**

A refrigerant is only acceptable for use in domestic refrigeration if it is possible to select materials, which together with the refrigerant will give the desired lifetime of the compressor. The compressor manufacturer has to produce compressors for many different refrigerants, e.g. R134a, R404a, R12, R22, R502 etc. To facilitate the logistics it is important to choose materials compatible to as many refrigerants as possible. We have chosen to use the same materials and production procedures for R600a and R134a. The only exception is the oil, where a mineral oil is used instead of a polyolester oil, because of the price difference.

To check that the desired reliability is obtained a variety of tests has been made:

**Capillary Tube Blocking.**

The change from R12 to R134a made necessary a change in materials and production material residuals due to differences in polarity of the refrigerant molecules. Isobutane is not a very polar molecule and it is therefore necessary to check whether some substances might be able to block the capillary tubes.

In order to investigate this, capillary tube blocking tests were carried out. The compressor is placed in a normal refrigeration system and runs under the conditions -35°C/45°C and with a housing temperature of 100°C. After running, the amount of residue in the capillary tube and the change in flow rate through the capillary tube are measured. After 200 hours there is no restriction at all in the capillary tube and no residue detected. After 1000 hours, the flow rate has changed 0.9% and 0.4 mg of residue is found. Gas Chromatography (G.C.) analysis of the residue indicates only the presence of mineral oil.

**Compressor Lifetime Tests.**

The test conditions listed in table 6 have been used for several years, allowing us to compare test results obtained here with results from tests with e. g. R134a (2). Some typical compressors tested are depicted in table 5
In two tests a standard well-refined refrigeration oil on mineral oil basis with a viscosity of 15 cSt at 40°C was used. For one of the tests a mineral oil at 7cSt was used.

<table>
<thead>
<tr>
<th>Compressor</th>
<th>TLES5K</th>
<th>TLES6K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Discharge pressure (Bar)</td>
<td>12.1</td>
<td>12.1</td>
</tr>
<tr>
<td>Suction pressure (Bar)</td>
<td>2.2</td>
<td>1.3</td>
</tr>
<tr>
<td>Winding temperature (°C)</td>
<td>145</td>
<td>145</td>
</tr>
<tr>
<td>On/Off (min.)</td>
<td>27.3</td>
<td>27.3</td>
</tr>
</tbody>
</table>

Table 6.

During testing gas samples were taken out for analysis. After the test the compressors were disassembled, and the parts and the oil were analyzed. Table 7 summarizes the evaluation of the disassembled compressors.

<table>
<thead>
<tr>
<th>Compressor</th>
<th>TLES5K</th>
<th>TLES6K</th>
<th>TLES6K</th>
<th>TL5F¹</th>
</tr>
</thead>
<tbody>
<tr>
<td>Min. Oil viscosity</td>
<td>15</td>
<td>15</td>
<td>15</td>
<td>POE:14</td>
</tr>
<tr>
<td>Wear</td>
<td>0</td>
<td>1-2</td>
<td>0</td>
<td>0-1</td>
</tr>
<tr>
<td>Cu-plating</td>
<td>0</td>
<td>1-2</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Valve-deposits</td>
<td>0</td>
<td>0-1</td>
<td>0</td>
<td>0-1</td>
</tr>
</tbody>
</table>

Table 7. 0=new. 1-2=Acceptable. 3-4=Not Acceptable. Reference is a TL5F with POE (14cSt) from life time testing with R134a (2).

After lifetime test of 2000h and 4000h respectively, wear at the level of R134a is seen. After a 8000h lifetime test the wear is still very acceptable. Some Cu-plating was found, but not in a degree that can harm the functioning of the compressor. No coking, but a very slight brownish deposit was found on the valves. The quantities however, were too small to allow any analysis. All other parts of the compressor including the motor were in a very acceptable condition. The results of the lifetime test with a mineral oil of 7cSt indicate, that oil viscosities below 15cSt are relevant for R600a.

It was found that the properties of the oil have not changed significantly during lifetime testing. The viscosity and TAN have not changed. The level of Fe in the oil was approx. 1 ppm and the level of Cu was found below 1 ppm.

Stability of the Gas Phase.
For a R12 system the well known reaction between R12 and the refrigeration oil, resulting in the formation of R22, may be taken as a measure of the stability of the hermetic system. In an earlier work (1) it was shown, that similar reactions
between R134a and POE does not occur.

Figure 3 shows the results of the gas analysis. The results represent gas samples from different compressors run in different series of lifetime testing. More than one quality of R600a has been used. Despite the mentioned differences no overall tendency for gas formation is seen. A small amount of CO and CO₂ is formed, but the amounts remain constant after the oxygen level has reached zero due to reactions. Other hydrocarbons present remain at a constant level during lifetime testing. Different impurity components at a level of 30 to 300 vol. ppm found in R600a from the beginning also remain at a constant level. Therefore it is concluded, that the compressors tested here with R600a together with mineral oil is chemical as well as thermal stable in a hermetic system.

CONCLUSION

In this paper the main thermodynamic and reliability aspects of R600a have been discussed. R600a is usable as a refrigerant in small hermetic compressors, where the necessary redesign has been made. The use of R600a requires 45-50% larger compression volume for the same capacity compared to R134a. This means in average use of more material.

Measurements and transient compressor simulations indicate that the theoretical COP gain of 20% is only met by a gain of 5%. The loss of COP gain might arise from higher flow losses and higher internal superheating due to the lower volumetric heat capacity of R600a.

It has been demonstrated, that it is possible to obtain the same compressor reliability with R600a and R134a. The system stability thermal as well as chemical is on same level for the two refrigerants.

LITERATURE