Use of a Ternary Blend in Existing Domestic CFC-12 Appliances

L. Snitkjaer
Danfoss-Flensburg GmbH; Germany

F. K. Eggert
Danfoss-Flensburg GmbH; Germany

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USE OF A TERNARY BLEND IN EXISTING DOMESTIC CFC-12 APPLIANCES

by L. Snitkjær, F.K. Eggert
Danfoss-Flensborg GmbH, P.O. Box 1443, D-2390 Flensburg.

ABSTRACT

It is generally accepted that HFC-134a will be the substitute for CFC-12 in the production of new household refrigerators and freezers. However, there will still be a need for servicing existing domestic CFC-12 refrigerators and freezers. To satisfy this need the drop-in substitute must be fully compatible, thermodynamically and chemically. The drop-in blends are among the most interesting refrigerants, as they seem to fulfill the above requirements.

Below the ternary blend 52% HCFC-22/15% HFC-152a/33% HCFC-124 will be evaluated and compared with CFC-12 on the basis of available thermodynamic properties, laboratory and application tests. The evaluation criteria are capacity, COP, start and stall characteristics.

The influence of inherent differences between test conditions and thermodynamics of the two refrigerants will be discussed.

Nomenclature

<table>
<thead>
<tr>
<th>Symbols</th>
<th>Meaning</th>
</tr>
</thead>
<tbody>
<tr>
<td>p</td>
<td>pressure</td>
</tr>
<tr>
<td>T</td>
<td>temperature</td>
</tr>
<tr>
<td>COP</td>
<td>coefficient of performance</td>
</tr>
<tr>
<td>e</td>
<td>evaporation</td>
</tr>
<tr>
<td>c</td>
<td>condensation</td>
</tr>
<tr>
<td>1</td>
<td>reference saturated liquid</td>
</tr>
<tr>
<td>2</td>
<td>reference saturated vapour</td>
</tr>
</tbody>
</table>

1. INTRODUCTION

International agreements of a ban on the production, marketing and sales of CFC-12 mean that in future this refrigerant will only be available in small quantities in the form of regenerated CFC-12. The need for servicing existing CFC-12 refrigerators and freezers on the market will still exist for a number of years ahead, however, which calls for a solution to this problem.

However, the possibilities are limited, as it is an absolute requirement of a refrigerant substitute that it can be used as a drop-in. Thus the cooling system must function with the drop-in without any design change after completed servicing.

1.1 HFC-134a

It would seem obvious to use HFC-134a as the servicing refrigerant, because this will be available to any fitter due to the fact that HFC-134a is the substitute chosen for CFC-12 in the production of new household refrigerators and freezers. However, the fact is that HFC-134a shows adverse tendencies in systems that have previously been in contact with CFC-12. The presence of even small quantities of chlorine deriving from decomposed CFC-12 will cause operating trouble. The mentioned chlorine residues are almost always present in systems having operated with CFC-12, as they occur at high operating temperatures, which also represent the main reason for service calls.

Also the presence of small residues of oil will cause problems, as mineral oil as well as synthetic oil not are soluble with HFC-134a. Cleaning of the cooling system to counter above problems in most cases will not compare favourably with the value of the appliance in question. In addition, CFC-12 compressors are not designed to use HFC-134a, and the capacity would not be sufficient with HFC-134a.

1.2 Ternary Blends

As a solution to the servicing problem a ternary blend appears to be suitable, as it offers many points of resemblance with CFC-12.

Ternary blends of the type referred to here (HCFC-22/HFC-152a/HCFC-124) will not represent an
alternative to HFC-134a, as the presence of HCFC-22 and HCFC-124 makes the life of the blend limited due to the ozone-depleting nature of these refrigerants (ODP > 0).

In this paper the applicability of a ternary blend as the servicing refrigerant in former CFC-12 systems is examined. The blend being examined 52% HCFC-22 / 15% HFC-152a / 33% HCFC-124 has the technical designation KCD9439A. Its trade name is SUVA Blend MP39. The refrigerant data, including the refrigerant used in the test series, come from DU PONT DE NEMOURS INTERNATIONAL S.A.

Section 2 deals with circumstances being special for KCD9439A and different from CFC-12. Section 3 contains relatively simple theoretical comparisons between KCD9439A and CFC-12. Laboratory measurements to determine the capacity, COP, start and stall characteristics are dealt with in section 4. Application measurements are presented in section 5. In section 6 the final conclusions will be drawn.

2. SPECIAL PROPERTIES OF TERNARY BLENDS

2.1 Selection of Lubricant
The application of the ternary blend as drop-in refrigerant will not be unproblematic because the use of mineral oils as lubricant, will be out of question as this combination leads to risks for valve coking. Therefore it requires a change of lubricant to a synthetic oil when servicing with KCD9439A. if the appliance has previously operated with mineral oil.

2.2 Temperature glide
Another, but rather an important problem, is the temperature glide caused by the use of a ternary blend as a result of the blend not being azetotropic.
At a given evaporation pressure the evaporating temperature will rise during the process of evaporation. There will be a temperature difference of 4 - 7°C from the condition in which the whole refrigerant takes the form of a saturated liquid to the condition where everything is a dry saturated vapour. The extent of the temperature glide depends on the pressure. The biggest difference occurs at low pressures.

The temperature glide among other things gives rise to problems referring to the known CECOMAF/ASHRAE conditions. Is the pressure corresponding to a desired temperature to be chosen in the condition of saturated liquid or of dry saturated vapour? Neither of the mentioned pressures is correct, as the pressure closest to the correct one must be somewhere in between. The complexity of this problem appears from table 1, which also shows the arithmetic average pressure giving an average evaporating temperature $T_{ave}$.

<table>
<thead>
<tr>
<th>$P_e$ [kN/m²]</th>
<th>$T_{z1}$ [°C]</th>
<th>$T_{z2}$ [°C]</th>
<th>$T_{glide}$ [°C]</th>
<th>$T_{ave}$ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1,390</td>
<td>-25,0</td>
<td>-18,9</td>
<td>+6,1</td>
<td>-22,0</td>
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<tr>
<td>1,063</td>
<td>-31,2</td>
<td>-25,0</td>
<td>+6,2</td>
<td>-28,1</td>
</tr>
<tr>
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<td>-21,8</td>
<td>+6,2</td>
<td>-25,0</td>
</tr>
</tbody>
</table>

Others (references /1/ and /2/) have solved above mentioned problem by choosing the arithmetic average pressure lying exactly between the saturated liquid and dry saturated vapour conditions. The same method has been applied in this paper.

3. THEORETICAL COMPARISONS

By analyzing the thermodynamic properties of a refrigerant one gets a theoretical picture of how the refrigerant will behave in different situations of operation (at different evaporating temperatures). The applicability of the analysis improves by drawing a comparison to a familiar refrigerant. Thus in the following theoretical examinations of KCD9439A parallels have been drawn to CFC-12. The results of the analysis are to be considered as qualitative rather than quantitative, as the used
theory takes no account of the compressor design, including the size of the clearance volume, which has a decisive influence on the final result.

Table 2 shows data for CFC-12 and KCD9439A, respectively, which will be dealt with below.

Table 2. Comparisons between CFC-12 and KCD9439A ($T_e = 55^\circ$C)

<table>
<thead>
<tr>
<th>Column</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
<th>13</th>
<th>14</th>
<th>15</th>
<th>16</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_e$</td>
<td>$(^\circ$C)</td>
<td>10^6 N/m²</td>
<td>$(^\circ$C)</td>
<td>COPa</td>
<td>COPb</td>
<td>$P_e$</td>
<td>$P_s$</td>
<td>$P_e-P_s$</td>
<td>$T_{end}$</td>
<td>COPa</td>
<td>COPb</td>
<td>$P_e$</td>
<td>$P_s$</td>
<td>$P_e-P_s$</td>
<td>$T_{end}$</td>
<td>COPa</td>
</tr>
<tr>
<td>-45</td>
<td>0.50</td>
<td>13.66</td>
<td>13.2</td>
<td>27.1</td>
<td>169</td>
<td>1.42</td>
<td>1.70</td>
<td>0.46</td>
<td>0.59</td>
<td>0.40</td>
<td>15.29</td>
<td>14.6</td>
<td>32.2</td>
<td>3.22</td>
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<td>12.9</td>
<td>16.9</td>
<td>168</td>
<td>1.71</td>
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<td>0.78</td>
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<td>15.29</td>
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<td>1.24</td>
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<td>11.0</td>
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<td>1.23</td>
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<td>13.66</td>
<td>11.8</td>
<td>7.5</td>
<td>133</td>
<td>2.53</td>
<td>3.03</td>
<td>1.85</td>
<td>2.07</td>
<td>1.62</td>
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<td>-5</td>
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<td>2.41</td>
<td>15.29</td>
<td>12.6</td>
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<td>125</td>
<td>3.24</td>
<td>3.84</td>
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<tr>
<td>+5</td>
<td>3.63</td>
<td>13.66</td>
<td>10.0</td>
<td>3.8</td>
<td>106</td>
<td>4.03</td>
<td>4.84</td>
<td>3.81</td>
<td>4.17</td>
<td>3.45</td>
<td>15.29</td>
<td>11.5</td>
<td>4.0</td>
<td>110</td>
<td>4.27</td>
<td>5.06</td>
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<tr>
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<td>13.66</td>
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<td>2.8</td>
<td>94</td>
<td>5.34</td>
<td>6.44</td>
<td>5.24</td>
<td>5.69</td>
<td>4.79</td>
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<td>10.1</td>
<td>2.9</td>
<td>97</td>
<td>5.74</td>
<td>6.82</td>
</tr>
</tbody>
</table>

$T_{end}$: End Compression Temperature
COPa: Ideal COP. Without subcooling, 32°C superheated.
COPb: Ideal COP. With 32°C subcooling, 32°C superheated.

3.1 Vapour Pressure

Table 2, columns 1 respectively 8, 9 and 10, show the vapour pressures of CFC-12 and KCD9439A, at different evaporating temperatures. As for KCD9439A 3 different pressures are shown, of the temperature glide problems. It appears from the table that the average vapour pressure of KCD9439A is very similar to the vapour pressure of CFC-12 in the interval $-45 < T_e < -15^\circ$C.

However, at higher evaporating temperatures the vapour pressure of KCD9439A increases compared to CFC-12.

3.2 Pressure Difference

The pressure difference between the suction and discharge side of the compressor discloses more about how the compressor is affected by the refrigerant. Table 2, columns 3 and 12, show the pressure difference between the evaporating and condensing pressure of CFC-12 and KCD9439A, respectively, ($T_e = 55^\circ$C). The pressure difference is larger for KCD9439A than for CFC-12 during the whole interval of evaporating temperatures.

The larger pressure difference by using KCD9439A causes the compressor bearings to be more heavily loaded as well as the leakage loss in the clearance between piston-cylinder to rise, resulting in a poorer capacity. The load on gaskets and valves also increases.

3.3 Compression Ratio

The compression ratio influences the capacity of the compressor, as it affects the volumetric efficiency as well as the compression work. The smaller the compression ratio, the better the efficiency. Table 2, columns 4 and 13, show the compression ratio of the two refrigerants at different evaporating temperatures. From being equal at high evaporating temperatures, the compression ratio rises more for KCD9439A than for CFC-12, by decreasing evaporating temperature.

All other things being equal, the conclusion can be drawn that KCD9439A will have a steeper capacity characteristic than CFC-12, as the increased compression ratio for KCD9439A at low evaporating temperatures will affect the capacity. For the same reason KCD9439A will be more sensitive to the size of the clearance volume.

3.4 End Compression Temperature

The temperature of the gas at the end of the compression is interesting, because a high end compression temperature means degeneration of the oil, coking in the discharge valve and carbonization. A high end compression temperature also raises the entire temperature level of the compressor, including the suction gas temperature, which results in a reduced capacity, just as the
working temperature of the motor will increase resulting in reduced efficiency.

Table 2 shows the end compression temperature of the two refrigerants in columns 5 and 14. The calculation of the end compression temperature is based on the assumption of isentropic compression, with the cylinder entrance temperature being fixed at 50°C. The difference between the two refrigerants only represents 3-4°C at high evaporating temperatures, whereas the difference at low evaporating temperatures is up to 12°C to the disadvantage of KCD9439A. Operating at low evaporating temperatures combined with high condensing temperatures and considerable cylinder entrance temperature results in an even higher end compression temperature.

3.5 Ideal COP
The ideal COP is defined as the enthalpy difference across the evaporation process divided by the enthalpy difference across the compression process, presuming isentropic compression.

In table 2 the ideal COP appears as a function of the evaporating temperature, the condensing temperature being constant and equal to 55°C. The cylinder entrance temperature is fixed at 50°C. Columns 6 and 15 represent the condition of 32°C superheating and no subcooling. Similarly, columns 7 and 16 represent the ideal COP for the condition of 32°C subcooling and 32°C superheating. Table 2 shows that in both cases the ideal COP characteristic for KCD9439A has the greatest slope.

Furthermore, it is shown that the capacity (COP) increases by approx. 19% by changing from a condition without subcooling to the same condition with subcooling. This goes for both refrigerants.

The values in table 2 are based on the same cylinder entrance temperature. As the temperature level is expected to be higher when using KCD9439A, cf section 3.4, it means – other things being equal – that the COP will be smaller by using KCD9439A than shown in table 2.

4. COMpressor MEASUREMENTS

To illustrate the properties of KCD9439A used in a hermetic compressor, tests have been made using a 7.3 cm3 reciprocating compressor. All measurements have been made on the same compressors, first with CFC-12 and mineral oil, and then with KCD9439A and synthetic oil. No compressor designs were made. This is due to the requirement for the use of KCD9439A as a drop-in that satisfactory results must be achieved without changes of design.

4.1 Capacity
Figure 1 shows that the capacity by using KCD9439A is at the same level as CFC-12 when T = 28°C. The capacity characteristic, however, has a greater slope for KCD9439A than for CFC-12. The capacity characteristic is in accordance with the theoretical considerations. However, the measured capacity level with KCD9439A is influenced by the fact that the evaporation pressure was chosen as the arithmetic average pressure across the temperature glide, which probably ought to be lower considering the fact that the real evaporation process in an appliance does not start with a 100% saturated liquid.

This means that the capacity obtainable in a system probably will be smaller than shown in figure 1, because the curves for KCD9439A ought to be displaced x °C to the right.

4.2 Coefficient of Performance (COP)
The circumstances applying to the capacity characteristic are reflected in the COP characteristic. Thus the two refrigerants have the same COP at low evaporating temperatures. See figure 2. At rising evaporating temperatures the COP improves by using KCD9439A. At high evaporating temperatures the COP difference between CFC-12 and KCD9439A is reduced because KCD9439A needs a higher torque due to the larger capacity, which leads to a poorer motor efficiency. As mentioned under 4.1 the curves for KCD9439A ought to be displaced to the right in this case too.

4.3 Stall Characteristic
Comparative measurements between KCD9439A and CFC-12 establish that the compressor needs more torque by using KCD9439A than by using CFC-12. At a given evaporating temperature and a given voltage the compressor can build up a larger counter pressure with CFC-12 than with KCD9439A, before the compressor gets below the RPM at which it stalls.

In terms of torque this means that the requirement of the motor will be approx. 5% higher, on the
Figure 1. Capacity Characteristic for a 7.3 ccm Reciprocating Hermetic Compressor. Without subcooling. With 32°C superheating.

Figure 2. COP Characteristic for a 7.3 ccm Reciprocating Hermetic Compressor. Without subcooling. With 32°C superheating.
assumption that the requirements using KCD9439A are the same as of CFC-12. Further investigations on applications are needed.

4.4 Start Characteristic
A critical start situation is influenced by the temperature level in and around the compressor and the percentage of saturated liquid in the system, which again gives rise to reference problems.

Measurements show that the compressor can start at a higher equalizing pressure when using KCD9439A as when used with CFC-12. Converting this pressure to the corresponding temperature it can be established that the necessary starting voltage with KCD9439A will not exceed the corresponding necessary starting voltage with CFC-12 at the same equalizing temperature. Further investigations on applications are needed.

5. APPLICATION TESTS

5.1 Description of Test Object
For the application tests a 330 l freezer was chosen. The design principle of the cooling system is shown in figure 3. The evaporator consists of 7 sections, evenly distributed in the height of the appliance. The condenser is split up into two sections, one section as mullionheater and one as main condenser. The condenser is of the skin type. The measuring points for the pressure and temperature measurements, respectively, on the system appear from figure 3.

5.2 Tests
The freezer was put through the following test:

a) Testing of the appliance with CFC-12. The tests were made with a compressor corresponding to the original compressor, only with known calorimeter data.

b) Testing of the appliance with KCD9439A. The same compressor as described under a) was used, except that the compressor was charged with alkyl benzene. Before measuring the appliance the compressor was measured on a calorimeter with KCD9439A.
The following individual tests have been carried through:

- No load pull down at 32°C ambient temperature
- Continuous operation at 32°C ambient temperature with no load
- Thermostatic operation at 25°C ambient temperature. During these tests the temperature was measured in 0.5 kg test packages, one in each section.

Thus the measurements were made with identical components, except for the refrigerant and the lubricant. The refrigerant charge when operating with KCD9439A is 150g and 155g with CFC-12.

5.2.1 No Load Pull Down. Test Results.
Figures 4 and 5 show the temperature and pressure progress of the test conducted with CFC-12 and KCD9439A, respectively.

Figure 4. No load pull down, 32°C ambient temperature, CFC-12.

Figure 5. No load pull down, 32°C ambient temperature, KCD9439A.
Figure 4 shows that using CFC-12 the system apparently works in a satisfactory way. The course of temperature across the evaporator shows a considerable pressure loss. The temperature at evaporator inlet is 3-5°C higher than at evaporator outlet. The superheating during the last part of the evaporator is a bit on the large side. This shows that the amount of refrigerant ought to be increased. The course of temperature and pressure across the condenser indicates that the capacity of the capillary tube is a little too high.

Figure 5 shows, evaluated on temperature during evaporation, that the chosen charge of refrigerant, secures a quick utilization of the evaporator using KCD9439A. By comparing the temperature measured at condenser outlet (pos 10), with the equivalent saturated liquid temperature to the measured pressure (pos 42), and corresponding equivalent saturated vapour (pos 42), it can be seen that the measured temperature (pos 10) is considerably higher than the temperature for saturated liquid (pos 42). This results in a relatively big quantity of saturated vapour in front of the capillary tube.

In contrast to the results using CFC-12, the temperature distribution during evaporation is constant using KCD9439A. The reason for this is the pressure losses in the evaporator compensating for the above mentioned temperature glide.

5.2.2 Continuous Operation. Test Results.

The test results from this test is shown in table 3.

The observations from the "no load pull down test" also occur during the "continuous operation test". The capacity of the condenser are not utilized, since the condenser outlet temperature does not reach the saturated liquid temperature equivalent to the measured condensing pressure using KCD9439A. It can also be seen that the end evaporating temperature does not reach the same level using KCD9439A compared to CFC-12. The result leads to higher average air temperature because of a smaller capacity using KCD9439A.

Table 3. Continuous operation. Test results.

<table>
<thead>
<tr>
<th>Point of measurement</th>
<th>CPC-12</th>
<th>KCD9439A</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient temperature</td>
<td>42°C</td>
<td>32.0</td>
</tr>
<tr>
<td>Compressor House temperature</td>
<td>5°C</td>
<td>32.0</td>
</tr>
<tr>
<td>Discharge temperature</td>
<td>42°C</td>
<td>96.0</td>
</tr>
<tr>
<td>Condensing pressure</td>
<td>42°C</td>
<td>98.0</td>
</tr>
<tr>
<td>Equivalent liq. temperature</td>
<td>42°C</td>
<td>11.24</td>
</tr>
<tr>
<td>Equivalent vap. temperature</td>
<td>10°C</td>
<td>11.63</td>
</tr>
<tr>
<td>Outlet condenser temperature</td>
<td>40°C</td>
<td>46.5</td>
</tr>
<tr>
<td>Suction pressure</td>
<td>41°C</td>
<td>46.5</td>
</tr>
<tr>
<td>Equivalent liq. temperature</td>
<td>41°C</td>
<td>46.5</td>
</tr>
<tr>
<td>Equivalent vap. temperature</td>
<td>41°C</td>
<td>46.5</td>
</tr>
<tr>
<td>Inlet evaporator temperature</td>
<td>14°C</td>
<td>46.5</td>
</tr>
<tr>
<td>Outlet evaporator</td>
<td>16°C</td>
<td>46.5</td>
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<tr>
<td>Heat exchanger temperature</td>
<td>20°C</td>
<td>46.5</td>
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<tr>
<td>Maximum air temperature</td>
<td>6.1</td>
<td>46.5</td>
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<tr>
<td>Minimum air temperature</td>
<td>6.1</td>
<td>46.5</td>
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<tr>
<td>Average air temperature</td>
<td>6.1</td>
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</tr>
<tr>
<td>Power consumption</td>
<td>6.1</td>
<td>46.5</td>
</tr>
</tbody>
</table>

5.2.3 Thermostatic Operation. Test Results.

The following tests were made with the thermostat in the positions warm mid and cold. The results provide the basis to compare capacity and power consumption etc. using KCD9439A respectively CPC-12. See table 4.

Table 4 shows that at maximum -18°C storage temperature the capacity of the compressor is 19% lower and the power consumption 7.5% higher when using KCD9439A in the test appliance.

This is not in accordance with the above shown calorimetric measurements. One of the main reasons to this is that the chosen arithmetic suction pressure by the calorimetric measurements does not correspond to application measurements.

The relatively high pressure fall, occurring in the evaporator in question, even contributes to limiting the
Table 4. Thermostatic operation, 25°C ambient temperature. Test results.

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>CFC-12</th>
<th>KCD9439A</th>
<th>CFC-12</th>
<th>KCD9439A</th>
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<tbody>
<tr>
<td>Thermostatic position</td>
<td>Warm</td>
<td>Mid</td>
<td>Cold</td>
<td>Warm</td>
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<tr>
<td>Test package temperature</td>
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<td>-20.1</td>
<td>-27.4</td>
<td>-17.0</td>
</tr>
<tr>
<td>Running time %</td>
<td>44.2</td>
<td>55.2</td>
<td>84.1</td>
<td>52.7</td>
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<tr>
<td>Periods/h</td>
<td>4.2</td>
<td>3.9</td>
<td>0.70</td>
<td>3.1</td>
</tr>
<tr>
<td>Power consumption kWh/24h</td>
<td>1.66</td>
<td>1.94</td>
<td>2.48</td>
<td>1.79</td>
</tr>
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</table>

Temperature rise across the evaporation, with the consequence of a limited reduction in the capacity of the evaporator.

As the pressure fall of most evaporators for household refrigerators and freezers is considerably smaller, the temperature glide will give a considerable temperature rise during evaporation. This will result in a reduced evaporator capacity and in most cases an unfavourable temperature distribution in the appliances.

In most of the refrigerators and freezers the temperature level is regulated by an evaporator thermostat placed on the evaporator. The above mentioned temperature distribution during evaporation leads to a reduction in regulation properties.

In refrigerators/freezers with two evaporators connected in series this will mean that the temperature difference between the refrigerating and the freezing sections will be radically changed, resulting in difficulties satisfying the requirement of standards to storage temperatures.

6. CONCLUSIONS

Measurements have been made of the ternary blend KCD9439A with the purpose of examining its applicability as a drop-in refrigerant in connection with the future servicing of CFC-12 appliances.

* Operating with KCD9439A prevents the use of mineral oils as lubricant due to a higher end compression temperature.

* The temperature glide caused by using KCD9439A means that the conventional reference conditions are not unambiguous.

* On account of other vapour pressures when using KCD9439A, the compressor bearings, valves etc. are more heavily loaded than when using CFC-12. The demand to motor torque increases by approx. 5% compared to CFC-12.

* Calorimetric measurements shows unchanged or better results with regard to capacity and COP using KCD9439A instead of CFC-12. Application tests show 19% lower capacity and 7.5% higher power consumption using KCD9439A. This means that the chosen test conditions with KCD9439A by calorimetric measurements may not be correct.

* A general evaluation of the applicability of KCD9439A as substitute when servicing CFC-12 appliances shows according to our test results that this will lead to changes of regulation properties, reduced capacity and higher power consumption.

* Reliability and lifetime using ternary blends have not been dealt with in this paper. Test results are presented in reference /4/.
REFERENCES


