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STAGED EVAPORATION SYSTEM FOR REFRIGERANT BLENDS WITH LARGE TEMPERATURE GLIDE

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ABSTRACT

The development of refrigerant blends with low GWP and moderate flammability is important to lower the environmental impact of the refrigerant use. With the ban of chlorinated substances, the possible choice of refrigerants is limited. The paper presents different possible blends (using CO₂) with refrigerants as R-152a or R-32, and also R-134a. Many of those blends exhibit large temperature glides during the phase change (in the range of 20 to 30K). Such large temperature glides lead to irreversibilities at the heat exchangers. The average condensing temperature is higher compared to usual refrigerant blends (R-407C or R-410A) and the average evaporating temperature is lower, so the COP is also lower.

The new system design develops a staged evaporation in order to limit drawbacks of such large temperature glides. The new system has been tested on a test bench and compared to a R-407C system. The new system has shown higher energy performances for some of the possible blends, and seems to be a promising design for heating as well as for cooling modes with a lower environmental impact due to the average GWP of the possible blends.

1. INTRODUCTION

The Montreal Protocol has led to the ban of chlorinated molecules. HFCs have been introduced to replace CFCs and HCFCs. Due to their possible impact on climate change related to their GWP, research is needed to find blends with low GWP but maintaining or more likely improving energy efficiency of refrigeration systems.

Constraints have to be considered for any new molecules selection; toxicity and flammability are key issues for easy handling of refrigerants, so as it is well known the list of possible refrigerants or refrigerant blends with low GWP is very limited. Blends of R-744 (CO₂), with HFCs: R-152a, R-32 or R-134a are analyzed in this paper as possible alternatives. These blends exhibit large temperature glides because of the significant difference of the critical pressure of CO₂ (31°C) compared to the 3 studied HFCs. High temperature glide refrigerant mixtures increase irreversibility in current heat exchangers if the fluid on the other side of the heat exchanger wall cannot be cooled (or heated) along also a large glide of temperature. Additional irreversibilities are generated and the global energy efficiency of system using large temperature glide refrigerant blend is lower compare to pure refrigerant systems.

2. REFRIGERATION CYCLE WITH STAGED EVAPORATION

For a well designed air condenser working with a pure refrigerant the difference of temperature between the air at the condenser inlet and the sub cooled liquid varies typically from 7 to 10K with a 2K sub-cooling. For a mixture of R-744/R-152a (0.20/0.80 %mass) the difference between inlet air temperature and the average condensing temperature is superior to 17K implying a relative higher pressure ratio, and irreversibility generation.

A new cycle is presented in order to mitigate the drawbacks related to high glide temperature blends in heat exchangers. The new cycle takes also advantage of the possible composition control of the circulating composition. The new system is named Staged Evaporation and Enhanced Sub-Cooling (SEESC heat pump) (Rached, 2003; Clodic, 2003), which can run either in cooling or heating mode. The first target application is residential heating and cooling. The cooling mode is presented in this paper.

In cooling mode indoor units IU1 and IU2 (see Figure 2) run as evaporators and outdoor units OU1 and OU2 as condensers. For the sake of clarity heat exchangers are presented as separated entities as well the compressors. For the test bench also, all heat exchangers are separated units for easiness of measurements but to save costs and
materials, both indoor and outdoor units could be realized as a single heat exchanger with 6 ports. In the same way 2 compressors are represented in Figure 2 and two compressors are also used in the test bench but a single one with an intermediate pressure port can replace them.

The SEESC cycle is now described: enhanced sub-cooling is realized in the sub-cooler (S-C) by the evaporation of the mass flow rate (MFR) \( m_2 \), which is a fraction of the total MFR \( m_1 \) leaving the sub-cooler (Figure 2). The remaining MFR \( m_3 \) \((m_3 = m_1 - m_2)\) is expanded by E-V1 in the first partial evaporator IU1. MFR \( m_3 \) is leaving IU1 in two-phase flow. In the separator (S), the liquid and vapor phases are separated in a vapor MFR \( m_5 \) and in a liquid MFR \( m_4 \), which is expanded in IU2.

MFR \( m_4 \) is compressed by compressor 1 (COMP 1), mixed to \( m_5 \) and \( m_2 \) mass flow rates at point 2 and the total reconstituted MFR \( m_1 \) is compressed by the second compressor (COMP 2).

MFR \( m_1 \) is desuperheated, then partially or totally condensed in the 2 outdoor units OU1 and OU2 (evolution 3 to 4 in Figure 1), then the end of the condensation and/or the sub-cooling occur in S-C (evolution 4 to 5). Depending on outside temperature and the needed cooling capacity, condensation in the two outdoor units could be only a partial condensation. In order to limit the temperature glide in those heat exchangers the sub-cooler SC achieves the condensation.

The MFR \( m_2 \) is expanded by E-V3 (evolution 5 to 13) at the intermediate pressure. \( m_2 \) is totally or partially evaporated in S-C before to be sucked in 2 by compressor 2. The refrigerant state in 2 is controlled by E-V3 in order to control the temperature at the discharge port of COMP2 (point 3).

\( m_3 (m_1 - m_2) \) is expanded by E-V1 at the intermediate pressure in the first evaporator IU1 (evolution 5 to 8), and partially evaporated (evolution 8 to 9). E-V1 controls the vapor quality at IU1 outlet and so the compositions of the vapor and the liquid phases.

The MFR \( m_2 \) in liquid phase (rich in high boiling components) is leaving S in state 11, and is expanded by E-V2 (evolution 11 to 12) at the low pressure of the cycle and evaporates in IU2 (evolution 12 to 1). Then MFR \( m_4 \) is sucked by COMP1, which compresses only \( m_4 \). \( m_4 \) is blended at the intermediate pressure with \( m_5 \) and \( m_2 \) to build up again \( m_1 \) compressed by COMP2.

The SEESC concept permits to take advantage of new scroll compressors that have 2 suction ports: one at the usual evaporating pressure, and the other one at an intermediate level, usually used for enhanced sub-cooling. In the proposed concept, this intermediate port is not only used for enhanced sub-cooling but also to evaporates at 2 levels of pressure in order to realize a staged evaporation of zeotropic blend with large glide of temperature. This
double stage evaporation decreases the temperature difference between the refrigerant blend and the external medium (air or water) in the evaporators permitting to limit irreversibilities.

3. QUALITY CONTROL AND COMPOSITION CONTROL

The separation process, between the two evaporators, increases the low volatile component composition (R-152a or R-134a) through the second evaporator IU2, which in turn decreases temperature glide and heat exchange irreversibility.

![Diagram of principle of staged evaporation and phase separation.](image)

Expansion valves E-V1 and E-V2 operations control MFRs \( m_3 \) and \( m_4 \) in both heat exchangers IU1 and IU2. This MFR control modifies the quantity of liquid stored in the separator S. When E-V2 is nearly closed, the liquid level in the separator S increases, which implies that the vapor volume decreases in S and so increases high volatile component (R-744) circulation composition.

E-V1 controls both MFR \( m_3 \) and the vapor quality at IU1 outlet. Associated with the control of E-V2, as indicated previously, the quantity of liquid in the separator S will vary. For a large temperature glide blend composed of R-744 and R-152a in nominal mass composition of 0.20/0.80 \%, Table 1 shows that depending on the vapor quality \( Q \), the vapor composition \( y_{R152a} \) varies from 35 to 78 \% when the quality is varying from 10 to 90 \%. Those variations indicate that the association of the control of MFR \( m_3 \) by E-V1 and the separation process in S permit to change rapidly the circulation compositions at the intermediate and at the lower pressures of the circuit.

<table>
<thead>
<tr>
<th>( Q )</th>
<th>0.1</th>
<th>0.25</th>
<th>0.5</th>
<th>0.75</th>
<th>0.9</th>
</tr>
</thead>
<tbody>
<tr>
<td>( x_{R152a} )</td>
<td>0.84</td>
<td>0.89</td>
<td>0.93</td>
<td>0.95</td>
<td>0.96</td>
</tr>
<tr>
<td>( y_{R152a} )</td>
<td>0.35</td>
<td>0.47</td>
<td>0.64</td>
<td>0.74</td>
<td>0.78</td>
</tr>
</tbody>
</table>

E-V1 and E-V2 are used for MFR modulation and capacity control. Experimentations have demonstrated a 20 \% variation in capacity with the described system.

Because of the composition difference between the liquid and the vapor phase, the less volatile component circulation compositions are higher in the second evaporator and the temperature glide decreases through this evaporator.

**Enhanced separation system**

To enhance the composition separation process, an additional receiver is connected to the liquid separator S. This receiver (REC2) is used to store liquid rich in low volatile components. The consequence is a rapid increase of their circulation composition in high volatile components, which permits to increase the refrigerating capacity.
The system operation is as follows:
In normal mode, the ON/OFF valves 1 and 2 are closed. The receiver volume REC2 is in vapor phase.
In separation mode, ON/OFF valve 2 is opened, E-V2 is closed, and compressor 1 is stopped.
In mixing mode, the ON/OFF valve 1 is opened, the liquid receiver is drained to IU2 until the normal mode is reached.

Increasing high volatile component composition is specially interesting in winter time at low outdoor temperature when significant increase of the heat pump capacity is needed. Previous papers (Kasuka, 2000) and (Masatoshi et al, 1994) have indicated the interest of composition variation even with blends such as R-407C.

4. FIRST TEST RESULTS AND SYSTEM PERFORMANCES

The SEESC system has been tested using several refrigerant blends R-407C as a reference, blend 1 (R-744/R-152a; 0.20/0.80), and blend 2 (R-744/R-152a/R-134a; 0.095/0.32/0.585).

OU1 and OU2 are fin-and-tube air heat exchangers and IU1 and IU2 are plate water heat exchangers. Two scroll compressors of 8 and 12 m$^3$/hr are used for the staged compression. All expansion valves are electronic expansion valves. The liquid-vapor separator has been realized by the laboratory as well as the circuit.

The air-to-water heat pump is installed in a climatic chamber and a dedicated unit on the air side permits to balance exactly either the heat released by the outdoor unit in cooling mode or the heat absorbed in heating mode. On the water loop, also a heating-cooling unit is installed for energy balancing.

Thermocouples, PT100, pressure sensors, 3 flowmeters and a wattmeter are installed in order to make mass and energy balances on the refrigerant, air and water circuits. All sensors and also expansion valves are connected to a supervision system where all data are recorded and also the control system permits to control the opening of the expansion valves.

The baseline is a R-407C residential heat pump with a cooling capacity of 12 kW at the reference point T1 of ISO 5151. The reference system is running with a single scroll compressor with a volumetric flow rate of 19.4 m$^3$/hr. For the SEESC system, the two scroll compressors have been chosen in order to have the same total volumetric flow rate.

Table 2: Result comparison between the baseline and the SEESC systems.

<table>
<thead>
<tr>
<th>Mixture</th>
<th>Tair (°C)</th>
<th>Mean Twater (°C)</th>
<th>Comp. (kW)</th>
<th>Capacity (kW)</th>
<th>COP High</th>
<th>Mean</th>
<th>Low</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-407C</td>
<td>30</td>
<td>10.25</td>
<td>6.90</td>
<td>12.6</td>
<td>1.82</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Blend 1</td>
<td>29</td>
<td>16.5</td>
<td>4.80</td>
<td>12.7</td>
<td>2.65</td>
<td>2.320</td>
<td>0.80</td>
</tr>
<tr>
<td>Blend 2</td>
<td>30</td>
<td>10.5</td>
<td>3.87</td>
<td>11.0</td>
<td>2.85</td>
<td>1.877</td>
<td>0.57</td>
</tr>
</tbody>
</table>
For the test conditions, it can be seen that the coefficient of performance is significantly improved by the SEESC system, specially with Blend 2 where the improvement is more than 50%. Those results are very first ones, just to confirm the interest of the new concept. A number of developments are necessary and also the design of air heat exchangers needs to be significantly modified in order to take advantage of the temperature glide. The tests have been performed on the same heat exchangers as the ones of the baseline system, with no significant modification except the separation into OU1 and OU2.

6. CONCLUSIONS

The number of possible molecules to create low GWP refrigerant blends neither flammable nor toxic is limited. To broaden the possible choice of refrigerant blends, the introduction of CO$_2$ as a blend component implies usually large glides of temperature during condensation and evaporation. The blends used through this paper show significant temperature glides. If they are to be directly used in current heat pump systems, the energy performances will be lower compare to current R-407C systems. The challenge is to design a system that permits to reach higher energy performances with high temperature glide blends.

Two solutions are introduced at the evaporator and condenser sides, and for each operating mode (cooling or heating), in order to decrease the mean temperature difference between the two media through the heat exchangers.

At the evaporator level, the staged evaporation in association with a phase separator and the use of a compressor with an injection port at an intermediate pressure permits to have:
1. a staged evaporation with two different mixture compositions in each of the evaporator;
2. a vapor mass flow rate rich in high volatile components absorbs heat from the cold source at an intermediate pressure (600–700 kPa) and so lowers the compression ratio;
3. a liquid mass flow rate rich in low volatile components expanded at the low pressure of the system;
4. a lower mean temperature difference between the two medium through the heat exchangers.
5. a cooling capacity control by controlling the quality of the main mass flow rate and so the separation process in the separator S.

At the high-pressure side, the sub-cooler at the condenser outlet exchanges heat between the condenser outlet mass flow rate and the mass flow rate expanded from the high-pressure to the intermediate pressure. This sub-cooler permits to:
1. carry on the condensation process when the sink temperature is higher than the saturated liquid temperature and allows the system operation at a lower condensation pressure;
2. sub-cool the outlet condenser mass flow rate and decrease the mixture mean condensation temperature.
3. lower the discharge temperature of the compressor by injecting fluid at a moderate temperature (<25°C) (Brand et al., 2000).

For a “reversible” heat pump system, the staged evaporation and separation system can be built in a compact way in order to simplify the piping configuration circuit.

For defined outdoor and indoor conditions, mixture components, compressor volumes, compression ratios, heat exchangers need be optimized to obtain a high performance heat pump system using the staged evaporation and enhanced sub-cooling concept.

In summary, it is possible to take advantage of large temperature glide blends by staging evaporation and making a large sub-cooling using the intermediate pressure suction port of staged scroll compressors.

ACKNOWLEDGEMENTS

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