R32 Scroll Compressors Technology

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ABSTRACT

The purpose of the work is to describe technology used to design a scroll compressor with R32 refrigerant. Study what are the main differences and their implications compared to R410A refrigerant scroll compressors. And also study from mechanical scroll compressor design point of view up to its system integration, what are the main design changes taken in account R32 properties, what are the main options to handle high discharge refrigerant temperature, what are the advantages and consequences of each options, what solution will be chosen.

1. INTRODUCTION

At the end of 1995, the CFC was completely phased out in developed countries, and HCFCs refrigerants are limited for use to protect ozone layer according to Montreal Protocol. In September 2007, the 19th anniversary of the Montreal Protocol, governments agrees to advance the final phase out date for HCFCs. The replacement of refrigerant for the environment protection is always happening. Up to now, R134a, R407C, R410A and R22 are the refrigerant most widely used in the world on A/C application R22 is only limited in developing countries and the other three have more and more criticisms with the current environmental trends because of high GWP.

In general, the higher the non-flammability of a refrigerant is, the larger its GWP and the lower its thermodynamic efficiency (Ryuzaburo Yajima et al., 2000). Considering environment protection, there are several refrigerant candidates: R1234yf, R32, R290 and CO2. From Table 1 and Table 2, the R32’s GWP level is moderate (one third of the R410A GWP), but LCCP is the lowest. The LCCP evaluates the complete impact to environment from manufacturing to application of refrigerant and to the end of refrigerant system. It’s a very important evaluation indicator of refrigerant. In terms of flammability, toxicity and thermodynamic properties (thermal conductivity, vaporization heat and specific volume), R32 is also a good choice. Put together afore-mentioned reasons, R32 could be a very good substitute for a long term in future. This paper emphatically studied the feasibility use of R32 application and the consequences bring to the changes of scroll compressor as well as lubricant oil and refrigeration system.

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>ODP (R11=1.0)</th>
<th>GWP (IPCC AR4 CO2=1.0)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R22</td>
<td>0.055</td>
<td>1810</td>
</tr>
<tr>
<td>R410A</td>
<td>0</td>
<td>2088</td>
</tr>
<tr>
<td>R407C</td>
<td>0</td>
<td>1770</td>
</tr>
<tr>
<td>R32</td>
<td>0</td>
<td>675</td>
</tr>
<tr>
<td>R1234yf</td>
<td>0</td>
<td>4</td>
</tr>
<tr>
<td>R290</td>
<td>0</td>
<td>6.3</td>
</tr>
<tr>
<td>CO2</td>
<td>0</td>
<td>1</td>
</tr>
</tbody>
</table>
**2. R32 THERMODYNAMIC PROPERTIES**

For a refrigerant, thermodynamic properties are very important. They have a big impact on compressor and system performance and also have some requirements and restrictions to mechanical and chemical parts. Table 3 shows a comparison between R410A and R32.

**Table 3: Refrigerant thermodynamic properties**

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>R410A</th>
<th>R32</th>
</tr>
</thead>
<tbody>
<tr>
<td>Critical Temperature °C</td>
<td>69.7</td>
<td>78.4</td>
</tr>
<tr>
<td>Boiling Point °C</td>
<td>-51.6</td>
<td>-51.7</td>
</tr>
<tr>
<td>Heat of vaporization at boiling point(kJ/kg)</td>
<td>275</td>
<td>384</td>
</tr>
<tr>
<td>Thermal conductivity liquid(54.4°C) mW/(m.k)</td>
<td>0.07</td>
<td>0.11</td>
</tr>
<tr>
<td>Thermal conductivity vapor(7.2°C, 10bar) mW/(m.k)</td>
<td>0.014</td>
<td>0.012</td>
</tr>
<tr>
<td>Saturated pressure at 50°C (bar)</td>
<td>30.5</td>
<td>31.4</td>
</tr>
<tr>
<td>Specific volume vapour (18.3°C, 10bar) m³/kg</td>
<td>0.028</td>
<td>0.040</td>
</tr>
<tr>
<td>Liquid specific heat (25°C)</td>
<td>2275</td>
<td>2630</td>
</tr>
<tr>
<td>Molar mass (Kg/mol)</td>
<td>0.07</td>
<td>0.05</td>
</tr>
<tr>
<td>Volumetric capacity Vs. R410A (Thomas J LECK, 2010)</td>
<td>0</td>
<td>79.7%</td>
</tr>
<tr>
<td>Ratio of specific heat vapor (18.3°C, 10bar)</td>
<td>1.34</td>
<td>1.44</td>
</tr>
<tr>
<td>Specific heat at constant pressure (18.3°C, 10bar) J/kgK</td>
<td>1070</td>
<td>1171</td>
</tr>
</tbody>
</table>

- Critical temperature. R32 has a higher critical temperature which leads to less compressor superheat in evaporator and less flash gas losses and so the efficiency will be higher.

- Boiling point. R410A and R32 have a similar boiling point. The higher the boiling point, the lower the fluid's volumetric cooling capacity will be. So that’s a good point.

- Latent heat of vaporization. R32 heat is greater than R410A so that the required mass flow rate per unit cooling capacity is smaller and the COP is higher. But generally, higher latent heat of vaporization refrigerant cause high discharge temperature which is not wanted and should be controlled.

- Thermal conductivity. R32 conductivity is higher than R410A in liquid phase. This can help to reduce the heat transfer area of evaporator.

- Ratio of specific heat. This can impact the discharge temperature too. So R32 has got a higher discharge temperature.
Volumetric cooling capacity. R32 has got a significant high volumetric cooling capacity compare with R410A, which can help to reduce the system pipe size and increase the efficiency.

Compare to R410A, R32 has a better system efficiency and needs a smaller refrigerant charge but the compressor discharge temperature is higher. The direct and severe problem is the oil degradation which can cause various compressor failures like bearing seizure, low system and compressor efficiency. So the discharge temperature should be controlled at an acceptable level. As R32’s temperature and pressure are higher, compressor load will be increased. Moreover, the specific volume of R32 is higher than R410A, so the motor may not be well cooled at higher load conditions.

With high load and high temperature, compressor reliability will be affected compared with R410A equipment. Scroll sets should be able to endure more pressure difference especially for the involute close to discharge port where the condition is more severe. Compressor shell’s strength should be increased and the temperature of the shell should be also paid attention to avoid burst or scalding risk. Plastic parts like sealing parts should also be checked in case of over deformation or dissolution due to high temperature.

R32 has a higher ratio of specific heat, if keep the same volumetric ratio of involute, the pressure ratio will be higher than R410A. A compressor’s pressure ratio is optimized for the nominal point, so the volumetric ratio should be adjusted. As the discharge temperature is higher, this should be controlled to a safety level like 135°C. So the operating map will be reduced in left corner area (see Figure 1). To keep the same envelope, compressor needs some discharge temperature control solutions like liquid injection, vapor injection or wet suction.

![Figure 1: R32 Operating Map](image)

3. COMPRESSOR TECHNOLOGY DEVELOPED FOR R32

To develop a compressor with R32, the most cheaper and effective way is to work on the current platform. According to the difference in thermodynamic properties, compressor designed for R410A application may need some adjustments to get a better performance and appropriate application conditions if R32 refrigerant is applied.

If applying R32 refrigerant to a R410A system without changing the compressor displacement, then the system capacity will increase by about 10% compared to R410A refrigerant. In this case, evaporator and condenser need to be enlarged to adapt to higher capacity. As the pressure and ratio of specific heat for R32 is higher than R410A, the pressure in scroll pocket will be higher than R410A, with the same scroll sets as R410A; loads will be higher than R410A. Figure 2 shows that all R32 loads are higher than R410A and moved to right side. More loads create more heat and R32 could not well cool down motor. So the motor and compressor efficiencies will be worse than R410A.
To keep the same cooling capacity with the same application, the compressor displacement should be reduced which can be realized by reducing the generating circle radius, the number of involute wraps or the flank height. Changing the number of involute wraps will impact the volumetric ratio which is optimized by nominal condition. This is not a good way.

Decrease of flank height is the easiest way because this has less impact to production. Figure 3 shows that the axial gas load is increased as the pressure in scroll pocket is higher than R410A during compression. There will be more losses on thrust bearing. Tangential gas load keeps on the same level.

Compressor torque will be increased too. From the two above figures, it can be seen that the increase of load is not too much, so the impact on efficiency can be controlled.

\[ F_{tg} = \Delta P \cdot A = \Delta P \cdot R_g \cdot H \cdot \left(2\Phi + \pi\right) \]  

Where:

- \( F_{tg} \): tangential force of scroll sets
- \( P \): pressure in scroll chamber
- \( A \): area
- \( R_g \): generating circle radius
- \( H \): scroll sets height
- \( \Phi \): angle

\( \Delta P \) is the same, if the decrease of height is larger than the decrease of \( R_g \), \( F_{tg} \) will be increased (see Figure 4). Compressor torque will be increased too. From the two above figures, it can be seen that the increase of load is not too much, so the impact on efficiency can be controlled.
In consideration of scroll strength, the pressure of R32 is close to R410A and a little higher. But the ratio of specific heat of R32 is higher than R410A. So the isentropic exponent is higher too. This will result in a smaller pressure difference at the beginning of involute (after compression) in the left top corner of operating map where the pressure difference is decreased about 10% than R410A. The high-low separator like O-ring, pressure separator cap should be re-qualified. Because the absolute pressure difference for R32 on those parts is higher than for R410A and increased by about 3.8%.

4. HIGH DISCHARGE TEMPERATURE MANAGEMENT

As above-mentioned, the main concern for R32 application on A/C system is slight flammability and high discharge temperature.

Even though the flammability properties of R32 are quite mild as compared to other alternatives such as propane, it still is classified as A2L by ASHRAE Std 34. Safety standards including UL 250/471, IEC 335-2-24, and EN378, likely will need to be revised to give guidance for how to safely handle this new class of refrigerant.

High discharge temperature may cause carbonization of lubricant oil and yield the whole compressor spoiled. So in order to adopt R32 refrigerant in different applications, it is necessary to take some measures to decrease the discharge temperature.

As is well-known, vapor injection can be used to increase system capacity and performance. Also it has the function to decrease discharge temperature. As shown in Figure 5 a portion of the condensed liquid with high temperature and high pressure is expanded through an expansion valve into the heat exchanger, which acts as sub-cooler. Superheated vapor with low temperature and low pressure is then injected into scroll directly, and mixed with

**Figure 4:** Gas force in involute

<table>
<thead>
<tr>
<th>Model</th>
<th>Load (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AXIAL GAS</td>
<td>0</td>
</tr>
<tr>
<td>RADIAL GAS</td>
<td>5000</td>
</tr>
<tr>
<td>TANGENTIAL GAS</td>
<td>10,000</td>
</tr>
</tbody>
</table>

**Figure 5:** Vapor Injection system diagram

**Figure 6:** Liquid Injection system diagram

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intermediate compressed gas with high temperature. Thus the discharge temperature can be decreased by mixture indirectly. And the mass flow rate is modulated by superheat of heat exchanger outlet through expansion valve.

Another option, liquid injection could be a better solution for high discharge temperature directly. As shown in Figure 6, portion of the condensed fluid “i” is injected into scroll through an electronic injection valve, which will absorb the heat of intermediate compressed gas to vaporize. Thus the discharge temperature can be lower down effectively. And the injected mass flow rate can be easily modulated by injection valve.

Compared to vapor injection, the liquid injection system takes much more advantages, e.g. less complexity, lower applied cost, large operation map, easier assembly. Liquid injection can be applied to most of application comparing to vapor injection. On the other hand, liquid injection will not produce any system capacity improvement.

Wet suction also can be one of technologies to decrease temperature, however its application is limited and affect compressor reliability since it is to be achieved by controlling compressor suction dryness.

5. LUBRICATION

Lubricant selection is a key task, because lubricant must satisfy the compressor specification and the system specification. Lubricant selection is done to keep same level of quality and reliability as compressors using HFC with POE or PVE and HCFC with MO or AKB lubricants.

On compressor side of view, the following tests are proceeded systematically:

− ASHRAE 97 test
− Oil lubricity test
− Miscibility/Solubility
− Compressor component compatibility (motor compatibility and material, gasket, …)
− Compressor reliability (preliminary)
− Compressor reliability
− Additive depletion (if used)
− Compressor performances impact (EER, noise, OCR, …)
− Field tests

Compressor reliability is based on compressor running in extreme situation outside the recommended application range in order to evaluate the impact on it. For example, some of the tests run typical conditions in order to have:

− Higher load on bearing (i.e.: severe summer conditions)
− Higher stress on parts (pressure, temperature) (i.e.: severe winter conditions)
− High lubricant dilution (i.e.: defrost, low superheat…)
− Strong transient case (i.e.: winter start, charge migration …)

Depending on the compressor technology, different life testing sequences are used to cumulate the potential damage due to each life test mode conditions. On tested compressors, performance changes carry out an oil analysis, a visual inspection and a parts measurement.

Most of the R32 compressor designs are derived from R410A compressor. Tests carried out on our lab for performance and reliability evaluation show that the developed lubricant for present R410A compressors brings the performance and reliability targets level on R32 compressors. Lubricant on sliding parts is behaving satisfactory.

On system point of view, the oil flowing with the refrigerant must not be trapped in system high pressure area (i.e.: liquid receiver, piping, condenser conduit …) or in low pressure area (i.e.: evaporator, suction gas pipe …). If there is a bad oil return to the compressor this could affect the compressor mechanical reliability. Miscibility curves give a good indication of the potential behavior for the lubricant on system with R32 (Figure 7). It’s identified that the
miscibility of POE with R32 behaves similarly to mineral oil with R22 on which we have more than 20 years’ experiences.

It is obvious that in some applications the present lubricant is satisfactory in most of unit running time, but some transients or applications conditions could lead for example to a bad oil return and can impact dramatically system reliability. Nevertheless it should be possible to use existing lubricants with limited miscibility if the unit system design integrated these conditions (i.e.: piping sizing and design, no dead volume in heat exchanger, special design for liquid storage …). Then the OEM has to check that in all situations the oil flow back to the compressor oil sump is secured. This is not new, for example this was the situation after the Montreal protocol, when R407C unit using alkylbenzene lubricant or for the use of R502 with mineral oil wherein the miscibility was limited.

In other hand the screening evaluation shown that some lubricants qualified for applications using R134a, R404A, R407C and R410A, doesn’t satisfy the miscibility expected. This shown that the R32 refrigerant could behaves differently when it is used alone versus when it is used on blended refrigerant like R407C [R32/R125/R134a (23/25/52)] or R410A [R32/R125 (50/50)](CNAM IFFI, 2012). The miscibility curve has shown also a phase separation on higher evaporating temperature which is not expected. These criterias have been chosen in order to satisfy the most critical case comparing the behavior based on existing applications experience, like R22-MO and R410A –POE.

R32 liquid density is lower than for POE mixture at low temperature and higher at high temperature. Balance is around 20°C. This will have been included in miscibility selection process (Figure 8) (CNAM IFFI, 2012).

![Miscibility Curves](image)

**Figure 7:** Miscibility curves

In other hand VLE curves (curves showing the viscosity, solubility versus pressure and temperature) are used to find the best compromise between oil return and mechanical lubricant specifications. Tribology sliding conditions like contact pressure, speed, materials and surfaces characteristic motivate the choice of a lubricant with or without additive. To stay on sustainable environment, additive selection must comply with the Danfoss Negative (banned substances) list and be in line with Danfoss grey list. (Not recommended substances to use)

### 6. CONCLUSION

Compared with the other candidate refrigerants, R32 is a quite good refrigerant on both environment and mechanical point of view. R32’s flammability is slight and its thermodynamic properties are good for COP improvement and the increase of pressure and loads is not too much and high discharge temperature can be controlled by liquid injection. Moreover, the strength requirement of scroll sets can be reduced due to smaller pressure difference in scroll sets. But lubricant should be re-qualified on miscibility and tropology side. Put together the above describes, R32 is likely to become a major refrigerant in the future air-condition domain.

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