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Refrigeration Cycle and Compressor Performance for Various Low GWP Refrigerants

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ABSTRACT

The recent meetings of the global leaders in COP21 have gathered global consensus to fight the global warming. One of the important approaches in aiding toward fighting global warming is to use environmental friendly refrigerants. Recently a few new low GWP refrigerants have been introduced. In this paper, we will discuss the energy efficiencies of some new low GWP refrigerants as compared to the existing ones.

Cycle performance calculations assuming an ideal vapor compression cycle from a pressure-enthalpy diagram and those from a comprehensive compressor mathematical model will be shown and compared. Refrigerants which are considered are R32, R134A, R404A, R407C, R1234ZE and R1234YF, and the results quoted will be normalized using data from R410A, which is currently a widely used refrigerant.

The results shows that, when compared to the predictions from the compressor mathematical model, the results from the P-h diagram using the ideal cycle may overestimate the COP by up to 2 times, and this is caused by the up-to 50% of underestimation on the compressor work input.

1. INTRODUCTION

The COP21 United Nations climate summit that was held in Paris November 2015, after 13 days of negotiations by negotiators representing 195 countries, reached an agreement on the effort on climate change which covers both the developed and the developing countries. One of the agreements made is to limit the average rise in global temperature to be 2 degrees Celsius above pre-industrial times. One of the causes of the global warming is due to the use of refrigerants, hence many new green and environmentally more friendly refrigerants had been introduced. In this paper, we will compare the energy efficiencies and refrigerant cycle performance when using these new low GWP refrigerants with the existing ones. The basis of the comparison will be the ANSI/AHRI 540 testing standards, which is suitable for positive displacement refrigerant compressors. The testing cycle for the refrigerant will be based on the idealised vapor-compression refrigeration cycle. With these conditions imposed on all the refrigerants tested, the coefficient of performance (COP), cooling capacity, work input required, as well as the working pressure of the refrigerants will be compared and discussed. The effects of these new low GWP refrigerants and their properties on the compressor performance will be evaluated, discussed and compared. Compressor performance parameters such as work input, discharge pressure and temperature, volumetric efficiency, mechanical efficiency will also be shown and discussed.

2. MATHEMATICAL MODEL

There are two approaches used in this paper to compare the performance of these refrigerants. One approach is to use an idealized vapor compression cycle on a P-h diagram and the other is to use a more comprehensive mathematical simulation model for a rolling piston compressor.
This brief description for the compressor mathematical model will be presented below for completeness, detailed model can be found in ref [3]. The compressor used is rolling piston type, the schematic of which is shown in fig. 1. The volume $V(\theta)$ of the chamber of the rolling piston compressor can be expressed in terms of the length of the compressor $l$, radii of cylinder $R_c$ and rotor $R_r$, the rotational angle $\theta$ and vane thickness $t_v$, as below.

$$V(\theta) = f(l, R_c, R_r, \theta, t_v)$$  \hspace{1cm} (1)

The variation of the fluid properties in the chamber can be obtained by applying First Law of Thermodynamics,

$$\dot{E}_{in} - \dot{E}_{out} = \frac{d(mu)_c}{dt}$$  \hspace{1cm} (2)

where $\dot{E}_{in}$, $\dot{E}_{out}$ and $(mu)_c$ the energy in and out of the chamber as well as the change in internal energy of the working chamber, respectively.

Real gas properties of the refrigerant that relates enthalpy $h_c$ of the working fluid with the pressure $P$ and specific volume $\nu$,

$$h_c = f(P, \nu)$$  \hspace{1cm} (3)

The conservation of mass in the working chamber,

$$\sum \frac{dm_i}{dt} - \sum \frac{dm_o}{dt} = \frac{dm_c}{dt}$$  \hspace{1cm} (4)

where $m$ is the mass of the working fluid in chamber and subscript i, o, c represents in, out and chamber respectively.

Assuming that the flow through the valves is steady one-dimensional and adiabatic, the mass flow can be expressed as,

$$\frac{dm_2}{dt} = C_d A_2 \sqrt{2(h_1 - h_2)} v_{s2}$$  \hspace{1cm} (5)

where $C_d$ indicates non-isentropic and flow losses, $A$ is the flow area, $\nu_s$ the specific volume of refrigerant and $h$ enthalpy of the refrigerant. The subscript 1, 2 and s represents upstream, downstream and isentropic conditions respectively. The model also includes kinematic, roller dynamics, thermodynamics, valve dynamics, in-chamber heat transfer, mechanical frictional and lubrication [2,3,4].

The model is written in Fortran programming language, solving simultaneous equations using 4th order Runge-Kunge numerical integration method. The model has been verified using R22 as working fluid operating at -23.3°C and 54.4°C at 2875 rev/min, see fig. 1.

![Figure 1: Comparison between measured and predicted results](image-url)
Fig. 1 shows that the discrepancies between measured values and prediction are within 10%. The model also includes effects of in-chamber heat transfer, internal leakages and frictional losses [3].

3. REFRAIGERANTS AND THEIR PROPERTIES

Properties of the refrigerants such as the GWP, critical temperature and pressure are listed in table 1. As we can observed, the GWP of newer refrigerants are lower, with R1234YF has only 0.2% of that of the R410A. The table also shows that the critical temperatures are all well above the condensing temperature in the applications and hence the refrigeration cycle will operate at the subcritical regions.

In this paper, the performance of various refrigerants shown in Table 1 will be compared using ANSI/AHRI 540 2015 standard operating condition, henceforth refers to as standard condition. Two approaches are used: (i) employing the ideal vapor compression cycle as shown in fig. 2 and, (ii) using a more comprehensive compressor model as described in section 2.

Table 1 Properties of refrigerants

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Refrigerant type</th>
<th>GWP normalised *</th>
<th>Critical Temperature (°C)</th>
<th>Critical Pressure (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R32</td>
<td>HFC</td>
<td>0.32328</td>
<td>78.11</td>
<td>5782</td>
</tr>
<tr>
<td>R134A</td>
<td>HFC</td>
<td>0.68487</td>
<td>101.06</td>
<td>4059.3</td>
</tr>
<tr>
<td>R404A</td>
<td>HFC</td>
<td>1.87835</td>
<td>72.12</td>
<td>3734.9</td>
</tr>
<tr>
<td>R407C</td>
<td>HFC</td>
<td>0.84962</td>
<td>86.14</td>
<td>4639.4</td>
</tr>
<tr>
<td>R410A</td>
<td>HFC</td>
<td>1.00000</td>
<td>71.34</td>
<td>4901.2</td>
</tr>
<tr>
<td>R1234ZE</td>
<td>HFO</td>
<td>0.00287</td>
<td>109.36</td>
<td>3634.9</td>
</tr>
<tr>
<td>R1234YF</td>
<td>HFO</td>
<td>0.00192</td>
<td>94.70</td>
<td>3382.2</td>
</tr>
</tbody>
</table>

*GWP values shown are normalized with respect with R410A

Figure 2 shows the idealized refrigeration cycle on a P-h diagram operating under the standard conditions. Under this condition, the refrigerant enters the compressor at 11°C of super-heat and leaving the condenser at 46°C as saturated liquid with no subcooling. The compression is assumed isentropic and there is no pressure loss at the evaporator, condenser and along the pipelines. The refrigerant will be throttled through the expansion valve and enter the evaporator at 10°C.

![Figure 2: Ideal Refrigeration Cycle on a P-h diagram](image)
4. RESULT AND DISCUSSIONS

Calculations from the idealized cycle on P-h diagram (henceforth refers to as “calculation”) and predictions from the compressor model (henceforth refers to as “prediction”) have been made using operational conditions spelt out in ANSI/AHRI 540 2015. The results are normalized, with respect to R410A, a commonly used refrigerant in the refrigeration system.

Figure 3 shows that all the predicted Coefficient of Performance (COP) are lower than that of the calculated. This is because the prediction includes mechanical losses and the internal leakages. The difference between the two varies significantly from 35% to more than 60%.

Figure 3: COPs of various refrigerants from calculation and predicted

The cooling capacity of the refrigerant depends on the refrigerant mass flow rate. For given displacement volume and operating conditions, the mass flow rate depends on the specific volume of the refrigerant. Fig. 4 shows the specific volume alongside with the mass flow rate. As expected, the higher the specific volume, the lower the mass flow rate. R404A and R410A has similar mass flow rate as they have similar specific volume.

Figure 4: Specific Volume and the predicted mass flow rates

Another factor that will affect the cooling capacity is the latent heat of the refrigerant at the evaporating condition, in this case 10°C. Figure 5 presents the refrigerating capacity, the mass flow rate and the latent heat of the refrigerant at the evaporator condition. It shows that R32 gives the highest refrigerating capacity, which is 10% more than R410A while R1234ZE is the lowest, which is more than 60% lower than R410A.
Figure 5: Variations of mass flow, refrigerating capacity and latent heat at evaporator

The second parameter that affects the COP of the refrigeration cycle is the work input. Fig. 6 shows the comparison between the calculated and the predicted work. As seen from the figure, the predicted work input is higher than that of the calculated and the difference can be easily more than 50%. This is due to flow losses, volumetric and mechanical efficiencies considered in the predicted values.

Figure 6: Variations of work input from calculated and predicted

In the rotary vane compressor, the friction loss occurs mainly at 6 rubbing areas, namely the eccentric and inner surface of the roller, surface between roller and cylinder, eccentric face and cylinder head face, vane tip and roller as well as vane and slot. As seen from fig. 7, for a given refrigerant, when the pressure difference between suction and discharge (P_d-P_s) is high, the contact force between the surfaces will be large and hence results in a higher friction loss.
Figure 7: Variations of $P_d - P_s$ and friction power (predicted)

Figure 8 shows the variation of the predicted mechanical efficiencies for various refrigerants. It shows that R410A and R32 have the highest values. It would expect that for a higher $P_d - P_s$ the mechanical efficiency will be lower, however the influence of the indicated work will also affect the mechanical efficiency.

Figure 8: Variations of $P_d - P_s$, mechanical efficiency (predicted) and indicated work for various refrigerants.

Figure 9 shows that the volumetric efficiency for various refrigerants are very similar to each other, because the refrigerant's experiencing similar suction valve losses with suction heating effect was not considered. The marginal differences in volumetric efficiencies were caused by the differences in operating pressures.

Figure 9: Variations of $P_d - P_s$, mechanical efficiency (predicted) and indicated work for various refrigerants.
Figure 9: Variations of $P_d - P_s$ and volumetric efficiency (predicted)

Figure 10 shows clearly that the torques of the motor shaft for various refrigerants are dependent on the differences between $P_d$ and $P_s$.

Figure 10: Variations of $P_d - P_s$ and average torque (predicted)

5. CONCLUSION

This paper presents comparison of performance for various refrigerants when applied to a vapour compression cycle. Calculations of the performance data assuming a simplified cycle using a textbook based $P$-$h$ diagram (henceforth refers to as calculations) have been presented alongside with the predictions from a more comprehensive compressor mathematical model (henceforth refers to as predictions) together with operational data for condenser and evaporator sides using ANSI/AHRI 540 2015 standard operating condition. The refrigerants tested are R32, R134A, R404A, R407C, R1234ZE and R1234YF. The results show that:

1. Performance data obtained from the textbook based $P$-$h$ diagram using an idealized refrigeration cycle deviated significantly from those predicted using a more comprehensive mathematical model. It is therefore concluded that $P$-$h$ diagram based calculations are not suitable for accurate performance calculation estimation.
2. These “calculations” may over estimate COP values from 30% to 60%, when compared with the more comprehensive COP predications.
3. For a given compressor displacement volume, the cooling capacity for R32 is 10% higher than that of R410A and the lowest is the R1234ZE, which is 60% lower than R410A.
4. P-h diagram calculations underestimated compressor power input by as much as 40% to 80%. For the same compressor and operational conditions, R1234YF requires the least power input, which is 40% lower than R410A, while R32 requires the highest power input of 70% higher than that of R410A.
5. Refrigerant R32 seems to produce the most frictional loss which is due to its large pressure difference between the $P_d$ and $P_s$.
6. The volumetric efficiency of the various refrigerants tested are not differ significantly, this is because the same compressor was used without considering the suction heating effect.
7. The motor torque is significantly dependent on the pressure difference between the $P_d$ and $P_s$.

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