INFLUENCE FACTORS IN PERFORMANCE OF THE TWO-STAGE COMPRESSION TRANSCRITICAL R744 CYCLE

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ABSTRACT

The present study is a theoretical analysis of a two-stage transcritical cooling cycle using R744 (carbon dioxide) as a refrigerant. The effect of the intercooling process on performance in the two-stage transcritical system is subjected to varying pressures in a gas cooler. Performance comparison between the single stage and two-stage cycle is also subject to the same operating conditions. We assessed the effects on system performance of pressure in the gas cooler, isentropic compressor efficiency, amount of intercooling between the two compression stages and discharge temperature of the refrigerant from the gas cooler. Thermodynamic analysis was carried out with CoolPack software, considering an evaporation temperature of -10°C and refrigeration capacity of 5 kW. Results show the coefficient of performance (COP) for the two-stage cycle is superior to the single stage cycle. Two-stage compression was found to reduce energy consumption. On the whole, two-stage compression and the intercooling process are significant in increasing COP for these systems.

1. INTRODUCTION

In the first decades of the twentieth century carbon dioxide (CO₂/R744) was widely used as a refrigerant, primarily in ship refrigeration systems, as well as for air conditioning systems and stationary applications. Alexander Twining was the first to recommend CO₂ as a refrigerant in an 1850 patent. However, the first system to run on carbon dioxide, proposed by Thaddeus Lowe (Kim et al., 2004), only emerged in 1860. Peak use of CO₂ occurred from 1940 to 1955 for refrigerated maritime transport. In Europe, refrigeration machines were limited to using CO₂ due to legal restrictions on toxic or inflammable refrigerants such as ammonia (NH₃) and sulfur dioxide (SO₂). The emergence of CFCs in 1940 led to the gradual disappearance of old fluids (ammonia, carbon dioxide, sulfur dioxide) for most applications. The main reason for the rapid growth in CFC use was the sudden loss of capacity in carbon dioxide-driven systems when ships approached tropical areas where temperatures were higher (Lorentzen, 1995). Meanwhile, ammonia continued to gain ground, dominating the large-scale industrial refrigeration industry

With the advent of environmental problems, identified in 1980, provoked by chlorinated refrigerants, the industry began to research viable alternative substances. The Norwegian professor Gustav Lorentzen believed the use of CO₂ could resurge. A bibliographical review indicated that the primary incentive for the reappearance of R744 as a refrigerant was the first experimental results of an automotive air-conditioning prototype presented by Lorentzen and Pettersen in 1992. Based on these and other findings, interest in R744 as a refrigerant has increased considerably over the years, mainly because of resistance to the fluorocarbon industry. R744 also has excellent ODP (Ozone Depletion Potential) and GWP (Global Warming Potential) characteristics. Among natural refrigerants such as ammonia, hydrocarbons (propane, butane, isobutane) and carbon dioxide, the last is often presented as the best option. This is because hydrocarbons are extremely inflammable and ammonia highly toxic, while R744 is non-toxic and non-inflammable.

R744 is currently applied in two types of refrigeration cycles. The first is known as transcritical, single or two-stage, and is mainly found in “light” commercial refrigeration equipment and automotive air-conditioning. The second is the cascade cycle, combining two simple stage cycles, where R744 is the refrigerant for the low-temperature cycle. Evaporation temperatures vary between -50 and -30°C with condensation between -30 and -10°C. These cycles have significant applications in large industrial
refrigeration systems. All laboratory research based on R744 focuses on small equipment or the “light” commercial segment. This is attributed to test platform size: the usual problems of supplying driving force and thermal load are magnified by the high effect of specific carbon dioxide refrigeration. This means that even when using small test platforms, there are large amounts of heat and energy, and substantial heat volumes are rejected by the cycle. There is a wide range of alternative compressors available to small plants through several manufacturers (Pearson, 2006).

In the present study, the performance of a two-stage compression system with intercooling is investigated under various pressures in a gas cooler, R744 discharge temperatures in the gas cooler, compressor efficiencies, amount of intercooling between compression stages and presence or not of an internal heat exchanger in the cycle. In order to obtain a comparison parameter, a simulation was performed for both types of transcritical cycles: simple and two-stage compression.

2. THEORETICAL MODELING AND SIMULATION

In order to investigate performance in the two-stage compression transcritical refrigeration cycle, the simulation code from CoolPack software (DTU, 2001) was used to calculate desired characteristics for the cycle and obtain relevant thermodynamic parameters (DTU, 2001). The software was developed on the EES-Engineering Equation Solver (Klein and Alvarado, 1995) platform. Thermodynamic properties of R744 in EES are determined by applying the fundamental equation of state for the liquid, developed by Span and Wagner apud Özgur (2008).

In this comparative study of transcritical single and two-stage compression cycles, the following project and performance parameters are considered: gas cooler, discharge temperature of the refrigerant from the gas cooler, isentropic efficiency of the compressor, presence or absence of an internal heat exchanger (SGHX), intermediate pressure, degree of intercooling, compression work, and energy consumption. These are presented graphically to illustrate varying performance trends. Operating parameters used to simulate operating conditions for the system are presented in table 1, using reference parameters obtained by Özgur (2008).

Table 1. Operational parameters for the refrigeration system.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Evaporation temperature</td>
<td>-10°C</td>
</tr>
<tr>
<td>Refrigeration capacity</td>
<td>5 kW</td>
</tr>
<tr>
<td>Isentropic efficiency of compression</td>
<td>60%</td>
</tr>
<tr>
<td>Pressure in the gas cooler</td>
<td>95 bar</td>
</tr>
<tr>
<td>Discharge temperature of the gas cooler</td>
<td>32°C</td>
</tr>
<tr>
<td>Degree of Intercooling</td>
<td>25°C</td>
</tr>
<tr>
<td>Thermal efficiency of the internal heat exchanger</td>
<td>50%</td>
</tr>
<tr>
<td>Intermediate pressure</td>
<td>5013 kPa</td>
</tr>
<tr>
<td>Discharge temperature in the first stage</td>
<td>56°C</td>
</tr>
<tr>
<td>Superheating in the evaporator</td>
<td>5°C</td>
</tr>
<tr>
<td>Superheating in the suction line</td>
<td>1°C</td>
</tr>
</tbody>
</table>

As demonstrated in the schematic diagram corresponding P-h diagram (Fig.1), the two-stage cycle is characterized by points 1-2-3-4-5-6-7-8-9. In state 1, the suction gas is compressed in the first stage and discharged (state 2). Discharge gas is cooled in an intermediate heat exchanger (state 3) before the cold refrigerant is introduced into the “hot side” of the heat exchanger (the vapor suction side is known as “cold”). Intercooling occurs following the first stage of compression, reducing the discharge temperature in second-stage compression. According to Özgur (2008), this decrease in the second stage allows the refrigerant to reach lower temperatures at the gas cooler outlet, producing greater refrigeration capacity. Table I exhibits
operational data for the system. These were entered into Coolpack simulation software to obtain the corresponding P-h diagram.

Figure 1. P-h Diagram for the two-stage compression transcritical refrigeration cycle.

Actual specific enthalpies for the overheated gas refrigerant at the compressor outlet in the first stage (low stage – LS), and second stage (high stage – HS) are obtained by Eqs. (2) and (3), below.

\[ h_2 = \frac{(h_{2,i} - h_1)}{\eta_{i,LS}} + h_1 \]  
\[ h_4 = \frac{(h_{4,i} - h_3)}{\eta_{i,HS}} + h_3 \]

where \( \eta_i \) is the isentropic efficiency of each compressor and \( h_{n,i} \) is the enthalpy of the overheated refrigerant gas at the compressor outlet for an isentropic compression process (Fatouh and El Kafafy, 2006).

Variations of thermodynamic properties are calculated by applying volume control formulations for each component of the cycle, determining the characteristics of the proposed refrigeration cycle. Using the principle of mass conversion and considering the working liquid flow for each component is permanent, the system and volume control formulation applied to the mass produces the principle of mass conversion. Applying the general control volume formulation for energy with axis operation and frontiers, disregarding kinetic and potential energies, interactions of heat and operational frontier of the system are equal to the sum of enthalpy flow through a frontier, resulting in the equations presented below.

a) Refrigeration effect (evaporator)

\[ q_e = (h_1 - h_f) \]  

b) Heat rejection effect in the gas cooler

\[ q_{gc} = (h_4 - h_3) \]
c) Heat rejection effect in the intermediate internal heat exchanger (intercooler)

\[ q_{\text{int}} = (h_2 - h_3) \]  \hspace{1cm} (5)

d) Total compression

\[ w_{\text{comp}} = \frac{(h_{4,\text{is}} - h_5)}{\eta_{\text{is,HS}}} + \frac{(h_{2,\text{is}} - h_5)}{\eta_{\text{is,LS}}} \]  \hspace{1cm} (6)

e) Coefficient of performance

\[ \text{COP} = \frac{Q_{\text{in}}}{w_{\text{comp}}} = \frac{(h_1 - h_3)}{(h_{4,\text{is}} - h_5)/\eta_{\text{is,HS}} + (h_{2,\text{is}} - h_5)/\eta_{\text{is,LS}}} \]  \hspace{1cm} (7)

3. RESULTS AND DISCUSSIONS

Fig. (2) illustrates the effect of using an internal heat exchanger on performance in single and two-stage compression cycles as pressure in the gas cooler increases. Irrespective of the presence or not of the heat exchanger, performance of the two-stage compression system is shown to be superior to the single stage cycle under the same operating conditions. Differences in performance between the cycles remain almost constant with increased pressure. The same figure also demonstrates that, at a pressure of 95 bar, performance in the single stage cycle is the same with or without the heat exchanger, growing apart as pressure in the gas cooler increases. This type of cycle is less sensitive to heat exchangers with regard to the coefficient of performance. In accordance with Cecchinato et al. (2009), heat exchange between the low pressure gas and the high pressure fluid after expansion benefits heat rejection by the intercooler. Thus, the two-stage compression cycle with an internal heat exchanger displays significantly better performance than those without it.

![Figure 2](image)

**Figure 2.** Effect of an internal heat exchanger (SGHX) on cycle performance.

Fig. (3) demonstrates the influence of gas cooler outlet temperatures on performance in both cycles. It has already been established that lowering the gas cooler outlet temperature of the liquid refrigerant improves system performance. This is primarily due to the increase in refrigeration capacity for the cycle resulting from this parameter since the fluid is subject to a greater enthalpy increase in the evaporator as evaporator inlet temperature decreases. Analysis of figure 6 indicates that the COP for both transcritical cycles grows considerably within a gas cooler temperature outlet variation range of 6°C. Performance of the two-stage cycle is superior to that of the single stage system in 15% of cases.
Fig. (4) exhibits the influence of isentropic compression efficiency cycle performance at gas cooler pressures of 95 to 105 bar. Global isentropic compression efficiency (including electrical losses of the motor) depends on the system compression ratio. As the latter increases, isentropic efficiency falls (Cecchinato et al., 2009). The COP of both cycles improves with the increase in compression efficiency as a result of less energy losses during compression. The performance of both cycles at 95 bar is greater in relation to operation at 105 bar owing to the lower pressure ratio developed in the system.

Fig.(5) below depicts the effect of gas cooler pressure fluctuation on performance in the two transcritical cycles. Once again the two-stage compression cycle shows superior performance. Optimal high pressure is observed in both cycles for corresponding optimum performance, attributed the highly sinuous nature of isotherms above the critical point for R744. Sudden variations in compressor discharge pressure may therefore significantly alter system performance. There are several studies in the literature suggesting methodologies to determine optimal high pressure in transcritical cycles. Optimal discharge pressure for both cycles in predicted operating conditions is approximately 80 bar.
Fig. (6) presents compression work variations of high and low pressure compressors. As intermediate pressure increases, work required by the compressor in the first stage (booster) grows, resulting in the need for less compression work in the second stage. With regard to the cycle under analysis, optimal intermediate pressure is around 48 bar, where the lowest compression required by the system is achieved (total).

Figure 6. Influence of intermediate pressure on compression work during stages.

Figure 7 illustrates energy consumption per day for both transcritical cycles. As expected, the two-stage cycle shows energy consumption 11% lower than the single stage compression cycle. This makes it a consistent alternative in the “perpetual” search for systems with low electrical energy consumption. The two-stage cycle is also more sensitive to gas cooler pressure variations (high pressure) with regard to energy consumption. Similarly, energy consumption at optimal intermediate and gas cooler pressure is lower than other pressure conditions for both cycles.

Figure 5. Influence of gas cooler pressure on cycle performance.
Figure 7. Influence of gas cooler pressure on energy consumption for both cycles.

Figure 8 demonstrates how performance in the two-stage compression cycle grows with an increase in intercooling. Once more, the important effect of gas cooler pressure variation (high pressure) on performance is shown. A critical evaluation of pressures developed in the system is therefore necessary since this operational parameter has the greatest influence on transcritical refrigeration systems, whether operating in single or two-stage compression. Cavallini et al. (2005) proposed an intercooler with copper tube-in-tube heat exchanger with the R744 flowing inside three pipes (ID 6 mm, OD 8 mm) fed in parallel and inserted inside a 20 mm ID (22 mm OD) copper tube. Water flows inside the outer tube in counter-flow to R744. The IC was designed for 1°C temperature approach between the two fluids. Inter-stage heat rejection depends heavily on the presence of the internal heat exchanger and on its effectiveness (Cecchinato et al., 2009). The obvious conclusion is that the internal heat exchange between vapour at low pressure and the high pressure fluid before throttling enhances the benefit of inter-stage heat rejection. The application of gas injection into a two-stage CO2 cycle can reduce compressor power consumption and decrease compressor discharge temperature by providing intercooling effects. The gas injection technique can increase the mass flow rate in the gascooler. To obtain a proper injection mass flow rate, the first- and second-expansion devices must be precisely controlled in a two-stage CO2 cycle. The cooling COP of the two-stage gas injection cycle was maximally enhanced by 16.5% over that of the two-stage non-injection cycle in the experiments (Cho et al., 2009).

Figure 8. Influence of the degree of intercooling on performance in the two-stage cycle.

4. CONCLUSIONS
It is an established fact that theoretical transcritical cycles operating on R744 reduce energy efficiency in comparison with traditional gas compression cycles. However, the favorable characteristics of heat transfer, pressure loss and the compression process of an actual system may partially compensate for the intrinsic thermodynamic deficiency of this fluid.

The analysis developed in the present study demonstrates that staged compression with intercooling through a secondary external liquid is the best alternative for increasing energy efficiency. Since the amount of heat rejected between compression stages depends primarily on the intermediate pressure adopted, an optimal value is needed for this project parameter. Thus, using an internal heat exchanger between the suction line and the line before the expansion device also substantially benefits energy efficiency in the single stage compression cycle.

Isentropic efficiency values for compressors and gas cooler efficiency values are important parameters for achieving lower gas cooler outlet temperatures. Analysis of Fig.(3) shows COP fluctuates with gas cooler discharge temperatures. However, ambient temperature is a limiting factor for obtaining reduced temperatures. Applying intercooling during compression is a more effective mechanism for increasing energy efficiency than other alternatives, such as staged expansion, applied to this system architecture (Cecchinato et al., 2009).

The current reality of global warming demands refrigeration systems using less electrical energy and alternative refrigerants with low GWP. This research shows that lower electrical energy consumption is achieved by using a two-stage transcritical cycle with intercooling. Fig. (8), shows that the amount of R744 intercooling has a significant impact on system performance.

5. REFERENCES


