PERFORMANCE ANALYSIS OF TWO-STAGE TRANSCRITICAL REFRIGERATION CYCLE OPERATING WITH R744

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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.*

Keywords*: Natural refrigerants, Carbon dioxide, Performance, Two-stage compression, Transcritical cycle.*

1. INTRODUCTION

In the first decades of the twentieth century carbon dioxide (CO_2/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 CO_2 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 CO_2 occurred from 1940 to 1955 for refrigerated maritime transport. In Europe, refrigeration machines were limited to using CO_2 due to legal restrictions on toxic or inflammable refrigerants such as ammonia (NH_3) and sulfur dioxide (SO_2). 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 CO_2 could resurge. A bibliographical review indicated that the primary incentive for the reappearance of R744 as a refrigerant was the experimental results of an automotive air-conditioning prototype presented by Lorentzen and Pettersen em 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.

Natural refrigerants are frequently referred to as "alternatives", suggesting they will likely never be considered a primary option. Should this continue to be true, it is due to several factors and circumstances. However, regardless of natural refrigerant results, particularly R744, substantial impact has already been observed on the development of refrigeration systems in the area. Each new application of carbon dioxide introduces another model design to increase efficiency, compactness or esthetics. These improvements have often been accepted and applied to other refrigerants and systems. An excellent example is the microchannel heat exchanger. It is one of the reasons why R744 systems and their components are successfully challenging the cost base. Some R744 refrigeration models are several times smaller, with the same capacity and greater efficiency than conventional systems. This is because they present significant technological advances (Hrnjak, 2007).

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 -30ºC with condensation between -30 and -10ºC. 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).

Cecchinato et al. (2009) developed an thermodynamic evaluation and optimization of different two-stage transcritical carbon dioxide cycles. Five different cycles are studied: basic single stage, single-throttling with two-stage compression cycle, split cycle, phase separation cycle and single-stage cycle coupled with a gas cooling circuit. Each basic cycle is analysed for the effect of internal heat transfer between different streams of refrigerants. In the case of two-stage compression, intermediate cooling between the compressor stages is present. The autors investigated the influence of such parameters on performance of diverses two-stage compression cycles: presence of internal heat exchanger, compressors isoentropic efficiency and external cooling agent temperature.

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.

1.1 Fundamentals of the Transcritical Refrigeration Cycle with R744

Although R744 has poor thermodynamic properties with regard to energy efficiency in a traditional reverse refrigeration cycle, it is considered an efficient solution to anthropic global warming. This is because it contains characteristics beneficial to the environment and can be obtained from industrial process waste based on a vision of sustainable production (Cecchinato et al., 2009).

When compared with conventional refrigerant fluids, the most notable property of R744 is its low critical temperature of 30.98°C. Refrigerants with lower critical temperatures than the ambient temperature – tropical regions – cannot use the condensation process for heat rejection. Instead, the refrigeration cycle becomes transcritical, located between subcritical and supercritical temperatures (Danfoss, 2008). As per Fig. (1), the greatest difference between cycles using 744 and R134a for example, is in the line following compression. In the conventional system, the phase change (vapor – liquid) known as liquid condensation, occurs after compression. However, in the transcritical cycle gas is highly overheated and cooled, without phase change, in a *gas cooler.*

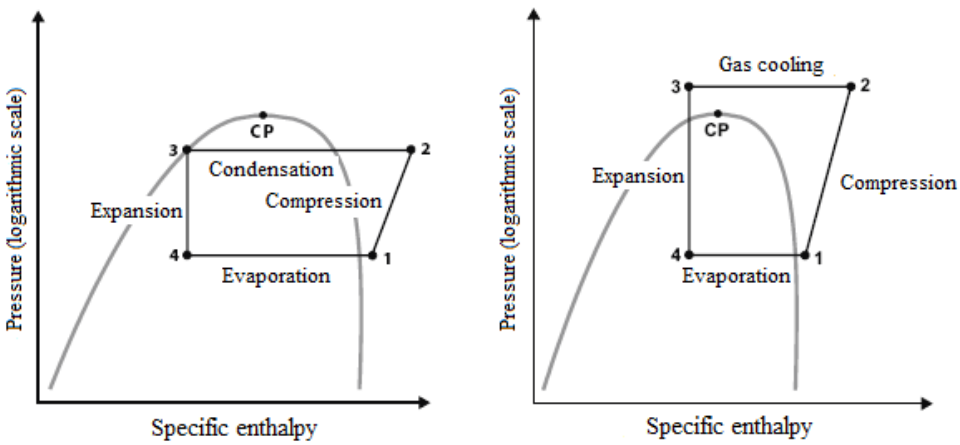

Figure 1. Pressure Diagram-Enthalpy for the subcritical cycle with R134a and transcritical cycle with R744.
Source: Rohrer (2006).

This is a unique characteristic of the R744 cycle, where gas at supercritical temperatures is cooled in a *gas cooler* rather than a condenser. It is important to note that supercritical R744 properties vary rapidly during the gas cooling process, making the *gas cooler* project important and challenging. The heat discharge process (gas cooling) occurs at constant pressure above the critical point. Temperature during this process fluctuates continuously between the inlet and outlet of the *gas cooler*. Fig.(2) illustrates the configuration of the transcritical cycle in relation to simple stage compression. Fig. (3) demonstrates configuration in the two-stage transcritical compression cycle, the object of analysis for this study. The compressor, expansion device and evaporator perform the same functions in conventional subcritical systems. The role of the *gas cooler* was discussed earlier. In accordance with Kim et al. (2004), using an internal heat exchanger reduces irreversibility during refrigerant expansion. It is also applied in order to lower discharge temperature of the liquid in the *gas cooler*, increasing COP. Internal heat exchangers are effective for: performance improvement, undercooling the refrigerant to avoid bubble formation at the expansion device inlet, completely evaporating the fluid in the suction line to prevent mechanical damage to the compressor (Klein et al., 2000). Chen and Gu (2005) confirm that

internal heat exchangers are necessary in R744 systems. Bibliographic research determined that the type of expansion device (capillary tube or expansion valve) used differs among authors. According to Madsen et al. (2005), capillary tubes are recommended in applications where evaporation pressure is constant and discharge temperature from the *gas cooler* varies no more than $\pm 10\,\mathrm{K}$ from the project condition. Capillary tube is more flexible to sudden ambient temperature changes, it can provide good control of optimal high pressure, offering maximum COP and greater energy efficiency (Agrawal and Bhattacharyya, 2009).

Diagrams of transcritical simple stage and two-stage compression cycles are shown in Figs. 2 and 3, respectively.

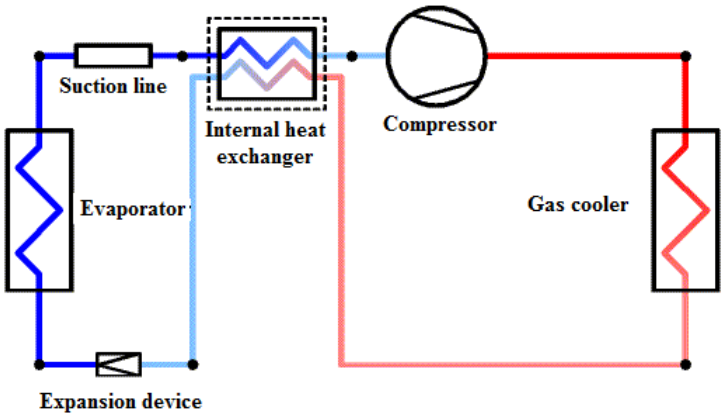

Figure 2. Diagram of a transcritical single stage compression refrigeration cycle**.**
Source: Adapted from CoolPack software.

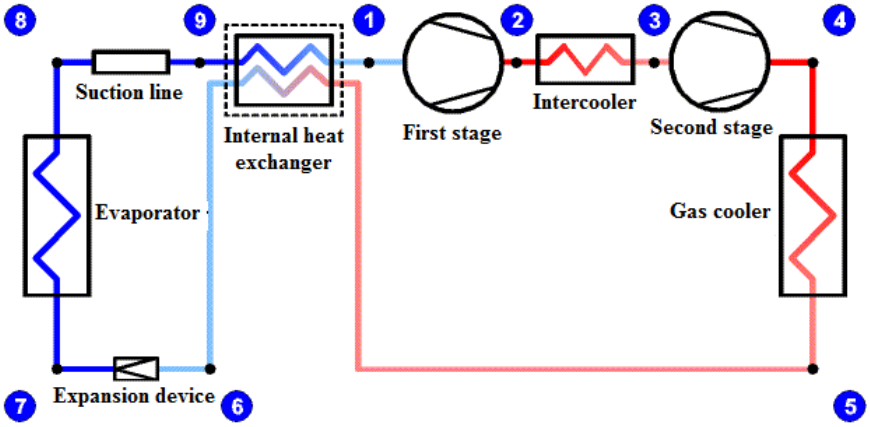

Figure 3. Diagram of a two-stage transcritical compression refrigeration cycle.
Source: Adapted from CoolPack software.

1.2 Multistage Pressure Transcritical Cycle

A significant portion of refrigeration installations operate between evaporation and condensation temperatures whose difference varies from 50ºC to 80ºC. Although such a marked temperature difference presents a series of operational problems, it also imposes a search for non-trivial solutions. One of these is multistage compression, which implies an increase in initial installation cost in relation to simple stage compression. In contrast, the use of multiple stages reduces some of the problems resulting from high temperature differences, as well as lowering compression power (Stoecker and Jabardo, 2002). Common transcritical cycles have elevated compressor discharge temperatures in the range of 100ºC, primarily owing to the need to achieve high pressures in the gas cooler (approximately 95 bar). A solution adopted to improve performance in transcritical cycles with R744 is the use of two-stage compression with intercooling. This reduces compressor discharge temperatures in the second stage and produces lower gas cooler outlet temperatures, thereby providing greater refrigeration capacity. Energy efficiency is an essential requirement for any substitute of traditional refrigerant liquids. Thus, R744 can only be successful in operating conditions of more sophisticated cycles and/or more effective components in order to compensate for the lower energy efficiency in the transcritical cycle of reference (Cecchinato et al., 2009).

Intercooling aims to decrease overheating of the refrigerant as it leaves the low pressure stage. Initially, it may seem that this cooling would result in reduced total compression. In compressed air, intercooling is carried out at a

relatively high temperature to facilitate cooling by the ambient air. However, in refrigeration systems, intercooling occurs at somewhat low refrigerant temperatures, which requires an additional cost. Intercooling does not imply a significant reduction in compression power, and can even increase it. The main reason for applying this process is to limit refrigerant temperature at compressor discharge. In installations containing reciprocating compressors, high discharge temperatures may compromise compressor lubrication and reduce the useful life of discharge valves (Stoecker and Jabardo, 2002).

In two-stage air compression, optimal intermediate pressure corresponds to the geometric mean between aspiration pressure (suction) and discharge (gas cooler), as shown in Eq. (1) (Stoecker and Jabardo, 2002). According to Yang et al. (2007), optimal intermediate pressure for the two-stage transcritical cycle can be estimated through the classic two-stage air compression formula. An iterative method must be used in order to determine actual optimal intermediate pressure (Agrawal et al., 2007), which is a crucial choice parameter (Cecchinato et al., 2009).

$$P_{INT} = \sqrt{P_{SUCTION} \cdot P_{DISCHARGE}} \tag{1}$$

2. THEORETICAL MODELING AND SIMULATION

2.1. Simulation Code and Validation

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).

2.2. Operation and Performance Parameters

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). Optimal intermediate pressure was estimated by Eq. (1).

Table 1. Operational parameters for the refrigeration system.

Parameter	***Value***
Evaporation temperature	-10ºC
Refrigeration capacity	5 kW
Isentropic efficiency of compression	60%
Pressure in the *gas cooler*	95 bar
Discharge temperature of the *gas cooler*	32ºC
Degree of Intercooling	25ºC
Thermal efficiency of the internal heat exchanger	50%
Intermediate pressure	5013 kPa
Discharge temperature in the first stage	56ºC
Overheating in the evaporator	5ºC
Overheating in the suction line	1ºC

The thermal efficiency of the internal heat exchanger expresses how efficient the heat exchange utilises the temperature difference between the liquid and suction line. It is difficult to give a general guideline for the value of the thermal efficiency. For an internal heat exchanger (suction gas heat exchanger) a value between 0.2 and 0.5 should be a reasonable value (DTU, 2001).

2.3. Mathematical Model

As demonstrated in the schematic diagram (Fig.3) and corresponding P-h diagram (Fig.4), the two-stage cycle is characterized by points 1-2-3-4-5-6-7-8-9 in Fig. (3). 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 1 exhibits operational data for the system. These were entered into Coolpack simulation software to obtain the corresponding P-h diagram.

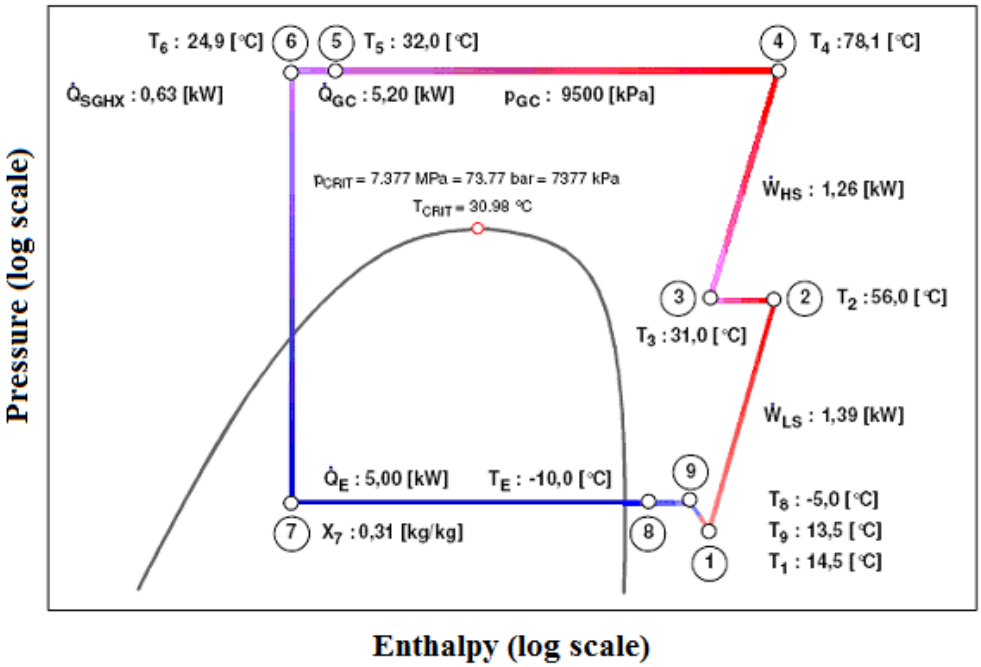

Figure 4. P-h Diagram for the two-stage compression transcritical refrigeration cycle.
Source: Adapted from CoolPack software.

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.

$$h_2 = \frac{(h_{2,is} - h_1)}{\eta_{is,LS}} + h_1 \tag{2}$$

$$h_4 = \frac{(h_{4,is} - h_3)}{\eta_{is,HS}} + h_3 \tag{3}$$

where η_{is} is the isentropic efficiency of each compressor and $h_{n,is}$ 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 conservation 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 quantification, disregarding kinetic and potential energies, interactions of heat in 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)

$$q_e = (h_1 - h_7) \tag{4}$$

b) Heat rejection effect in the gas cooler

$$q_{gc} = (h_4 - h_5) \tag{5}$$

c) Heat rejection effect in the intermediate internal heat exchanger (intercooler)

$$q_{\text{int}} = (h_2 - h_3) \tag{6}$$

d) Total compression

$$w_{comp} = \frac{(h_{4,is} - h_3)}{\eta_{is,HS}} + \frac{(h_{2,is} - h_1)}{\eta_{is,LS}} \tag{7}$$

e) Coefficient of performance

$$COP = \frac{q_e}{w_{comp}} = \frac{(h_1 - h_7)}{\left[(h_{4,is} - h_3)/\eta_{is,HS} + (h_{2,is} - h_1)/\eta_{is,LS}\right]} \tag{8}$$

3. RESULTS AND DISCUSSION

Fig. (5) 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.

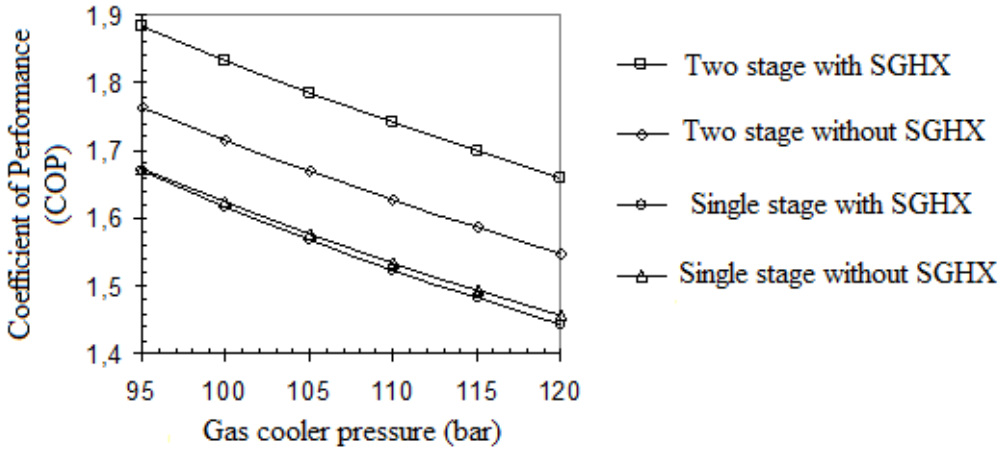

Figure 5. Effect of an internal heat exchanger (SGHX) on cycle performance.

Fig. (6) 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 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 6ºC. Performance of the two-stage cycle is superior to that of the single stage system in 15%.

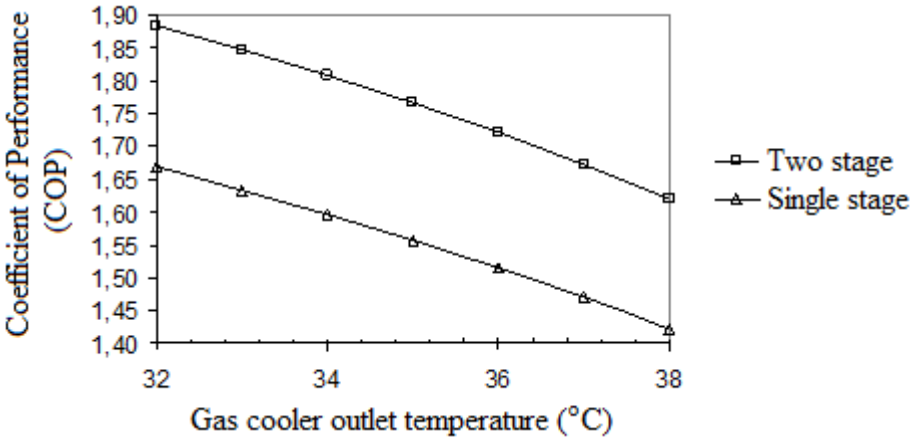

Figure 6. Influence of gas cooler outlet temperature on cycle performance.

Fig. (7) 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.

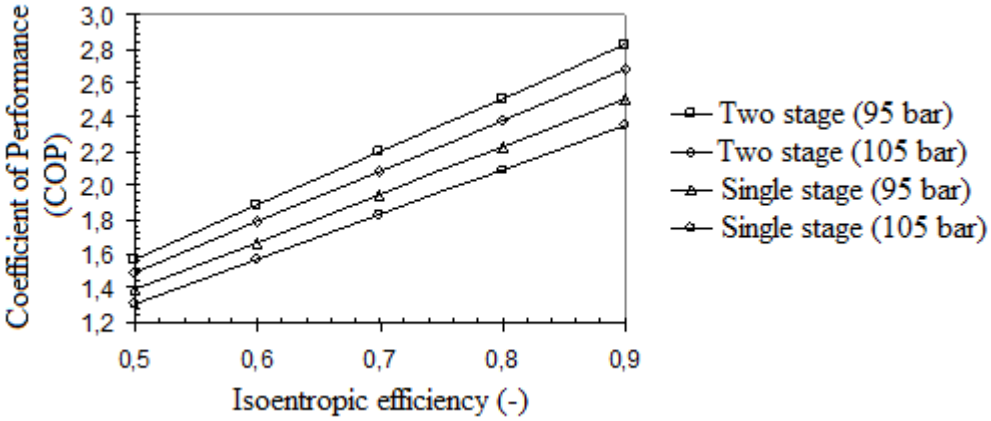

Figure 7. Influence of Isentropic compression efficiency on cycle performance.

Fig.(8) 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 to the highly sinuous nature of isotherms above the critical point for R744. Sudden variations in compressor discharge pressure may therefore significantly alter the 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.

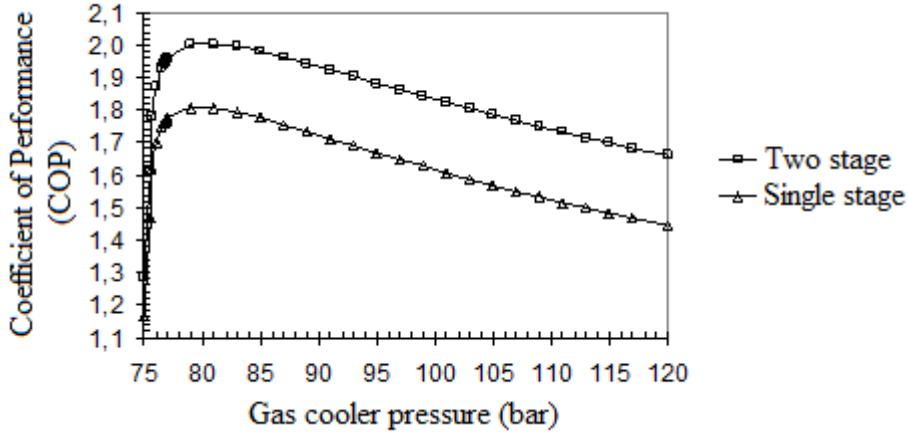

Figure 8. Influence of gas cooler pressure on cycle performance.

Fig.(9) 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 55 bar, where the lowest compression required by the system is achieved (total).

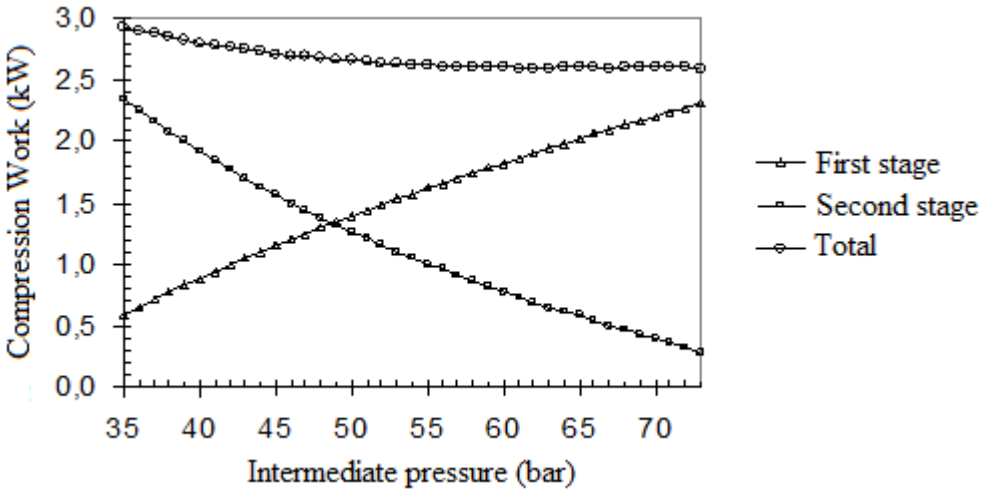

Figure 9. Influence of intermediate pressure on compression work during stages.

Figure 10 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.

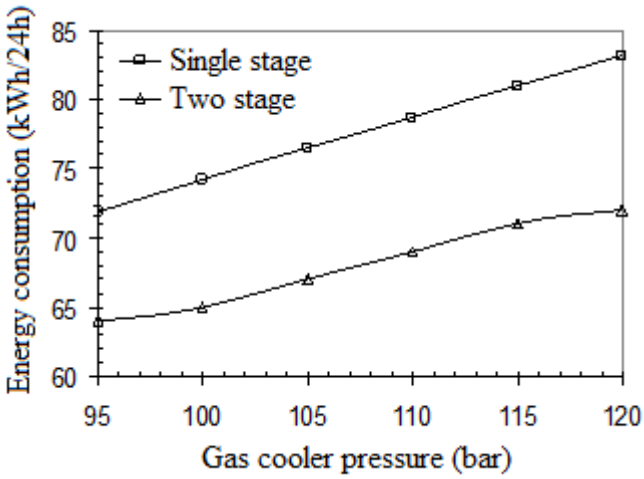

Figure 10. Influence of gas cooler pressure on energy consumption for both cycles.

Figure 11 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.

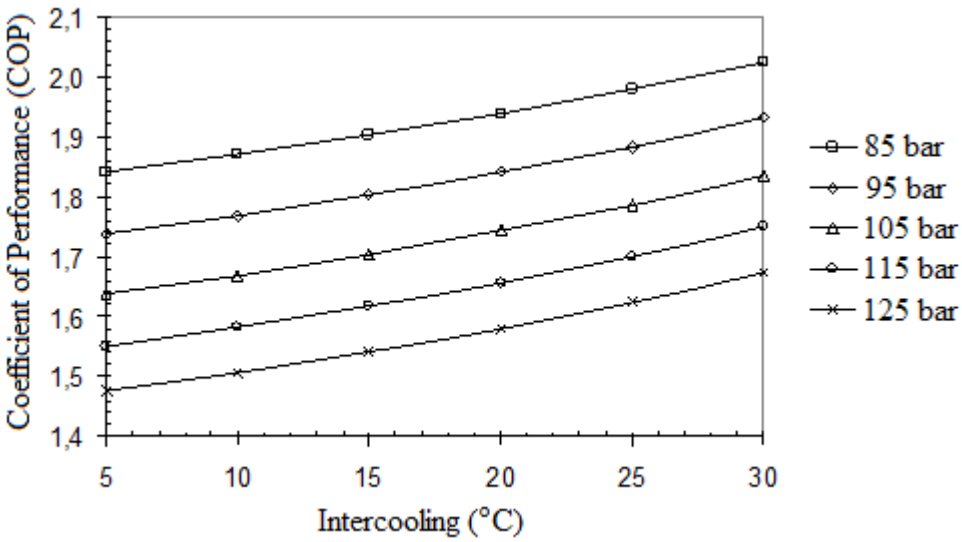

Figure 11. Influence of the degree of intercooling on performance in the two-stage cycle.

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.(6) 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. (11), shows that the amount of R744 intercooling has a significant impact on system performance.

5. ACKNOWLEDGEMENTS

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