CYCLE ANALYSIS OF A SINGLE-STAGE TRANSCRITICAL R744 SYSTEM IN SUPERMARKETS: THE IMPACT OF ISENTROPIC EFFICIENCY AND OIL

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Carbon dioxide is increasingly present in modern refrigeration systems, finding significant application in commercial cooling, especially in supermarkets. However, its application encounters many challenges due to its unfavorable thermodynamic properties. This serves as a starting point for numerous engineering solutions, as well as for many theoretical and experimental analyses of various systems utilizing R744. Some analyses often overlook the fact that, in addition to the refrigerant, oil also passes through all the system components in a certain quantity, which subsequently impacts heat transfer as well as the cycle itself. This paper presents a cycle analysis of a typical single-stage transcritical CO_2 system used for medium-temperature (MT) cooling applications in a supermarket. The analysis included the impact of oil, considering several realistic steady-state scenarios with predicted OCR (Oil Circulation Ratio) values. In addition to the effect of oil, the study was expanded by examining available equations for the isentropic efficiency of the compressor. The results showed that considering a realistic compression process leads to an efficiency reduction of 15.4% to 39.9%, while the presence of oil decreases it by 5.7% to 27.8% under varying conditions.

Key words*: R744; transcritical system; oil effect; OCR; isentropic efficiency*

1. Introduction

The global demand for environmentally sustainable refrigeration technologies and alternatives to synthetic refrigerants has grown significantly over the years. Initially, efforts were made to replace old synthetic refrigerants with new synthetic fluids with lower Global Warming Potential (GWP) factors, but it was ultimately concluded that natural refrigerants represent the true alternative regarding environmental criteria [1]. Carbon dioxide (R744, CO_2) is recognized as a sustainable option for a wide range of applications, being the only non-flammable and non-toxic natural refrigerant capable of operating at lower temperatures within vapor compression systems [2, 3].

However, engineers face numerous challenges that need to be addressed to make implementing technical carbon dioxide solutions acceptable. Some of these challenges include high operating pressures, lower efficiency at higher temperatures, and poor thermodynamic properties near the critical point (slopes of saturation curves, low latent heat, high irreversibility during throttling, etc.). Additionally, transitioning from existing, simple refrigeration systems to new, more complex ones poses a challenge for users and the service sector. The growing popularity of R744 refrigeration systems is also reflected in the increasing number of review papers in the available literature, which explore various system configurations, components, and heat and mass transfer phenomena [3-9].

Supermarkets account for 3–4% of the total electricity consumption in developing countries [10]. Considering that 30–50% of this energy is used for refrigeration [11], approximately 540 kWh/m^2 annually [12], the choice of the refrigeration system to be implemented is very important.

In supermarkets, R744 refrigeration systems are typically encountered in three main configurations: indirect, cascade, and transcritical [4]. In indirect refrigeration systems, R744 is used as a secondary refrigerant with phase change, preferred over single-phase refrigerants due to lower power requirements for pump operation, smaller pipe diameters, and excellent heat transfer characteristics [13]. Cascade refrigeration systems use R744 in the lower cascade, while the upper cascade typically employs refrigerants such as R134a (or its HFO alternatives) and hydrocarbons like propane (R290) [14–15]. The third configuration, transcritical CO_2 systems, is the most commonly used in practice due to its cost-effectiveness, and the advantage of operating with a single refrigerant. Unfortunately, it has been shown that basic configurations of these systems (booster systems) consume more energy than conventional freon-based systems in warm climates [16–18].

In standard supermarkets, the refrigeration systems typically meet cooling demands at two temperature levels. The temperature of frozen products in freezers is maintained at the low-temperature level (LT), while the medium-temperature level (MT) is used, for example, in cold rooms and refrigerated display cabinets. For this reason, some refrigeration systems are designed as two-stage systems. However, it is not uncommon for cooling to be required at only one of the two temperature levels. In such cases, a single-stage system is used.

Cycle analyses of R744 refrigeration systems typically consider only the pure refrigerant, disregarding the fact that oil also circulates throughout the system. This deliberately overlooks its direct impact on the thermodynamic properties of the working fluid [19]. Oil is used for lubricating and sealing the compressor, and synthetic oils are employed in R744 systems. While PAO oils can be used in subcritical systems, PAG and POE oils are more widely applied today due to their superior properties in the transcritical region. The main difference lies in their miscibility with CO_2: POE oils are fully miscible, whereas PAG oils are partially miscible.

The available literature presents the impact of oil in R744 refrigeration systems in various ways. To consider this impact, it is first necessary to determine the amount of oil circulating through the system, which is typically represented by the Oil Circulation Ratio (*OCR*) value. In the paper by Zhang et al. [20], the effect of oil (with a constant *OCR* = 2%) was included in calculating states and energy balances for experimental data reduction in a reversible automobile R744 air-conditioning system. However, the direct effect of oil on system performance was not considered. The paper by Zhu et al. [21] focused on experimentally determining the *OCR* at two locations in a single-stage R744 ejector system, analyzing the influence of various operating parameters on *OCR*. Additionally, a theoretical analysis was conducted for other types of ejector cooling systems. The impact of *OCR* on the heating performance of a heat pump in electric vehicles was experimentally analyzed and presented in the paper by Li et al. [22]. The refrigerant R134a and PAG46 oil were used, and it was found that proper oil charge is of great importance. Oil charge was specifically examined in the paper by Tang et al. [23] for the CO_2 automotive air conditioning system under different oil quantities and operating conditions. The study determined a critical *OCR* of 9.2%, which leads to zero superheat at the evaporator outlet. Regarding appropriate thermodynamic models and interpretations of oil-refrigerant mixtures, Thome [19] was among the first to present a thermodynamic model for calculating different oil-refrigerant mixtures. Despite focusing on outdated synthetic fluids, this paper represents a solid starting point. More recent thermodynamic models are primarily based on modified cubic equations of state and the thermodynamic properties of mixtures. The thermodynamic properties of the CO_2-PAG46 mixture, determined experimentally, were presented in the paper by Feja and Hanzelmann [24], while the thermodynamic properties and a model in the form of the Redlich-Kwong-Soave cubic equation for the CO_2-PAG68 mixture were presented in the papers by Domin et al. [25] and Feja et al. [26], respectively. Despite numerous studies on the effect of oil on heat transfer (mainly evaporation phenomena), this field remains insufficiently explored in terms of oil's effect on overall system performance, as well as the appropriate thermodynamic models and properties for all oil-carbon dioxide combinations.

In this paper, an extended cycle analysis is presented, which includes the effects of oil and isentropic efficiency (η_{is}) by varying different *OCR* values and using various equations for η_{is}. The subject of the analysis is a typical single-stage transcritical R744 refrigeration system for MT cooling in supermarkets. The cycle analysis was conducted for different steady-state scenarios corresponding to the real operation of such a system. A typical system schematic and the cycle are shown in Fig. 1 and Fig. 2, respectively.

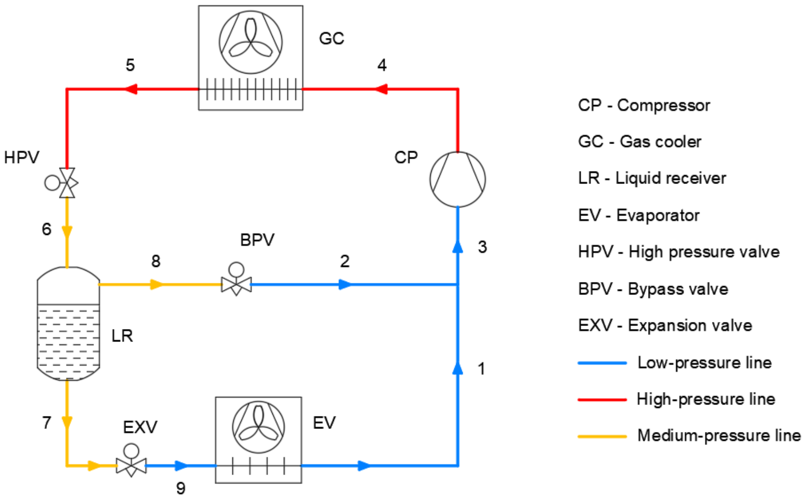

Figure 1 - System schematic of the single-stage R744 refrigeration system (MT)

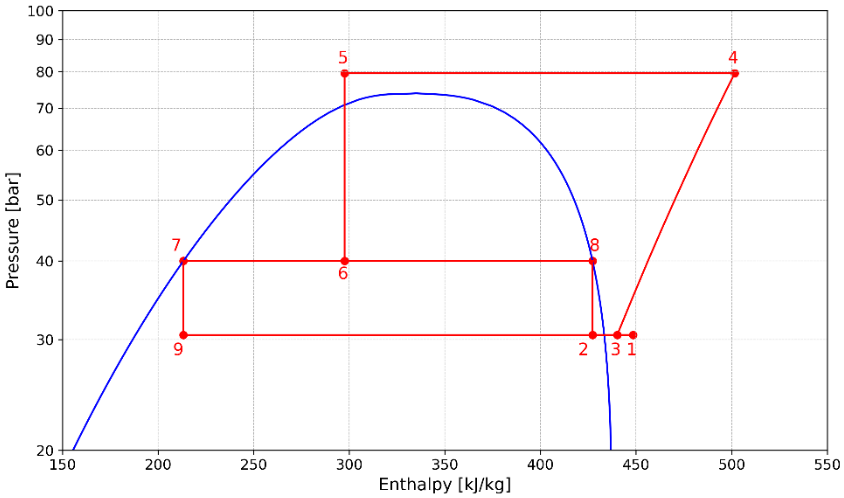

Figure 2 -Single-stage transcritical cycle in log *p-h* diagram

In practice, the majority of these systems are designed as condensing units, where most components are aggregated, while the evaporator and expansion valve are placed in the cold room. The condensing unit, in addition to the basic components, contains all the necessary fittings and automation for controlled operation in both subcritical and transcritical regions. However, such units are often designed without oil separators due to space constraints. Hermetic and semi-hermetic compressors of various types are typically used to compress the vapor and ensure refrigerant flow. These systems, which operate in the MT mode, usually provide air cooling from −5 °C to 5 °C.

2. Improvements in cycle analysis

The operation of a vapor compression refrigeration system is theoretically represented by a thermodynamic reverse Rankine cycle for defined operating conditions. To simplify and accelerate the cycle solution, the following assumptions are most commonly made in practice:

- the compression process is considered an isentropic process (s = const),
- the expansion process is considered an isenthalpic process (h = const),
- pressure drops in all heat exchangers and pipelines are neglected,
- heat gains and losses from/to the environment are negligible,
- the cycles operate in steady-state conditions,
- the cycles are performed by the pure refrigerant only (OCR = 0), and
- changes in kinetic and potential energy are neglected.

This paper presents two ways to improve the interpretation of the cycle. One of them is the isentropic efficiency of the compressor as a potentially known variable, which would result in a more realistic state at the compressor discharge. The second improvement to the analysis includes oil as a non-negligible component, leading to a more accurate determination of the cycle state.

2.1. Isentropic efficiency of the compressor

The isentropic efficiency of the compressor is a parameter that represents the deviation of the real compression process from the ideal (isentropic) one. It can thus be expressed as the ratio of specific compression works using the following equation [27]:

$$\eta_{is} = \frac{h_{4,is} - h_3}{h_4 - h_3} \tag{1}$$

where states 3 and 4 represent the suction and discharge (realistic) states of the compressor (Fig. 2), respectively, while state 4is represents the discharge state in isentropic compression.

This parameter is unique to the observed type (series) of the compressor and the defined operating conditions. In the available literature, the isentropic efficiency of the compressor is most commonly presented as a linear function on the compression ratio in the following form [28–34]:

$$\eta_{is} = A - B \cdot \left(\frac{p_{gc}}{p_e} \right) \tag{2}$$

where A and B are correlation coefficients dependent on the compressor type. It can also be represented in polynomial form, as demonstrated in the literature [35], where the correlation was experimentally derived. In [36], where a cubic equation of isentropic efficiency was used, it was shown that the maximum efficiency of CO_2 compressors is achieved for compression ratios of 2.5 to 3.5, and within this range, the maximum COP can be expected. On the other hand, some compressor manufacturers do not explicitly provide information on isentropic efficiency in their technical specifications, even though it is a significant parameter for evaluating the performance of the entire refrigeration system. However, these values can be indirectly determined.

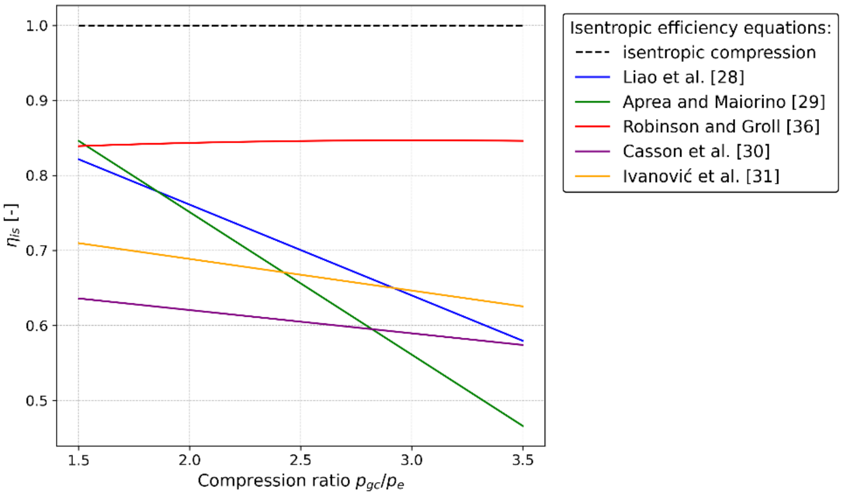

Figure 3 - Isentropic efficiency as a function of the compression ratio according to several authors

Figure 3 illustrates several equations of isentropic efficiency available in the open literature. The equation from the work of Liao *et al.* [28] was derived based on an experimental dataset provided by Danfoss for CO_2 compressors; however, the specific types of compressors involved were not defined. The study by Aprea and Maiorino [29] presented a similar equation (with a somewhat steeper decline) developed for a reciprocating semi-hermetic compressor. The cubic equation from Robinson and Groll [36] yields significantly higher and nearly constant values of isentropic efficiency, though the type of analyzed compressor remains unknown. The last two equations were developed for reciprocating semi-hermetic CO_2 compressors from different manufacturers. The equation from Casson *et al.* [30] was formulated for a Dorin compressor, while the equation from Ivanović *et al.* [31] was based on data from the selection software of Emerson. Notably, these two equations exhibit a similar trend.

It is also worth noting that the available literature provides limited information regarding the isentropic efficiency of other types of compressors, such as scroll, screw, and rotary. However, in the absence of such data, even approximate values for this parameter can contribute to a better representation of the system.

2.2. Effect of oil

As previously mentioned, oil is essential for the proper and safe operation of a refrigeration system. Yet, it also acts as a "contaminant" that (mostly negatively) impacts heat transfer between the refrigerant and the environment. It is well-known that oil is not confined solely to the compressor but circulates throughout all system components, with some components even retaining it. For this reason, neglecting oil in cycle analysis prevents the operation of the refrigeration system from being fully represented and adequately described.

The amount of oil circulating with the refrigerant varies depending on operating conditions and the complexity of the system itself. Installing an oil separator directly reduces this amount, but some oil always continues to flow with the refrigerant. The determined quantity of oil in the system is expressed as the mass fraction of oil in the overall mixture of refrigerant and oil, using the *OCR*:

$$OCR = \frac{m_o}{m_r + m_o} = \frac{m_o}{m_{tot}} \tag{3}$$

In a typical single-stage R744 system, the presence of a receiver (Fig. 1, LR) will affect the *OCR*, which will be discussed in the following section. Additionally, the vapor quality is an important parameter in defining the refrigerant's two-phase flow. It is evident that including the oil quantity in the analysis will also influence this parameter due to the total amount of the mixture, which can be expressed using the following equation:

$$x = \frac{m_{r,v}}{m_{r,v} + m_{r,l} + m_o} = \frac{m_{r,v}}{m_{tot}} \tag{4}$$

To incorporate oil into the determination of the cycle's state, it is necessary to define its enthalpy. In the paper [19], an equation is provided for calculating the specific heat capacity of oil as a function of temperature t and specific gravity g. By extending this equation, a straightforward formulation for determining the oil's enthalpy at a given temperature t can be derived:

$$h_o(t) = h_{o,ref} + 4.186 \cdot \frac{0.388 \cdot \left(t - t_{o,ref}\right) + 0.00045 \cdot \left(1.8 \cdot \left(t^2 - t_{o,ref}^2\right)/2 + 32 \cdot \left(t - t_{o,ref}\right)\right)}{\sqrt{g}} \tag{5}$$

where $h_{o,ref}$ represents the reference enthalpy of the oil at the reference temperature $t_{o,ref}$. The specific type of oil in the system influences the enthalpy through the specific gravity.

After determining the enthalpy of the oil, the state of the refrigerant-oil mixture can be established using the mixing rule as follows:

$$h_{mix} = \left(1 - OCR\right) \cdot h_r + OCR \cdot h_o \tag{6}$$

The general form of Eq. (6) can be used to determine nearly any state within the cycle. However, accounting for vapor quality in states where a two-phase refrigerant mixture occurs is essential. Oil does not undergo phase changes throughout the cycle, and its enthalpy changes more slowly than the refrigerant's. Consequently, the state of the mixture can generally be expected to have a lower enthalpy than that of the pure refrigerant.

3. Cycle analysis

The presented calculation method for the single-stage transcritical R744 cycle incorporates isentropic efficiency and oil. The analysis is performed under steady-state operating conditions and in transcritical mode, demonstrating how the Coefficient of Performance (*COP*) varies as a function of different parameters. The aim of the analysis is to determine the deviation compared to the basic cycle with isentropic compression and pure refrigerant.

The analysis implemented isentropic efficiency η_{is} as a variable using the five different equations. It was conducted at a constant ambient air temperature t_{amb} = 29 °C and an evaporation temperature t_e = −5 °C, while the gas cooler pressure was varied within the range p_{gc} = 75-120 bar. Additionally, the case of isentropic compression was included to determine the corresponding deviation.

Subsequently, oil was included in the cycle calculation. Since different isentropic efficiency equations were examined in the first part, only one equation (Ivanović *et al.* [31]) was used for further calculations, expressed in the following form:

$$\eta_{is} = 0.77294 - 0.04216 \cdot \left(\frac{p_{gc}}{p_e} \right) \tag{7}$$

For this analysis, PAG 100 oil was selected (density ρ_{PAG100} = 996 kg/m^3 at a temperature of t = 15 °C), as it is used in some modern CO_2 condensing units. Given that this oil is partially miscible with CO_2, the mixture's state is approximated using Eq. (6). The amount of oil, expressed as the OCR value, was varied from 0% to 10%, based on available literature data for CO_2 systems.

The analysis assumes that pure refrigerant vapor exits the receiver (separator) without any dissolved oil. This assumption enables the calculation of a new *OCR* value through the low-pressure components and is determined as follows:

$$OCR_e = \frac{m_o}{m_{r,l} + m_o} = \frac{OCR}{(1-x)} \tag{8}$$

The newly calculated OCR_e value is subsequently used to determine all states from the receiver outlet to the evaporator outlet by adapting Eq. (6).

The effect of oil is examined under two different conditions: in one part, the ambient air temperature is varied within the range t_{amb} = 29-36 °C, while in the other, the evaporation temperature is varied within the range t_e = −10-2 °C. However, in both cases, the optimal pressure for achieving the maximum *COP* was first determined for all varied conditions.

In the analysis, the receiver pressure was assumed to be p_m = 40 bar, the temperature difference between the air and CO_2 at the gas cooler outlet $\Delta t_{gc,out}$ = 3 °C, and the total superheat before the mixing state *SH* = 10 °C.

4. Results and Discussion

The results of the analysis are presented from an efficiency perspective through a diagram illustrating the *COP* as a function of the varied parameters.

Figure 4 illustrates how the *COP* varies with high pressure (gas cooler pressure) for different equations of η_{is}.

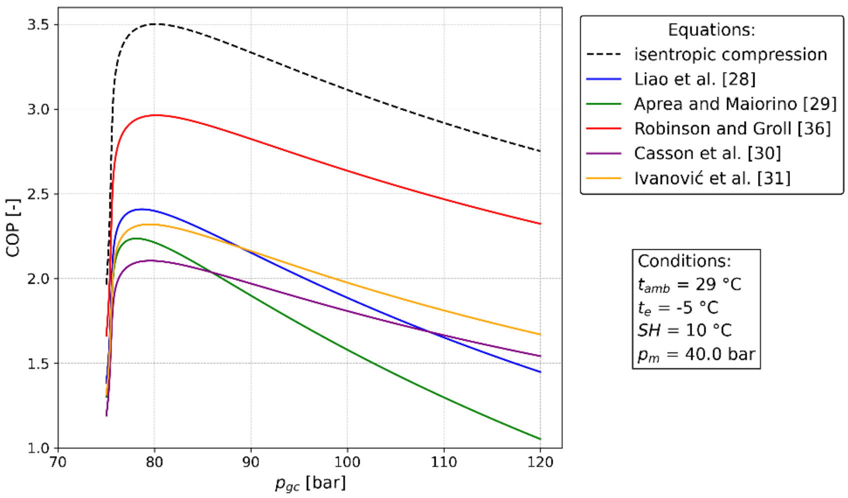

Figure 4 - *COP*-p_{gc} for different equations for η_{is}

Different equations affect the shift in optimal pressure, as observed in the displacement of the maxima of the depicted curves. Consequently, as anticipated, it can be concluded that the isentropic efficiency influences the optimal pressure (with a maximum difference of up to 2.1 bar). However, this influence also encompasses the impact of evaporation temperature through the compression ratio [31] and is not as significant as the ambient temperature.

If the maximum *COP* values (at optimal pressure) are compared, the following conclusions can be drawn. It is noticeable that the cubic equation by Robinson and Groll [36] shows the smallest deviation from the isentropic process (δ = 15.4%) due to unusually high and constant values of η_{is}. The equations by Liao *et al.* [28] and Aprea and Maiorino [29] result in the fastest decline in the *COP* with increasing high pressure, in accordance with the slopes of the η_{is} curves (Fig. 3), with their deviations under optimal conditions being δ = 31.2% and δ = 36.1%, respectively. On the other hand, when using the equations by Casson *et al.* [30] and Ivanović *et al.* [31], the decrease in the *COP* with increasing high pressure is more gradual. Both curves follow a similar trend, with deviations of δ = 39.9% and δ = 33.8%, respectively.

If the deviation among the equations that yield similar results is considered, taking the equation by Casson *et al.* [30] as the reference, the following values are obtained. The deviations in the *COP* for optimal conditions for the equations by Liao *et al.* [28], Aprea and Maiorino [29], and Ivanović *et al.* [31] are δ = 14.4%, δ = 6.2%, and δ = 10.2%, respectively.

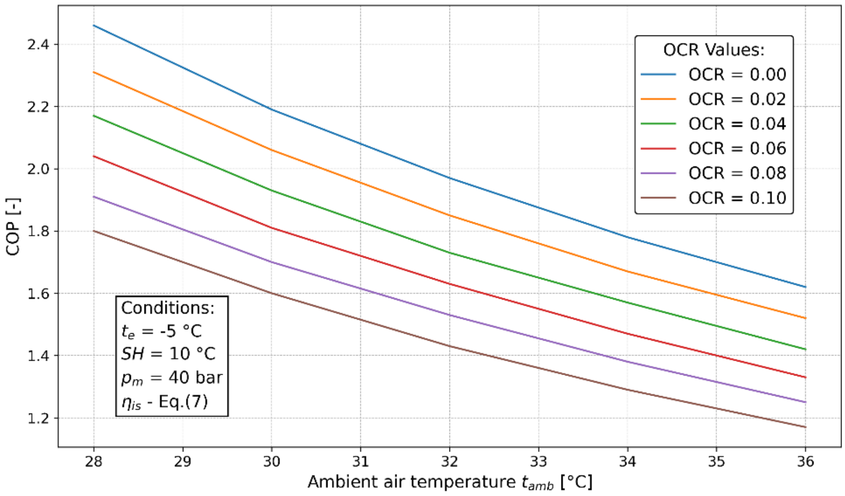

Figure 5 - *COP*-t_{amb} for different *OCR* values

Figure 5 illustrates how the *COP* varies with ambient air temperature for different OCR values.

As expected, by introducing oil into the cycle calculation, lower values of the *COP* are obtained. This can be explained by the fact that the lower enthalpy values of the oil directly affect the reduction in the enthalpy of the mixture, which results in an indirect decrease in the *COP*. Due to the significantly slower change in the enthalpy of the oil compared to the enthalpy of the refrigerant, this difference is particularly pronounced in states where the refrigerant is in a vapor state with a higher oil content. Naturally, with an increase in the amount of circulating oil (increased *OCR*), the *COP* further decreases. It should also be noted that for the assumed *OCR* values, higher OCR_e values are obtained, so in states dependent on it, the effect of oil is more significant.

With an increase in air temperature, the *COP* inevitably decreases, and with an additional increase in *OCR*, the reduction ranges from a minimum of 5.9% (*OCR* = 2%, t_{amb} = 30 °C) to a maximum of 27.8% (*OCR* = 10%, t_{amb} = 36 °C).

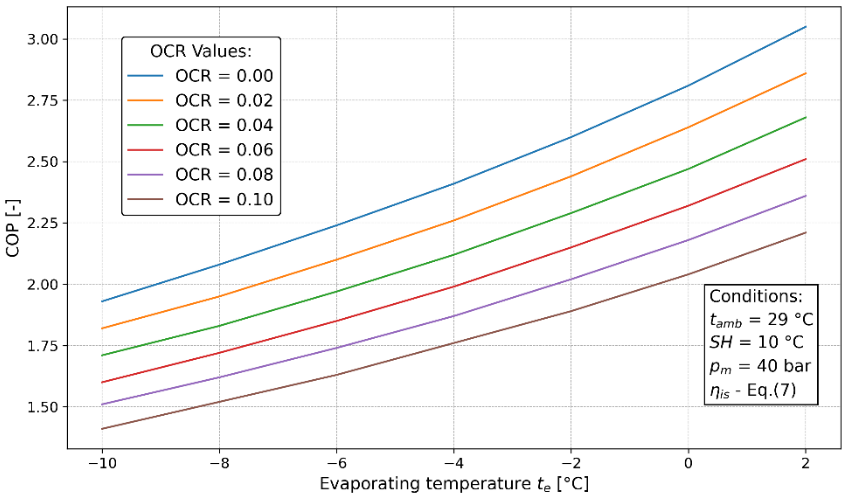

Figure 6 - *COP*-t_e for different *OCR* values

Figure 6 illustrates how the *COP* varies with evaporation temperature for different *OCR* values.

As expected, an increase in the evaporation temperature increases the *COP*, while an increase in the amount of oil decreases it. The analysis has shown that this decrease due to the addition of oil ranges from a minimum of 5.7% (*OCR* = 2%, t_e = −10 °C) to a maximum of 27.5% (*OCR* = 10%, t_e = 2 °C).

Finally, the analysis of the oil's effect revealed that changes in the optimal pressure are influenced by both the evaporation temperature (change in the range p_{gc} = 78.8-80.1 bar with a decrease in t_e) and the ambient air temperature (change in the range p_{gc} = 76.9-98.4 bar with an increase in t_{amb}). The oil has a negligible effect (for a change in *OCR* of 2%, around 0.1 bar).

5. Conclusion

In this paper, a typical single-stage transcritical R744 system is analyzed. As an extension to the standard cycle calculation, the impact of the compressor's isentropic efficiency and the influence of oil are included, aiming for a better approximation of the system's real operation.

Furthermore, a detailed calculation method is presented, which considers five different equations for isentropic efficiency from the available literature and the influence of oil while varying *OCR* values. The analysis was performed for steady-state conditions corresponding to the system's real operation. In the oil effect part of the analysis, only the optimal conditions (maximum *COP*) were considered, with variations in ambient air temperature, evaporation temperature, and *OCR*. It can be seen that all parameters directly influence the optimal pressure.

The analysis results showed the following:

- the use of η_{is} leads to a reduction in efficiency, with *COP* deviations of δ = 15.4-39.9% compared to the isentropic process,
- with t_{amb} variation, increasing the oil amount leads to a reduction in *COP* in the range of δ = 5.9-27.8% compared to the cycle with pure refrigerant, and
- with t_e variation, increasing the oil amount leads to a reduction in *COP* in the range of δ = 5.7-27.5% compared to the cycle with pure refrigerant.

In conclusion, the impacts of isentropic efficiency and oil significantly contribute to a better description of the real cycle of the refrigeration system. In the absence of relevant data, carefully adopted values can contribute to a more accurate representation. For more precise results, future work should include detailed oil-refrigerant interaction models and thermophysical property data. As experimental data are still limited, this study will be extended using measurements from a laboratory single-stage system. Although validation using data from real supermarket systems and manufacturers was considered, it has been postponed due to limited access to detailed proprietary information. From an engineering perspective, it has been shown that the proper selection of the compressor and oil separation system plays a crucial role in improving the system's efficiency.

Nomenclature

g	- specific gravity, [-]	*m*	- medium
h	- specific enthalpy, [kJkg^{-1}]	*mix*	- mixture
m	- mass, [kg]	*o*	- oil
p	- pressure, [bar]	*out*	- outlet
s	- specific entropy, [kJkg^{-1}K^{-1}]	*r*	- refrigerant
SH	- superheat, [°C]	*ref*	- reference value
t	- temperature, [°C]	*tot*	- total
x	- vapor quality, [-]	*v*	- vapor
Greek symbols		*Acronyms*	
δ	- percentage deviation, [%]	COP	- Coefficient of Performance
Δ	- difference, [-]	GWP	- Global Warming Potential
η	- isentropic efficiency, [-]	HFO	- Hydrofluoroolefin
ρ	- density, [kgm^{-3}]	LT	- Low-temperature
Subscripts		MT	- Medium-temperature
amb	- ambient	OCR	- Oil Circulation Ratio
e	- evaporator	PAG	- Polyalkylene Glycol
gc	- gas cooler	PAO	- Polyalphaolefin
is	- isentropic process	POE	- Polyolester
l	- liquid		

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Submitted: 15.3.2025.

Revised: 20.6.2025.

Accepted: 18.7.2025.