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Exergy analysis of CO_2 heat pump systems

A. Papadaki, A. Stegou-Sagia

School of Mechanical Engineering, Department of Thermal Engineering, National Technical University of Athens, 9 Iroon Polytechniou Str., Zografou 15780, Athens, Greece.

Abstract
Carbon dioxide (CO_2, R744), a natural refrigerant of beneficial properties found everywhere in our ambiance, can provide answer to the environmental problems caused by other refrigerants' use. The intention of this work is to outline the variation of exergy efficiency factor, COP and exergy flow related to the use of CO_2 in two stage and single stage heat pumps. The relevant mathematical models to the thermodynamic cycles were developed and an attempt was made for our efficiency and exergy losses results to be displayed. Moreover, fundamental process and system design issues of the applicable CO_2 heat pumps cycles were inaugurated, along with their properties and characteristics, comparing CO_2 use to that of R22 and its substitutes R407C and R410A applied in relevant conditions. Since exergy analysis is important theoretical basis for optimizing the systems operation and minimizing the losses, the results of this paper will advance the systems' design and performance.

Keywords: Exergy; Carbon dioxide; Heat pumps; Exergy analysis; Single stage cycle; Two stage cycle.

1. Introduction
Carbon dioxide is one of the most feasible answers to the contribution of the fluorocarbon refrigerants to global warming and ozone depletion, being a natural refrigerant with zero ODP (Ozone depletion potential), negligible GWP (Global warming potential), and very low cost. Global warming effect is considered to be the most prominent problem of the world climate. Refrigerants that are utilized in the heat and cooling systems have quite higher GWP than CO_2. Even refrigerants that were considered ozone layer friendly, such as HFC-134a, have GWP of many times greater than CO_2's (in HFC-134a is 1300 times) [1, 2]. In addition carbon dioxide (CO_2) is not toxic, flammable or corrosive. It is inexpensive and readily available. After the Montreal Protocol the interest for CO_2 cycles was so great that a large number of research developments have been commenced for the production of carbon dioxide's refrigeration system components.

1.1 CO_2 properties
Carbon dioxide furthermore has two exceptional properties, its most remarkable one being its low critical temperature T_{crit}, of 31.1◦C, compared to conventional refrigerants and working close or even above the critical pressure P_{crit} of 73.8 bar in vapour compression systems functioning in normal ambient temperatures [3, 4].
In a subcritical heat pump cycle, such low critical temperature is considered an inconvenience as heat cannot be delivered at temperatures greater than the critical temperature limiting consequently the operating temperature range. Additionally, heating capacity and the performance of the system are

relegated at temperatures inferior but close to T_{crit}, since the enthalpy of vaporization then is reduced [5], making the operation of a conventional heat pump avoidable at a heat rejection temperature near T_{crit}. Carbon dioxide's low critical temperature provides the opportunity to operate in a transcritical manner. In a transcritical heat pump, heat rejection (gas cooler) is operated above the critical pressure, heat delivery temperatures are no longer limited by T_{crit} and the evaporator is operated below that and for this reason the cycle is identified as transcritical.
The other unique property of CO_2 is the high working pressure required to use under typical heat pump conditions. Heat pump systems, both sub and transcritical, using CO_2, work at greater pressures than with the majority of other refrigerants. The operational pressures of subcritical CO_2 heat pumps reach as high as 60–70 bar, whereas for the transcritical pressures vary from 80 to 110 bar or even more. Although high pressure defies compressors' capability and components' robustness, it presents some benefits as well, providing to CO_2 a relatively high vapor density and an equally high volumetric heating capacity. This attribution offers the option for CO_2 to have a smaller working volume cycled in order to attain the same heating demand which permit the use of smaller components and more compact systems [3].
Nevertheless, the most important disadvantage of CO_2 cycle is that owing to huge expansion loss compared to conventional refrigerants' cycle it presents lower COP making the modifications of the cycle crucial [6]. Lorentzen [4] described more than a few customized cycles comprising of two-stage internal 'subcooling' and expansion options. By modifying the basic single-stage transcritical cycle a lot can be achieved. Some adaptations that are promising are dividing of flows, expansion via work generation instead of throttling, staging compression and expansion and the use of internal heat exchange. Trying to obtain higher efficiency values, we will employ the modification of the two-stage compression of the CO_2 with intercooling. Then we will compare these results to the equivalents of the single stage CO_2 and conventional vapour compression cycles. In order to model the total systems, and thereby investigate the possible operating conditions with replacement refrigerant mixtures, a computer code was created.

1.2 First and second law analysis
Studying the inefficiencies of existing systems our work focuses on the understanding of heat pumps cycles, their efficiencies and potentials for improvement, based on First and Second Laws of Thermodynamics. COP is used to evaluate performance of air-conditioning or heat pump from the viewpoint of the First Law of Thermodynamics. Exergy, being presented in an amount of works [7-13] corresponds quantitatively to the useful part of energy, the maximum possible amount of work a system, a flow of matter or energy can produce as it comes to equilibrium with an appointed reference environment. Exergy analysis combines the conservation of mass and energy principles with the second law of thermodynamics for the design of more efficient and environmental friendly systems. While efficiencies using energy are ambiguous for not being measures of "an approach to an ideal", exergy efficiencies are considered as such, measuring, in a way, the potential of the system for improvement [12].

2. Modelling of operation
2.1 Conventional heat pump's model
Figure 1 shows the heat pump's vapour / transcritical CO_2 compression cycle flow chart. The working fluid moves from the evaporator, which is connected to the low-temperature heat source into the compressor as a superheated vapour. Following, the compressed vapour, flows into the condenser which is connected to the high-temperature heat sink and respectively to the gas cooler for the CO_2. Here it condenses and afterwards, as a liquid, it undergoes expansion in the throttling valve. The throttled two-phase mixture, which is liquid for the most part, moves into the evaporator from which ensues the vapour that is then superheated and directed to the compressor to complete the flow cycle.

2.2 Transcritical two-stage CO_2 cycle with intercooling
Figure 2 shows the two stage CO_2 transcritical heat pump cycle with intercooling used. Here the saturated working fluid of state 2 moves from the evaporator into the low pressure (LP) compressor where it's compressed to state 3 before it enters the intercooler. There takes place the cooling, by external fluid, of the vapour which increases the mass of CO_2 vapour entering the high pressure (HP) compressor. Ambient air is taken as the external fluid. The saturated vapour from the intercooler at state 4 is compressed to state 5 and afterwards the super-critical vapour is cooled in the gas cooler to state 6.

CO_2 vapour is further cooled in the internal heat exchanger to state 7. CO_2 then expands in the expansion device to state 8 and evaporates to state 1 producing cooling effect. The internal exchanger in the system exists for system thermal efficiency improvement [14].

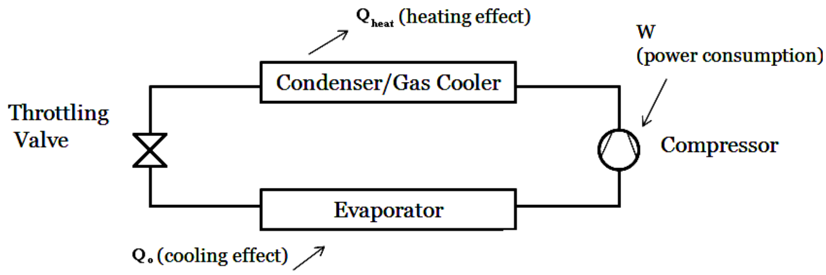

Figure 1. Schematic diagram of the single stage heat pump cycle

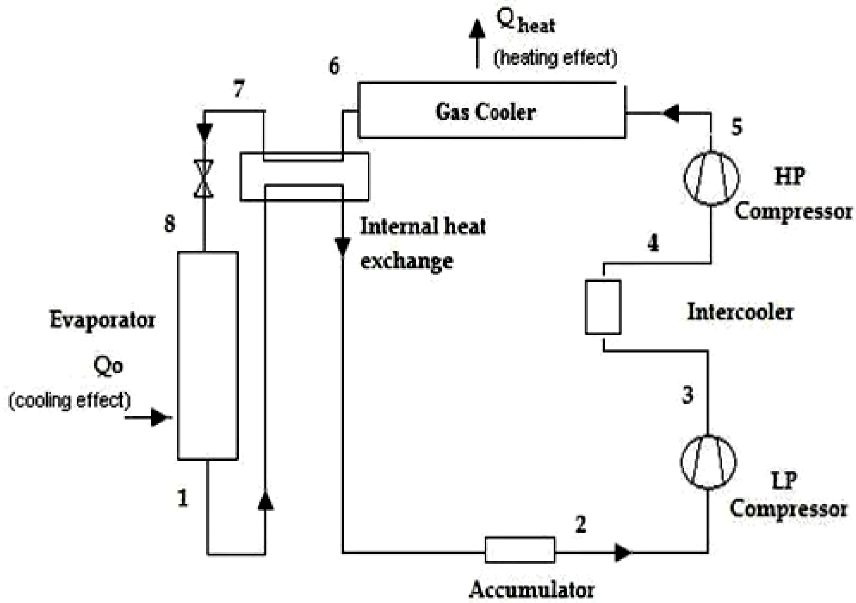

Figure 2. Schematic diagram of the two stage heat pump cycle with intercooling

2.3 Thermodynamic analysis

2.3.1 Single stage cycle

Based on the known equations for the exergy and energy analysis [15-17] of a heat pump cycle, as the one shown in Figure 1, we have:

The exergy efficiency factor ζ is

$$\zeta = COP.\left(\frac{T_{\mathrm{w}} - T_{\mathrm{a}}}{T_{\mathrm{w}}}\right) \tag{1}$$

with the coefficient of performance (COP) of the system being

$$COP = \frac{q}{\Delta e_{\mathrm{abs}} + \sum \Delta e_{\mathrm{loss}}} = \frac{(h_2 - h_4)}{(h_2 - h_4)\left(1 - \dfrac{T_{\mathrm{a}}}{T_{\mathrm{w}}}\right) + \sum \Delta e_{\mathrm{loss}}} \tag{2}$$

Exergy losses, for each component of the system are:

• Compression losses:

$$\Delta e_{\text{comp}} = T_{\text{a}}(s_2 - s_1) \tag{3}$$

• Additional losses due to the compressor motor:

$$\Delta e_{\text{mot}} = w\frac{1-\eta_{\text{mot}}}{\eta_{\text{mot}}}\frac{T_{\text{a}}}{T_{\text{w}}} \tag{4}$$

where η_{mot} is the compressor motor efficiency factor, w the specific compression power demand (h_2-h_1) and the heat from the heat pump motor absorbed by the heated substance [16].

• Condensation / gas cooler losses:

$$\Delta e_{\text{cond}} = (h_2 - h_4)\frac{T_{\text{a}}}{T_{\text{w}}} - T_{\text{a}}(s_2 - s_4) \tag{5}$$

• Evaporation losses:

$$\Delta e_{\text{evap}} = T_{\text{a}}(s_1 - s_5) - (h_1 - h_5) \tag{6}$$

• Throttling (isenthalpic process) losses:

$$\Delta e_{\text{thr}} = T_{\text{a}}(s_5 - s_4) \tag{7}$$

Therefore, summing up we obtain the total exergy loss:

$$\begin{aligned}
&\sum \Delta e_{\text{loss}} = \\
&\Delta e_{\text{comp}} + \Delta e_{\text{mot}} + \Delta e_{\text{cond}} + \Delta e_{\text{evap}} + \Delta e_{\text{thr}} = \\
&(h_5 - h_1) + \\
&\left[(h_2 - h_4) + (h_2 - h_1)\frac{1-\eta_{\text{mot}}}{\eta_{\text{mot}}}\right]\frac{T_{\text{a}}}{T_{\text{w}}}
\end{aligned} \tag{8}$$

The exergy efficiency factor is consequently given by the equation (1):

$$\zeta = \frac{(h_2 - h_4)(1 - \frac{T_{\text{a}}}{T_{\text{w}}})}{(h_2 - h_1)\left(1 + \frac{1-\eta_{\text{mot}}}{\eta_{\text{mot}}}\frac{T_{\text{a}}}{T_{\text{w}}}\right)} \tag{9}$$

The refrigerants compared to R744 are R407C and R410A and R22.

A variety of sources were used [18-24] to ensure the consistent application of property. The differences observed were minimal. It is taken into consideration in all relevant calculations the fact that R407C and R410A are non-azeotropic, since they show a different behaviour from pure substances [25].

Firstly, due to different evaporator and condenser inlet/outlet temperatures, we have to select condenser inlet temperature in opposition to the warm space temperature taking care of the condenser inlet and outlet temperatures to be sufficient so as to reject heat and finally liquid enthalpy at the expansion device and related property data being in position to achieve the suitable evaporator inlet temperature. The fluid behaves normally in all other points. Undeterred by the fact that this method of evaluation occupied in this study is not fully representative of a dynamic operation of a heat pump system, yet it sets up the foundations for understanding its thermodynamic performance.

2.3.2 Two stage cycle

The two-stage CO_2 transcritical heat pump cycle with intercooling is modelled modularly incorporating each individual process of the cycle. The state points in Figure 2 are defined as the conditions of the refrigerant characterized by its temperature, mass flow rate and quality.

Exergy losses, for each component of the system are:

• Compression losses:

$$\Delta e_{\mathrm{comp}} = \Delta e_{\mathrm{comp1}} + \Delta e_{\mathrm{comp2}} = T_{\mathrm{a}}(s_3 + s_5 - s_2 - s_4) \tag{10}$$

• Additional losses due to the compressors motors:

$$\Delta e_{\mathrm{mot}} = w \frac{1-\eta_{\mathrm{mot}}}{\eta_{\mathrm{mot}}} \frac{T_{\mathrm{a}}}{T_{\mathrm{w}}} \tag{11}$$

where η_{mot} is the compressors motor efficiency factor, w the specific compression power demand (h_5-h_4+h_3-h_2) and the heat from the heat pump motor absorbed by the heated substance.

• Intercooler losses

$$\Delta e_{\mathrm{ic}} = (h_3 - h_4)\frac{T_{\mathrm{a}}}{T_{\mathrm{w}}} - T_{\mathrm{a}}(s_3 - s_4) \tag{12}$$

• Gas cooler losses:

$$\Delta e_{\mathrm{gc}} = (h_5 - h_6)\frac{T_{\mathrm{a}}}{T_{\mathrm{w}}} - T_{\mathrm{a}}(s_5 - s_6) \tag{13}$$

• Evaporation losses:

$$\Delta e_{\mathrm{evap}} = T_{\mathrm{a}}(s_1 - s_8) - (h_1 - h_8) \tag{14}$$

• Expander valve (isenthalpic process) losses:

$$\Delta e_{\mathrm{ex}} = T_{\mathrm{a}}(s_8 - s_7) \tag{15}$$

• Internal heat exchanger:

$$\Delta e_{\mathrm{ihe}} = T_{\mathrm{a}}[(s_7 - s_6) - (s_1 - s_2)] \tag{16}$$

Therefore, summing up we obtain the total exergy loss:

$$\begin{aligned}&\sum \Delta e_{\mathrm{loss}} = \\ &\Delta e_{\mathrm{comp}} + \Delta e_{\mathrm{mot}} + \Delta e_{\mathrm{gc}} + \Delta e_{\mathrm{evap}} + \Delta e_{\mathrm{ihe}} = \\ &(h_8 - h_1) + \\ &\begin{bmatrix}(h_3 - h_4 + h_5 - h_6) + \\ (h_5 - h_4 + h_3 - h_2)\dfrac{1-\eta_{\mathrm{mot}}}{\eta_{\mathrm{mot}}}\end{bmatrix}\frac{T_{\mathrm{a}}}{T_{\mathrm{w}}}\end{aligned} \tag{17}$$

The exergy efficiency factor is consequently given by the equation (1):

$$\zeta = \frac{(h_3 - h_4 + h_5 - h_6)(1 - \frac{T_a}{T_w})}{(h_5 - h_4 + h_3 - h_2)\left(1 + \frac{1-\eta_{mot}}{\eta_{mot}}\frac{T_a}{T_w}\right)} \quad (18)$$

2.4 Assumptions

It is renowned that CO_2 refrigeration and air conditioning systems shows cooling COP more sensitive to ambient temperature variation than conventional systems, being therefore superior at sensible and low ambient temperature, and to some extent poorer at very high temperature. Consequently, it would be deceptive to base a comparison of CO_2 with the other refrigerants on design point conditions, which typically are at an extreme ambient temperature while the use of average seasonal conditions is wiser [26].

Our exergy efficiency, COP and exergy losses diagrams of the mixtures under consideration are schematized in comparison with the CO_2 and are plotted based on calculations, having taken into consideration the following assumptions:

The environmental temperature (T_a) is equal to 273 K [8, 27], while the temperature of the warm place (T_w) is considered 308 K [28]. The temperatures T_{con} and T_e are taken: T_{con} at the inlet of the condenser at the vapour saturation curve for R22, R407C and R410A and at the inlet of the gas cooler for R744, while T_e at the exit of the evaporator in the superheat region. Pressure drops in evaporator are for R22 [29] and R407C [30, 31] 135 kPa, for R410A [32] 85 kPa and for R744 [33] 100kPa, while during condensation the pressure drop varies for R22 (Judge et al, 2001) from 46 to 52 kPa, for R407C [30, 31] from 40 to 46 kPa and for R410A [31, 32] from 32 to 35 kPa, lessening with increasing condensation temperature; whereas correspondingly for R744 [28, 34] the already small (1 to 3 kPa as shown in [35]) pressure drop during cooling process of supercritical CO_2 decreases as inlet pressure of gas cooler increases having a temperature glide of approximately 61K [28]. In addition the isentropic compressor efficiency factor is chosen as 0,75 and the compressor motor efficiency factor as 0,85 in a endeavor to maintain a logical price for the evaluation.

The evaporator temperature (T_e) is taken as 263K for all condensing temperatures whilst condensation temperature (T_{con}) for the mixtures and the outlet temperature of the CO_2 gas cooler is ranging from 313 to 328 K. Accordingly the temperature ratio $\tau = (T_{con}/T_e)$ or $\tau = (T_{gc}/T_e)$ varies within the range of 1.19 to 1.25.

The featured two-stage CO_2 transcritical cycle configuration is solely a theoretical one to present the basis for performance comparison with other refrigerants. It is simulated and its performance is evaluated on the basis of maximum combined COP to obtain the optimum gas cooler and in-between pressures. These values are obtained for various operating conditions along with simultaneous variation of the compressors discharge pressure and intermediate pressure having a step size of 0.5 bar for each. The performance is evaluated on various evaporator temperatures T_e from 223 K to 243 K) and gas cooler outlet temperatures T_{gc} (308 K to 333 K) [36].

3. Results and discussion

The results attained in this analysis are comparison of refrigerants for exergy efficiency, COP and exergy losses (Figures 3 to 5). Properties of R22 are illustrated in plots by bold continuous lines, while R407C by thin discontinuous lines, R410A by thin continuous lines, R744 (single stage) by dotted lines and R744 (two stage) bold dotted lines.

Figure 3 shows the exergy efficiency factor as a function of temperature ratio τ. Exergy efficiency decreases when the temperature ratio τ increases. The curves' hollows are facing upwards.

The single stage heat pump working with R744 has the least favourable exergy behaviour with an exergy efficiency of 13% at the temperature ratio of $\tau = 1.25$ and 28% at the temperature ratio of $\tau = 1.19$. While the R744 of the two-stage transcritical heat pump features far better exergy performance compared to the latter, with an exergy efficiency of 31% at a temperature ratio of $\tau = 1.25$ and 38% at a temperature ratio of $\tau = 1.19$, demonstrating less variation on its performance with the change of temperature ratio.

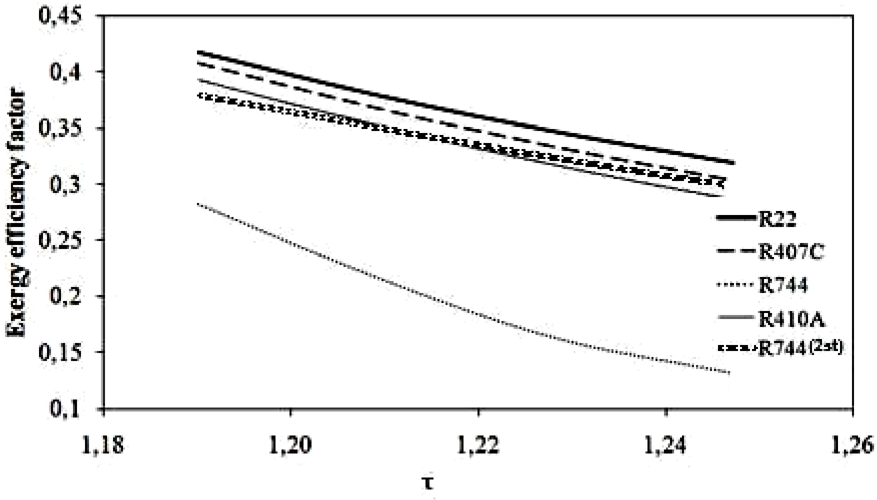

Figure 3. Variation of exergy efficiency factor for various temperature ratios τ

R22, on the other hand, presents the best exergy behaviour of all with an exergy efficiency of 42% at a temperature ratio of τ=1.19 and 33% at τ=1.25, followed by R407C (ζ = 41% at τ=1.19 and ζ = 32% at τ=1.25) and R410A (ζ= 40% at τ=1.19 and ζ = 29.5% at τ=1.25).
Figure 4 shows the disparity of COP of the heat pump system for each working refrigerant related to the temperature ratio τ, decreasing while the latter lifting as exergy efficiency factor does. COP ranges from 1.15 at temperature ratio of τ=1.19 (for R744) to 3.77 (for R22) at τ=1.25. There is a pointed increase in COP for the two-stage R744 system compared to the single stage one. Here, the single stage working R744 has likewise the worst behaviour, with COP to vary between 2.48 (at τ=1.19) and 1.15 (at τ=1.25), whilst the R744 of the two-stage transcritical heat pump features once more better comportment, with COP fluctuating amid 3.40 (at τ=1.19) and 2.70 (at τ=1.25).

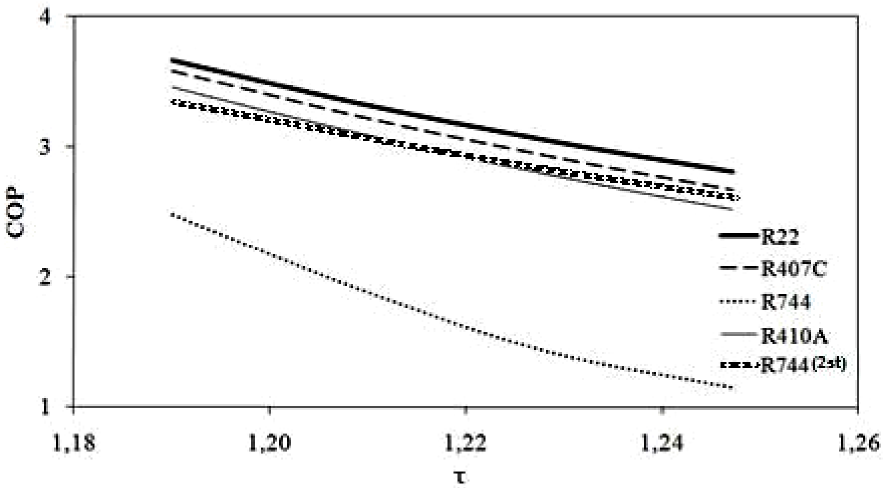

Figure 4. Variation of COP for various temperature ratios τ

The optimum performance is displayed yet again by R22, with COP of 3.77 at a temperature ratio of τ=1.19 and 2.88 at τ=1.25, followed by R407C with COP of 3.70 at a temperature ratio of τ=1.19 and 2.74 at τ=1.25 and R410A with COP of 3.57 at a temperature ratio of τ=1.19 and 2.58 at τ=1.25. R22 may seem more attractive to use from the efficiency aspect, however we have to bear in mind that it constitutes a harmful effect on the ozone layer with the result of extreme UV levels conducing to further environmental damage and several deadlines have been arranged depending on the country for complete R22 replacement in accordance to the terms established by the Montreal Protocol meetings.
The prices for COP and exergy efficiency factor are in agreement with those of Robinson and Groll [37] at the equivalent conditions' region.

Figure 5 presents the percentage of the major exergy losses for the two CO_2 systems. These are of the gas cooler and of the compressor and we can conclude that for the two stage CO_2 transcritical cycle the losses lessen dramatically. For the single stage heat pump working with R744 the compressor accounts for approximately 49% of the total cycle irreversibility and the gas cooler for the 25%, while respectively the percentage of exergy losses in the two-stage transcritical heat pump is 32% for the compressor and 20% for the gas cooler.

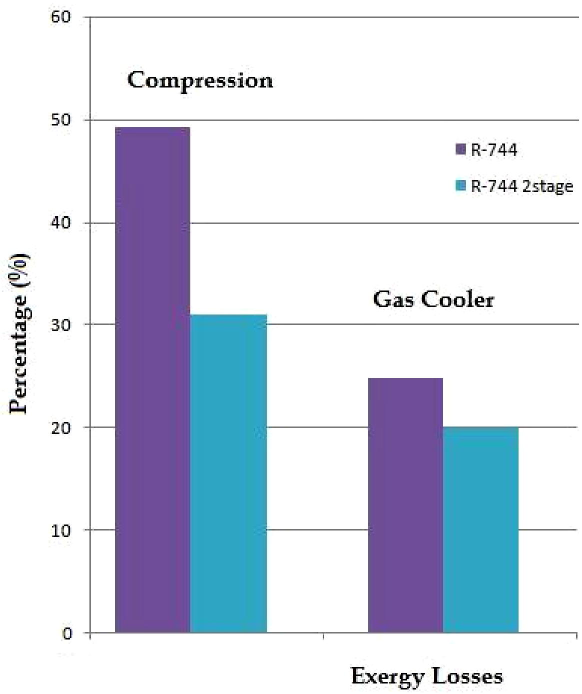

Figure 5. Exergy losses of the systems' components

As pointed out by Dincer and Rosen [38] "Exergy efficiency weights energy flows by accounting for each in terms of availability. It stresses that both losses and internal irreversibilities need to be dealt with to improve performance" and by Moran and Shapiro [39] "Exergy analysis is particularly suited for furthering the goal of more efficient energy use, since it enables the locations, types, and true magnitudes of waste and lost to be determined". Following the above described study the behaviour of the system can be improved, minimising individual exergy loss of each component and maximising efficiencies. Compressor efficiency is a major factor in enhancing the performance of the system, the smaller the compressor, the more prominent the compression losses. Generally speaking throttling losses can be reduced minimising the temperature difference before and after the throttling valve, as well as by decreasing the temperature differences in evaporator and condenser. This would also produce lower compression losses.

4. Conclusions

In this report we have made an effort to elucidate the diversity of the alternatively used refrigerant mixtures R407C and R410A replacing R22, and R744 replacing all of them in the field of exergy efficiency, COP and exergy losses depending on temperature ratio τ, for constant warm place temperature. The best exergy behaviour of all is presented by R22, with an exergy efficiency of 42% at a temperature ratio of τ=1.19. R744 may seem to fall short in comparison to the rest refrigerants for some conditions, nevertheless it is the most environment friendly of all and based on that and on its beneficial potentialities its use signifies a "new" ecological era for the field. As stated before, one of the downsides associated with transcritical cycles is that the system operates at a very high discharge pressure. There is a sharp reduction in optimum discharge pressure by adopting staging in compression. Inter-stage pressure is one of the most critical parameters for optimizing COP values. Moreover, by using highly efficient system components, the transcritical two-stage CO2 systems can be used more effectively. Two-stage transcritical heat pump working with R744 features far better exergy performance compared to the

single stage cycle with a pointed increase in COP for the two-stage R744 system. Furthermore for the two stage R744 transcritical cycle the losses lessen dramatically.
The evolution of exergy efficiency factor and COP are illustrated and collated in diagrams so as to clarify the differences of alternative refrigerants more accurately.

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Argyro Papadaki, Dipl. Mechanical Engineer, National Technical University of Athens, Greece, 2005; Post Graduate Specialization Diploma (MSc), Automation Systems (Control and Robotic Systems Direction), National Technical University of Athens, 2008; Post Graduate Specialization Diploma (MSc), Engineering Project Management, Hellenic Open University, Patras, 2012; PhD Candidate, Mechanical Engineering, National Technical University of Athens; Appointments: Research Assistant, Department of Thermal Engineering, National Technical University of Athens. Publications: Papers to international journals and international conferences; Memberships: Greek Technical Chamber, 2005.
E-mail address: apapadaki@gmail.com

Athina Stegou-Sagia, Dipl. Mechanical Engineer, National Technical University of Athens, Greece, 1979; PhD, Thermodynamics, Properties and Phase Change of Refrigerant Mixtures, 1986. Appointments: Ministry of Public Works, 1980-81; Researcher, 1981-88, Lecturer, 1988-92, Assistant Professor, 1992-97, Associate Professor, 1997-2006, Professor, 2006-, National Technical University of Athens; Director, Laboratory of Thermal Processes and Heat Transfer, National Technical University of Athens; Fellow, European Community; Experienced Researcher, Queen Mary and Westfield College, University of London, England, 1993-94. Publications: Numerous papers to international journals and international conferences; Books in Heat and Mass Transfer - Unit Operations. Memberships: Greek Technical Chamber, 1979-; Executive Council, Greek Association of Computational Mechanics, Athens, 1994-96, 1998-2000; National Technical University of Athens, The Senate, 2003-04.
E-mail address: asagia@central.ntua.com