EXERGY ANALYSIS AND REFRIGERANT EFFECT ON THE OPERATION AND PERFORMANCE LIMITS OF A ONE STAGE VAPOR COMPRESSION REFRIGERATION SYSTEM

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Rezumat. Lucrarea prezinta o analiza comparativa a efectului agentului frigorific utilizat asupra functionarii si performantelor unei instalatii frigorifice cu comprimare mecanica de vapori intr-o treapta. Parametrii si factorii ce influenteaza performantele (din punct de vedere al puterii frigorifice, coeficientului de performanta, consumului de energie, etc) sunt estimate pe baza analizei exergetice. Diverse studii de sensibilitate sunt prezentate comparativ pentru cativa agenti frigorifici (R22, R134a, R717, R507a, R404a). Sunt prezentate rezultate grafice si numerice. Influenta raportului de comprimare este evidentiata pentru functionarea sistemului cu acesti agenti frigorifici, afectand regimul de functionare (temperatura maxima acceptata), respectiv performantele sistemului. De asemenea, se studiaza efectul subracirii si supraincalzirii. In concluzie, se prezinta o analiza comparativa a coeficientului de performanta definit pe baza energetica si a randamentului exergetic.
Cuvinte cheie: comprimare mecanica, instalatie frigorifica, analiza exergetica, subracire si supraincalzire, efectul agentului frigorific.

Abstract. This paper deals with a comparative analysis of the refrigerant impact on the operation and performances of a one stage vapor compression refrigeration system. Parameters and factors affecting the performances (in terms of refrigeration power, coefficient of performance, mechanical work consumption, etc) are evaluated on the basis of an exergy analysis. Different sensitivity studies are presented in a comparative manner for some refrigerants (R22, R134a, R717, R507a, R404a). Graphical and numerical results are included. The effect of compression ratio is emphasized for the system operation working with these refrigerants, affecting the operation regime (maximum accepted temperature), respectively the performances of the system. Also the effects of subcooling and superheating are shown. As conclusion, a comparative analysis between energetic base COP and exergetic efficiency is presented.
Keywords: vapor compression, refrigeration system, exergy analysis, subcooling and superheating, refrigerant impact.

1. INTRODUCTION

This paper deals with a comparative analysis of the refrigerant impact on the operation and performances of a one stage vapor compression refrigeration system. The authors have chosen this type of system, the simplest one, since the aim of the paper is to present and propose an analysis model for comparing the operation of a VCR System with different refigerants, from performances point of view and from limitations in terms of compression ratio. As stated above, the aim is to undersand the mathematical procedure and not to focus on a complex system for the numerical application.

It is well known the fact that after 90's CFC and HCFC refrigerants have been forbiden due to clorine content and their high ozone depleting potential (ODP) and global warming potential (GWP). Thus, HFC refrigerants are used nowadays, presenting a much lower GWP value, but still high with respect to non-flourine refrigerants.

Many research papers have been published on this subject, of replacing “old” refrigerants with “new” ones [1-6]. Lately, many papers focused on researches about finding better and better refrigerants or mixtures, considering different criteria, as for example: ODP and GWP values, performances (COP, TEWI analysis, refrigerating power, compressor consumption, exergetic efficiency), flammability, mass flow rates limitations for safety operation, miscibility with oil, etc [1,3]. Also exergetic analyses and thermoeconomic optimization procedures have been published in the field [7-9].

This paper presents a comparative analysis of five refrigerants working in a one stage VCRS with subcooling and superheating. These five refrigerants are: 1,1,1,2-tetrafluoroethane (HFC-134a), chlorodifluoromethane (R22), ammonia (R717), a near-azeotropic blend (R404a) and an azeotropic blend (R507a). R507a is an azeotropic blend of pentafluorethane R125 and 1,1,1-trifluoroethane R143a with mass percentages of 50% / 50%. R404a is a near-azeotropic blend of R125 / R143a / R134a with mass percentages of 44% / 52% / 4%. Blends do not necessarily remain at constant temperature during constant pressure evaporation or condensation. R134a has the advantage of presenting zero ODP, but still a quite high value for GWP (1300 times higher that that of CO_2), being a chosen substitute for the "old" R12. Spauschus [2] presented the compressor and refrigerant system requirements for substituting R12 with R134a. Havelsky [6] also studied this replacement and others, as R22, R401a, etc. Experimental results on simple VCRS [1,5] have also been presented regarding the behavior of the system working with different refrigerants or mixtures.

We will focus our attention on the operating limits in terms of compression ratio, on system performances when wotking with different refrigerants and on exergy destruction rates in each component of the system to compare the effect of the refrigerants on the system operation.

2. SYSTEM DESCRIPTION

A one stage vapor compression refrigeration system is considered as numerical exemplification of the proposed study. The system is composed by a mechanical piston compressor, a condenser, a throttling valve and an evaporator, as shown in Figure 1. An usual operating cycle, with superheating and subcooling, is represented in p-h coordinates in Figure 2.

The refrigerant enters the compressor at state 1, with a superheating degree ΔT_{SH} with respect to the evaporation temperature T_V. It follows the irreversible compression process 1-2, characterized by an increase in entropy from state 2s (adiabatic reversible compression) to state 2. The refrigerant leaves the compressor as superheated vapor at pressure p_C and enters the condenser and subcooler, arriving in state 3 as subcooled liquid, that is further throttled during the process 3-4. Its pressure is the vaporization pressure p_V and the cycle is closed by a vaporization process 4-1 in the evaporator and superheater.

The simplicity of the system allows the reader to focus on the mathematical model, rather than on its operation.

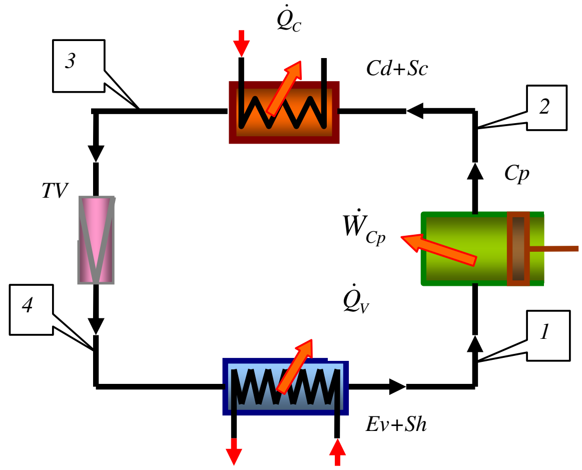

Fig. 1. One stage VCRS

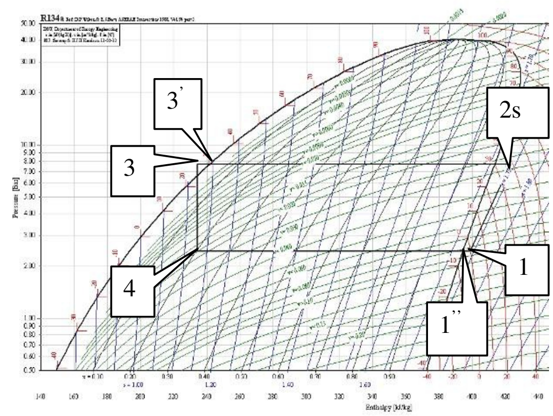

Fig. 2. Usual operating cycle, with superheating and subcooling (p-h coordinates)

3. MATHEMATICAL MODEL

The system is analyzed both from energetic and exergetic points of view. The reader should notice that the two approaches form a more powerful tool in the study and optimization of such systems and one does not exclude the other.

3.1. Energetic approach

This analysis is applied either to each device (seen as a control volume) or to the entire system (a control mass).

It is based on the First Law of Thermodynamics, whose mathematical expression for a control volume is:

$$\frac{dE_{CV}}{d\tau} = \sum_i \left(h + \frac{w^2}{2} + gz\right)_i \dot{m}_i - \sum_o \left(h + \frac{w^2}{2} + gz\right)_o \dot{m}_o + \dot{Q}_{CV} - \dot{W}_{CV} \quad (1)$$

where E represents system energy [J], τ stands for time [s], h is the specific enthalpy of refrigerant[J/kg], $w^2/2$ is the specific kinetic energy [J/kg], gz is the specific potential energy [J/kg], $\dot{m}$

is the mass flow rate of refrigerant [kg/s], $\dot{Q}$ and $\dot{W}$ [W] are the energetic exchanges of the control volume with its surroundings in form of heat flux and work rate (power).

The subscripts i and o stands for inlet and outlet states, respectively.

For steady state operation, eq.(1) becomes:

$$\dot{Q}_{CV} = \sum_o \left(h + \frac{w^2}{2} + gz \right)_o \dot{m}_o - \sum_i \left(h + \frac{w^2}{2} + gz \right)_i \dot{m}_i + \dot{W}_{CV} \quad (2)$$

Neglecting the variation of kinetic and potential energies (which is an appropriate assumption in a VCRS), equation (2) becomes:

$$\dot{Q}_{CV} - \dot{W}_{CV} = \sum_o (\dot{m}h)_o - \sum_i (\dot{m}h)_i \quad (3)$$

which is applied to each device of the system:

a) for the evaporator (phase change process in a heat exchanger):

$$\dot{Q}_V = \dot{m}(h_1 - h_4) \quad (4)$$

where $\dot{Q}_V$ [W] represents the refrigerating load, while the specific enthalpy h is determined by using the Engineering Equation Solver (EES) software;

b) for the condenser (phase change process in a heat exchanger):

$$\dot{Q}_C = \dot{m}(h_3 - h_2) \quad (5)$$

where $\dot{Q}_C$ [W]is the condenser thermal load;

c) for the compressor (compression process):

$$\dot{W}_{Cp} = \dot{m}(h_1 - h_2) + \dot{Q}_{Cp} \quad (6)$$

where $\dot{Q}_{Cp}$ is the heat rate generated in a non – adiabatic compressor, estimated as:

$$\dot{Q}_{Cp} = \dot{m}(h_2 - h_{2s}) \quad (6')$$

d) for the throttling valve (throttle process, isenthalpic):

$$\dot{m}h_3 = \dot{m}h_4 \quad (7)$$

The sign convention is the English one, namely all heat rejected by the system is negative and all heat received by the system is positive; the consumed mechanical work is negative.

The energetic efficiency of the system is measured by the coefficient of performance:

$$COP = \frac{usefull\ effect}{consumption} = \frac{\dot{Q}_V}{\left|\dot{W}_{Cp}\right|} \quad (8)$$

By applying the Second Law of Thermodynamics to each control volume, one could find the entropy generation in the system:

$$\dot{S}_{gen} = \frac{dS_{CV}}{d\tau} + \sum_o (\dot{m}s)_o - \sum_i (\dot{m}s)_i - \sum_j \left(\frac{\dot{Q}_j}{T_j} \right)_{ext} \quad (9)$$

where $dS_{CV}/d\tau$ is zero for a steady state operation regime; $\dot{Q}_j/T_j$ represents the heat flux $\dot{Q}_j$ exchanged by the system with the surroundings at T_j temperature level and s [J/(kgK)] represents the specific entropy of the refrigerant.

By applying Eq. (9) to each device of the system, one gets:

a) for the evaporator:

$$\dot{S}_{gen,V} = \dot{m}(s_1 - s_4) - \frac{\dot{Q}_V}{T_V} \quad (10)$$

where T_V [K] is the vaporization temperature;

b) for the condenser:

$$\dot{S}_{gen,C} = \dot{m}(s_3 - s_2) + \frac{\left|\dot{Q}_C\right|}{T_0} \quad (11)$$

c) for the compressor:

$$\dot{S}_{gen,Cp} = \dot{m}(s_2 - s_1) + \frac{\left|\dot{Q}_{Cp}\right|}{T_0} \quad (12)$$

d) for the throttling valve:

$$\dot{S}_{gen,TV} = \dot{m}(s_4 - s_3) \quad (13)$$

The specific entropy s [J/kgK] is determined for each state of the refrigerant by using the EES software.

In Table 1, one can notice how these state parameters of the refrigerant are determined in each state during the operating cycle.

Table 1

Determination of state parameters

State	Independent parameters	Specific parameters
1”	p_V, $x_{1''} = 1$	$h_{1''}$, $s_{1''}$, $v_{1''}$
1	p_V, $t_1 = t_V + \Delta t_{SH}$	h_1, s_1, v_1
$2s$	p_C, $s_{2s} = s_1$	h_{2s}, s_{2s}, v_{2s}
2	p_C, h_2	h_2, s_2, v_2
$3'$	p_C, $x_{3'} = 0$	$h_{3'}$, $s_{3'}$, $v_{3'}$
3	p_C, $t_3 = t_C - \Delta t_{SC}$	h_3, s_3, v_3
4	p_V, $h_4 = h_3$	h_4, s_4, v_4

where $h[J/kg]$, $s[J/kgK]$, $v[m^3/kg]$.

The enthalpy of the real state 2 is determined by introducing the compressor adiabatic efficiency, namely: $h_2 = h_1 + \frac{\eta_{Cp}}{h_{2s} - h_1}$, where the adiabatic efficiency is estimated by using an empirical relation: $\eta_{Cp} \approx 1 - 0.05 \frac{p_C}{p_V}$ [10].

By concluding this approach, all specific properties are determined in all states of the cycle, and also all energetic exchanges of the system with its surroundings are computed.

3.2. Exergetic approach

The exergetic balance equation for a control volume is:

$$\frac{dEx_{CV}}{d\tau}=\sum_j \dot{E}x_{Q_j}-\left(\sum \dot{W}-p_0\frac{dV}{d\tau}\right)+\sum_i(\dot{m}ex)_i-\sum_o(\dot{m}ex)_o-\sum I \quad (14)$$

where $\dot{I}$ [W] represents the exergy distruction rate, ex [J/kg] represents the specific exergy of the refrigerant, $\dot{E}x_{Q_j}$ [W] is the heat exergy rate.

For steady state operation, Eq. (14) becomes:

$$\sum \dot{I}=\sum_i(\dot{m}ex)_i-\sum_o(\dot{m}ex)_o+\sum_j \dot{E}x_{Q_j}-\sum \dot{W} \quad (15)$$

The specific exergy of the refrigerant is computed as:

$$ex=(h-h_0)-T_0(s-s_0) \quad (16)$$

The heat exergy rate $\dot{E}x_Q$ in Eq. (15) is expressed as:

$$\dot{E}x_Q=\dot{Q}\left(1-\frac{T_0}{T_{boundary}}\right) \quad (17)$$

In Eq. (16) and (17), the subscript "*0*" denotes the extensive parameters of the system brought in the restrictive dead state.

The standard parameters of the environment are: $T_0=299{,}15K$, $p_0=1bar$.

According to Guy – Stodola theorem, the exergy distruction rate is also computed by:

$$\dot{I}=T_0\dot{S}_{gen}. \quad (18)$$

or can be determined by applying Eq. (15) to each component of the system:

a) for the evaporator:

$$\dot{I}_V=\dot{m}ex_4-\dot{m}ex_1+\dot{Q}_V\left(1-\frac{T_0}{T_V}\right) \quad (19)$$

By replacing Eq. (16) in (19) and neglecting the variation of kinetic and potential energies, one gets:

$$\dot{I}_V=\dot{m}[(h_4-T_0s_4)-(h_1-T_0s_1)]+\dot{Q}_V\left(1-\frac{T_0}{T_V}\right) \quad (20)$$

By applying the same reasoning, one gets:

b) for the condenser:

$$\dot{I}_C=\dot{m}ex_2-\dot{m}ex_3==\dot{m}[(h_2-T_0s_2)-(h_3-T_0s_3)] \quad (21)$$

c) for the compressor:

$$\dot{I}_{Cp}=\dot{m}ex_1-\dot{m}ex_2+\left|\dot{W}_{Cp}\right|==\dot{m}[(h_1-T_0s_1)-(h_2-T_0s_2)]+\left|\dot{W}_{Cp}\right| \quad (22)$$

d) for the throttling valve:

$$\dot{I}_{TV}=\dot{m}ex_3-\dot{m}ex_4=\dot{m}[(h_3-T_0s_3)-(h_4-T_0s_4)]==\dot{m}T_0(s_4-s_3) \quad (23)$$

The overall exergy distribution rate is:

$$\dot{I}_{TOT}=\dot{I}_V+\dot{I}_C+\dot{I}_{Cp}+\dot{I}_{TV} \quad (24)$$

The exergetic efficiency of the system is evaluated:

$$\eta_{ex}=\frac{\dot{E}x_P}{\dot{E}x_F} \quad (25)$$

where the product exergy rate is:

$$\dot{E}x_P=\dot{Q}_V\left(1-\frac{T_0}{T_V}\right) \quad (26)$$

and the fuel exergy rate is:

$$\dot{E}x_F=\left|\dot{W}_{Cp}\right| \quad (27)$$

The exergy balance equation applied to the whole system gives:

$$\dot{E}x_F=\dot{E}x_P+\dot{I}_{TOT} \quad (28)$$

By combining Eq. (26) – (29), one gets:

$$\eta_{ex}=1-\frac{\dot{I}_{TOT}}{\dot{E}x_F}=\frac{\dot{Q}_V}{\left|\dot{W}_{Cp}\right|}\left(1-\frac{T_0}{T_V}\right) \quad (29)$$

Remembering Eq. (8) and the expression of *COP* for an inverse Carnot cycle working between T_0 and T_V temperature levels, one obtains:

$$\eta_{ex}=\frac{COP}{COP_{CC}\Big|_{T_V}^{T_0}} \quad (30)$$

This exergetic efficiency measures the system behavior with respect to a corresponding Carnot cycle working between the same vaporization temperature and the surroundings one. Thus, it measures the irreversibilities of the real operation with respect to a theoretical possible operation.

Graphical results are presented for these system performances, in a comparative manner for the studied refrigerants and for different sets of parameters values in some sensitivity studies.

4. RESULTS

In order to make a comparison of the refrigerant effect on the operation of the VCRS, the compression ratio, β, is varied between 2 and 16. The first check is on the refrigerant temperature at the compressor outlet, t_2, represented in Figure 3. One may suppose that the maximum allowed limit of operation is 140°C for this temperature, due to compressor oil inflammability limit. Thus, in this figure one could compare the maximum allowed compression ratio for the studied refrigerants. The ammonia (R717) has the lower limit in terms of β, as known. It is also interesting to notice the fact that R134a, which replaced R22 in domestic refrigerators, has a higher limit than the last one. The two near (azeotropic) mixtures are very close in terms of maximum compression ratio, as it was expected.

The corresponding vaporization temperatures are shown in Figure 4, where the dotted flashes represent the values corresponding to maximum compression ratio. One can notice that the system operating at a compression ratio of 5 could achieve a vaporization temperature of -10°C when operating with ammonia (R717), -15°C for R22 and 0°C for R134a. Also, when operating at the

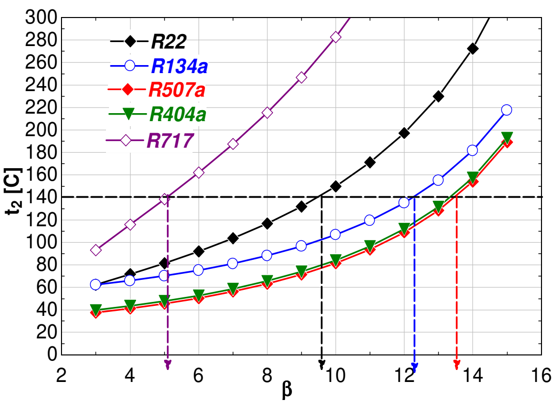

Fig. 3. Refrigerant effect on the maximum allowed compression ratio

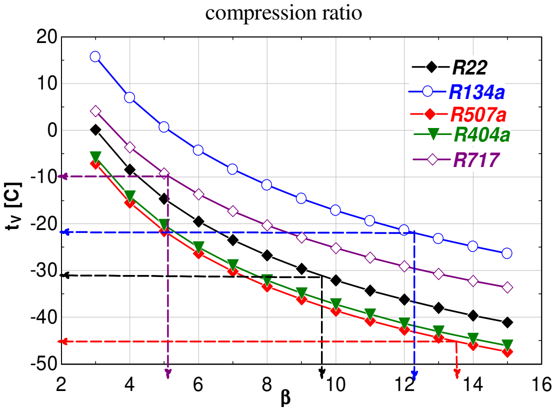

Fig. 4. Refrigerant indirect effect on the maximum attainable vaporization temperature

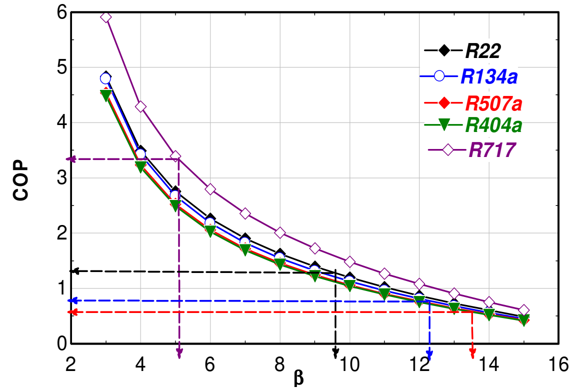

Fig. 5. Refrigerant indirect effect on the maximum attainable COP

maximum allowed value of β, R134a could lead to a possible minimum value of -20°C with respect to -30°C for R22 and -45 for the azeotropic mixtures. Ammonia (R717) is restricted to -10°C.

The comparison in terms of operation performances could be studied in Figures 5 (coefficient of performance), Figure 6 (exergetic efficiency), Figure 7 (inlet volume flow rate) and Figure 8 (specific refrigerating power). As it is known, R717 presents the highest refrigerating power, for intake volume flow rate close to the values of the other studied refrigerants.

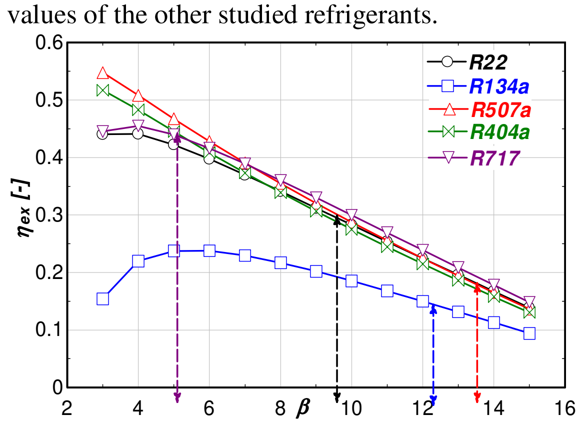

Fig. 6. Refrigerant indirect effect on the maximum attainable exergetic efficiency

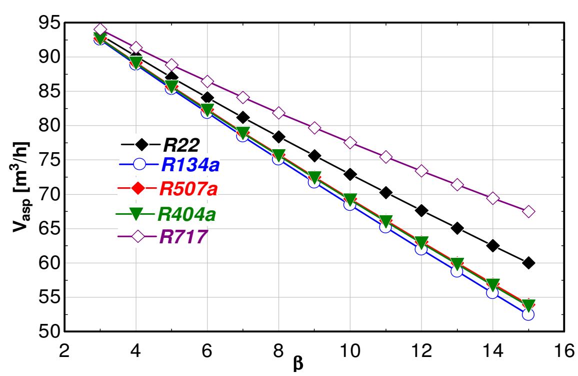

Fig. 7. Refrigerant volume rate at compressor inlet

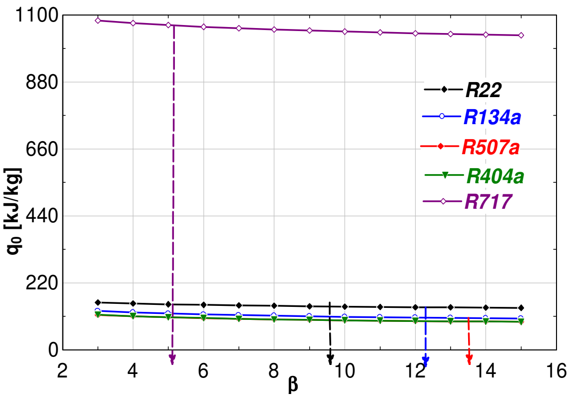

Fig. 8. Refrigerant indirect effect on the maximum attainable specific refrigerating load

In Figures 9-11 one could notice the effect of superheating on the maximum allowed compression ratio and in terms of coefficient of performance. As one might expect, a superheating degree (here of 10°C) increased the outlet temperature of the refrigerant reducing the maximum value of β and

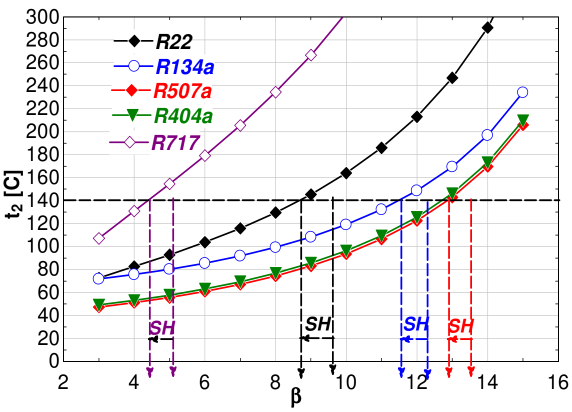

Fig. 9. Refrigerant effect on the maximum allowed compression ratio, with 10°C of superheating; comparison to the case of no superheating

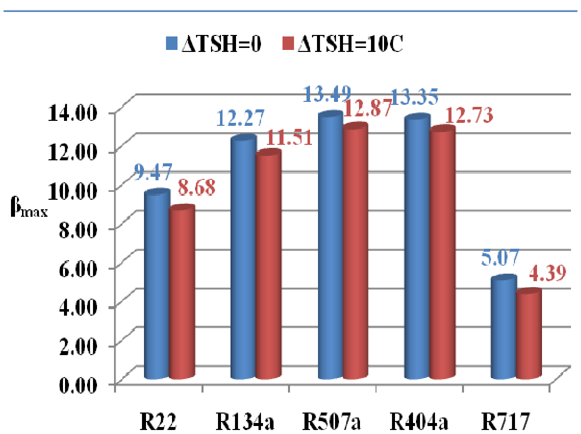

Fig. 10. Comparison between maximum allowed compression ratio, influence of superheating

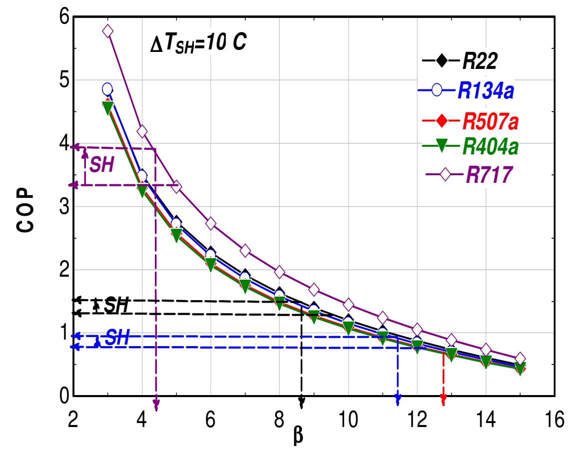

Fig. 11. Refrigerant effect on the maximum attainable COP, with 10°C of superheating; comparison to the case of no superheating

increasing the coefficient of performance. The most important effect on *COP* of superheating is presented by R717.

Regarding irreversibilities during VCRS operation, in Figure 12 one may see the repartition of exergy destruction rates on each component of the system (C-condenser, Cp-compressor, TV-throttling valve, V-evaporator) for the five refrigerants. One can notice that ammonia presents the highest values. It is also interesting to notice that the highest exergy destruction rate is in the compressor for R134a, R717 and almost for R22,

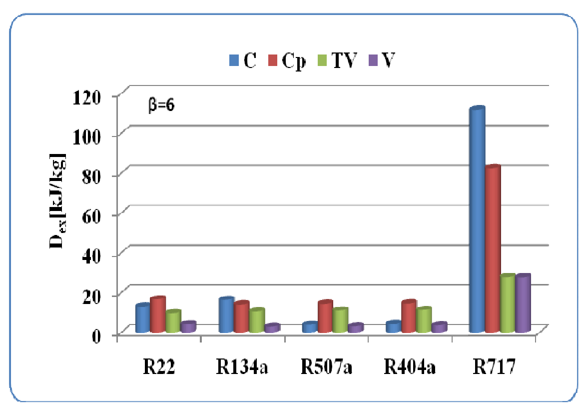

Fig. 12. Refrigerant effect on the exergy destruction rates in the condenser (C), compressor (Cp), throttling valve (TV) and evaporator (V), no superheating and β=6

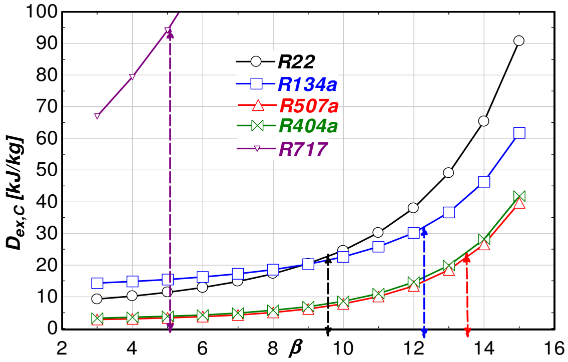

Fig. 13. Refrigerant effect on the exergy destruction rates in the condenser, no superheating

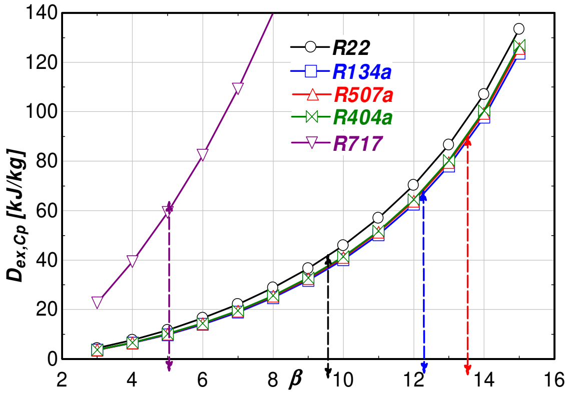

Fig. 14. Refrigerant effect on the exergy destruction rates in the compressor, no superheating

while the two azeotropic blends leads to a higher exergy destruction rate in the compressor, although in the studied numerical example, the temperature variation during phase change process is about 0.4°C. The lower values of the exergy destruction rates in the condenser are due to the lower values of the condensation temperature and thus the temperature gap between the refrigerant and the environnement during this phase change process. This information could be used when optimizing the system, in temrs of the component that should be focused, depending thus on the refrigerant.

Figures 13 and 14 show the variation of exergy destruction rates in condenser (Figure 13) and compressor (Figure 14) with the compression ratio and working refrigerant. Higher the compression ratio is, higher the destruction rates are. The differences when comparing the refrigerants are mostly due to condenser destruction rates. Ammonia has a special behavior, as the exergy destruction rate is very sensitive to β. Studying the behavior of the vapor compression refrigerating system in different conditions could bring important information about the components that should be optimized and about the most suitable refrigerant (among available alternatives).

5. CONCLUSIONS

A comparative analysis of the refrigerant impact on the operation and performances of a one stage vapor compression refrigeration system was presented. The effects of compression ratio and superheating were studied on the system operation and performances. Based on the exergy analysis, exergy destruction rates were estimated for each component of the system in a comparative manner for five refrigerants (R22, R134a, R717, R507a, R404a) proving a different behavior and thus bringing important information about system optimization when working with a specific refrigerant.

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