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Exergetic Comparison between Three Configurations of Two Stage Vapor Compression Refrigeration Systems

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***Abstract*—This study reports a comparison from an exergetic point of view between three configurations of vapor compression industrial refrigeration systems operating with R134a as working fluid. The performances of the different cycles are analyzed as function of several operating parameters such as condensing temperature and inter stage pressure. In addition, the contributions of component exergy destruction to the total exergy destruction are obtained for each system. The results are estimated to be used in the selection of the most advantageous configuration from an exergetic view point.

***Keywords*—Vapor compression, exergy, destruction, efficiency, R134a.

I. Introduction

ALTHOUGH it has good energy efficiency and can operates under wide temperature ranges, vapor compression refrigeration machine suffers from having huge electricity consumption especially in hot climate. Then the two stage vapor compression refrigeration systems were considered as a suitable solution for low and medium temperature applications like food storage and supermarket usages. In fact, the use of two compressors operating simultaneously leads to pressure ratio decrease. Consequently, the compressor electrical power input decreases significantly.

In the available literature, there is a large body of work that deals with the optimization of these systems according to energetic approach in order to improve their performances. However, most of them, are interested by having good performance (i.e. high COP that can reach 7 in some air cooled applications) without considering energy losses generated in such thermal processes that affect the quality of energy transfer [1].

The exergy analysis constitutes a more suitable approach for qualitative analysis of energy systems and it acquires a growing trend in last decades [2]-[4].

Several investigations are conducted on the exergy optimization of vapor compression cycles.

Exergy analysis was carried out on a two evaporator vapor compression refrigeration system by [5] using R1234yf, R1234ze and R134a as refrigerants. The effects of evaporator and condenser temperatures on the exergy destruction and exergy efficiency of the system were investigated. They found that the maximum exergy destruction occurs in the compressor. The best exergy efficiencies are obtained with R1234ze and R134a.

An ejector refrigeration system using R245fa as refrigerant was studied by [6]. More than the half of the total exergy destruction occurred in the ejector followed by the generator and the condenser. In this context, exergy destruction analyses were conducted numerically by [7] for vapor compression refrigeration cycles using R22, R134a, R410A and R717. In their study, optimal values of subcooling ranging from 4°C to 6°C are obtained by minimizing the total exergy destruction of the system. In addition, they found that the exergy efficiency of the latest system is strongly affected by the change of condensation and evaporation temperatures.

An exergetic analysis examined by [8] provides useful information for a two stage refrigeration machine based on the Voorhees's compression process (a process with a combination of a compression process initially at constant total volume and then near isentropic conditions). This study identifies the maximum exergy destruction rate is engendered in condenser. Hence, design improvement should be focused in this component.

Exergy analysis of two-stage vapor compression refrigeration cycle has been carried out by [9] in order to evaluate optimum inter stage pressure leading to the maximum COP and exergy efficiency for HCFC 22, R410A and R717. However, the authors observed that the condenser engenders an exergy destruction rate higher than other cycle components.

Ahamed et al. [10] reviewed on the effects of evaporating temperature, condensing temperature, subcooling and compressor pressure on the performances of vapor compression refrigeration systems from an exergetic consideration. Obtained results show that the major part of exergy losses is occurred in the compressor.

Fazelpour and Morosuk [11] have suggested that the exergy destruction within the expansion valve is the biggest contributor to the total exergy destruction in transcritical CO_2 refrigeration machine.

In this paper, an exergetic analysis is conducted on three configurations of two stage vapor compression systems with flash chamber. The operating mode of the cycles is presented, the exergetic balances are established for the different compounds. The exergy destruction rates are defined.

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II. Cycles' Descriptions

Figs. 1-3 show the schematic charts and p-H diagrams of the selected three system configurations.

The main components of the considered cycles are: A forced air cooled condenser Cd, two compressors CompHP and CompBP, a flash chamber and two air cooled evaporators EV1 and EV2 operating at two different temperatures. These refrigeration systems are designed to equip cold rooms used to store date fruits at 0°C, and poultry at -18°C.

Elements of regulation and control are added to the systems in the purpose to insure suitable operating conditions and to manage the energy consumption according to required refrigeration capacities. Their roles are detailed as follows: At the outlet of the compressor, the superheated refrigerant passes through the oil separator (OS) where the oil droplets are separated from the refrigerant to return to the compressor inlet. The refrigerant passes through the condenser to leave in liquid state. The Head pressure control regulator HPC keeps the condensing pressure at suitable value even if significant decrease in the air ambient temperature (for winter season). That permits to provide the required refrigerant mass flow rate through the thermostatic expansion valve TXV. The liquid refrigerant passes thereafter through the filter drier LFD where it is purified from impurities and moisture. At the output of heat exchanger HEX, the liquid passes through a solenoid valve (SV) to feed the TXV. This device permits to control the mass flow rate through the evaporator according to refrigeration load and adjust the superheat via a bulb fixed at the evaporator outlet tube. An evaporator pressure regulator EPR is installed after the evaporator EV1 to decrease its pressure until the low pressure at EV2. For more security, a check valve is incorporated to prevent the fluid return into EV2. The three cycles are designed on the same principle. The only difference between them is the placement of the flash chamber. In fact, for the first configuration TCFF the flash chamber is placed upstream of the first evaporator while for the TCSE system the flash chamber is located before the second evaporator EV2. For the third refrigeration system TCTE the flash chamber is placed after the condenser to feed the two evaporators.

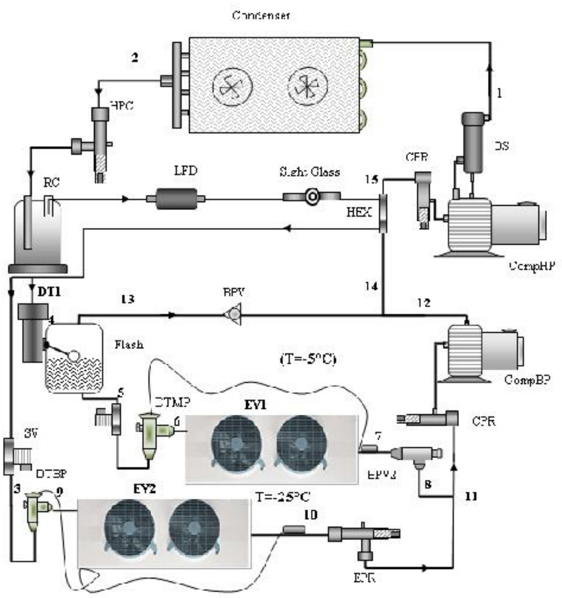
R134a is selected as refrigerant since it has favorable properties for vapor compression refrigeration cycles, low ODP, and high performance according to [5]; however, its high GWP could lead to usage boundaries in the future.

The following assumptions are considered for the analytic study:

- The system reaches a steady state and pressure drop in pipes is neglected.
- The flow across the valve is isenthalpic.
- The refrigerant at the condenser outlet is a subcooled liquid. A difference temperature design between the condensation and the ambient air is assumed to be approximately 10°K.
- Superheat is considered at the evaporator's outlets.
- A thermodynamic equilibrium is considered in the flash.

The considered operating and design parameters for the different refrigeration systems are indicated as:

- The pressure drops within the evaporators and the condenser are assumed to be 0.15 and 0.2, respectively [12].
- The pressure drop within the heat exchanger is 0.1 bars.
- The same degree of subcooling after the condenser and HEX is taken equal to 2°K.
- Degree of superheating after EV1 and EV2 are 7.5°K and 7°K, respectively [13].
- Effectiveness of liquid vapor heat exchanger is taken equal to 0.7.
- The pinch point of the heat exchanger is taken equal to 3°C.
- Isentropic compressors efficiencies are taken constant and equal to 85%.
- Electrical and mechanical compressors efficiencies are 97% and 90% respectively.
- The difference temperature design between evaporators and air cold space is 7°K.



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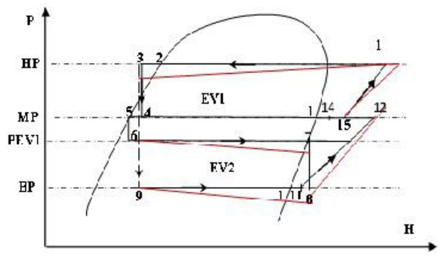

(b)

Fig. 1 (a) Flow chart of TCFE system (b) p-H diagram of TCFE system. CPR: Compressor pressure regulator–EPR: Evaporator pressure regulator-LFD: filter drier-HPC: head pressure Control-RC: receiver-SV: solenoid valve

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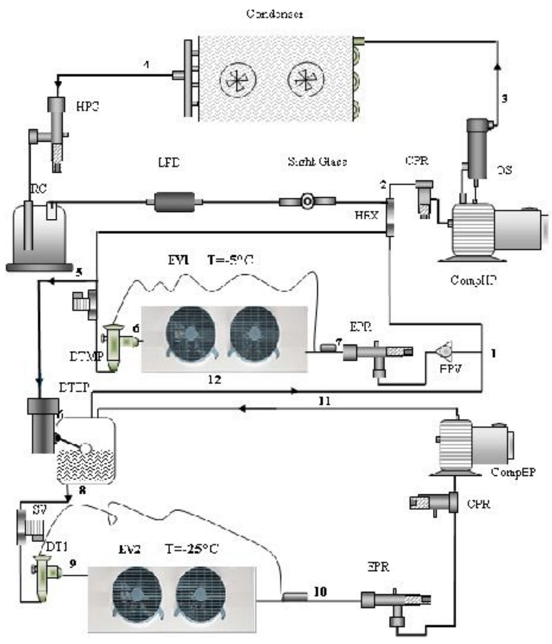

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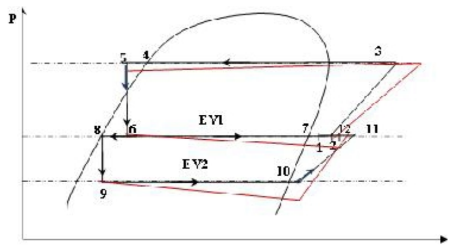


(b)

Fig. 2 (a) Flow chart of TCSE system (b) p-H diagram of TCSE system

III. Equations System

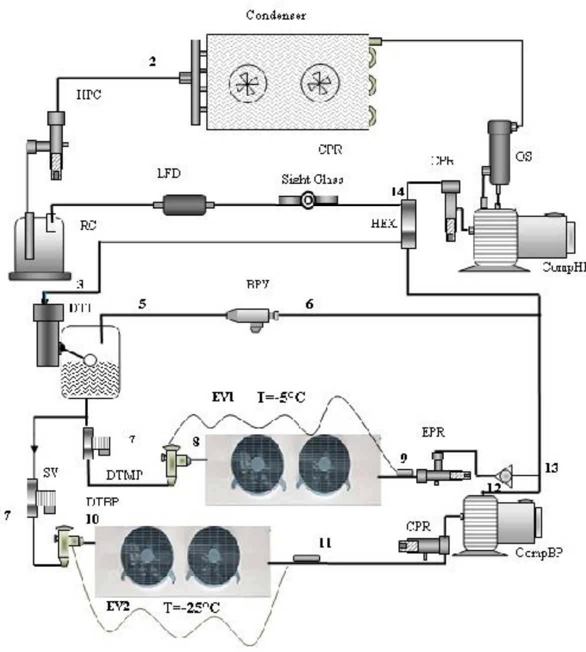
The exergy destruction within each component of the cycles can be arranged as [7]:

$$\dot{E}x_{dest} = \sum_{out}(1-\frac{T_0}{T_j})\times Q_j + \dot{W}_{out} + \sum_{in}\dot{m}\times\psi - \sum_{out}\dot{m}\times\psi = T_0\times\dot{S}_{gen}. \quad (1)$$

where Q_j is the heat transfer rate through the element at T_j temperature (kW), $\dot{W}_{out}$ is the work rate (kW), ψ is the physical exergy of the j element (kJ/kg), T_0 is the environment temperature, $\dot{m}$ is the mass flow rate (kg/s) and $\dot{S}_{gen}$ is the entropy generation (kJ/°C).

The following assumptions are considered to calculate the exergy rate at the different points:

- It is assumed that only physical exergies for refrigerant and air flows are considered.
- Kinetic and potential exergies of streams are usually neglected.
- Chemical exergy is also neglected since no chemical reactions occur within components of the analyzed cycles
- The pressure drops at airside are neglected [14].
- The reference environment state for the systems is T_0=25°C and P_0=1 bar.



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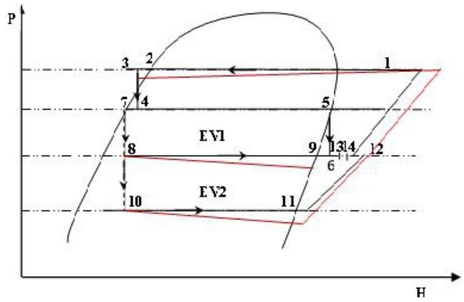

(b)

Fig. 3 (a) Flow chart of TCTE system (b) p-H diagram of TCTE system

The total exergy rate associated with the j[th] stream is written as:

$$E_j = \dot{m}_j \times \psi_j \quad (2)$$

where;

$$\psi_j = (h_j - h_0) - T_0(s_j - s_0) \quad (3)$$

h: specific enthalpy (kJ/kg) and *s*: specific entropy (kJ/kg. °C).

A. Simulation and Exergetic Analysis

Equations (4)-(15) are related to energy transferred, cooling capacities, mass flow rates, enthalpies and exergy destruction rates for each component are expressed as follows. The subscribes i and o, mean respectively, inlet and outlet of the considered component.

The condenser:

$$\dot{Q}_{Cd} = \dot{m}_T \times (h_{Cd,i} - h_{Cd,o}) \tag{4}$$

$$Ex_{dest,Cd} = \dot{m}_T \times [T_0(s_{Cd,o} - s_{Cd,i}) + \frac{T_0}{T_{Cd}}(h_{Cd,i} - h_{Cd,o})] + \dot{W}_{fanCd}\,. \tag{5}$$

The two compressors

$$\dot{W}_{Comp} = \dot{m}_{Comp,i}(h_{Comp,o} - h_{Comp,i}) \tag{6}$$

$$Ex_{dest,Comp} = \dot{m}_{Comp,i}[(h_{Comp,i} - T_0 s_{Comp,i}) - (h_{Comp,i} - T_0 s_{Comp,o})] + \dot{W}_{Comp,el}\,. \tag{7}$$

The evaporators

$$\dot{m}_{EVj,i} = \frac{\dot{Q}_{EVj}}{h_{EVj,o} - h_{EVj,i}} \tag{8}$$

$$Ex_{dest,EVj} = \dot{m}_{EVj,i}[T_0(s_{EVj,o} - s_{EVj,i}) - (h_{EVj,o} - h_{EVj,i})\frac{T_0}{T_{EVj}}] + \dot{W}_{fanEVj}\,. \tag{9}$$

The expansion valves:

$$h_{Ex,o} = h_{Ex,i} \tag{10}$$

$$Ex_{dest,TXv} = \dot{m}_{TXV,i} \times T_0(s_{TXV,o} - s_{TXV,i}) \tag{11}$$

The flash chamber:

$$\dot{m}_{flash,v} = \dot{m}_l \frac{h_{flash,l} - h_{flash,i}}{h_{flash,i} - h_{flash,v}} \tag{12}$$

$$Ex_{dest,flash} = \sum_j \dot{m}_{flash,i}(h_{flash,i} - T_0 s_{flash,i}) - \sum_j \dot{m}_{flash,o}(h_{flash,o} - T_0 s_{flash,o}). \tag{13}$$

The heat exchanger:

$$\dot{Q}_{HEX} = \dot{m}_{c,i}(h_{c,o} - h_{c,i}) = \dot{m}_{h,i}(h_{h,i} - h_{h,o}) \tag{14}$$

$$Ex_{dest,HEX} = \sum_j \dot{m}_{HEX,i} \times (h_{HEX,i} - T_0 s_{HEX,i}) - \sum_j \dot{m}_{HEX,o} \times (h_{HEX,o} - T_0 s_{HEX,o}). \tag{15}$$

where the subscripts c and h reflect the cold and hot fluids respectively.

The total exergy destruction is the sum of exergy destruction of the different components.

The exergy efficiency is given by:

$$\eta_{ex} = \frac{\left|\dot{Q}_{EV1}(1 - \frac{T_0}{T_{EV1}})\right| + \left|\dot{Q}_{EV2}(1 - \frac{T_0}{T_{EV2}})\right|}{\dot{W}_{input}} \tag{16}$$

where $\dot{W}_{input}$ is the total power input to the system defined by:

$$\dot{W}_{input} = \dot{W}_{CompHP,el} + \dot{W}_{CompLP,el} + \dot{W}_{fanCd} + \dot{W}_{fanEV1} + \dot{W}_{fanEV2}\,. \tag{17}$$

$\dot{W}_{fanCd}$, $\dot{W}_{fanEV1}$, $\dot{W}_{fanEV2}$ are the required electrical powers for compressors, condenser fans and evaporators fans respectively, defined according to manufacturer data.

$$\dot{W}_{comp,el} = \frac{\dot{W}_{comp}}{\eta_{el} \times \eta_{mec}} \tag{18}$$

III. Results and Discussion

A calculation code is established using the software EES to perform exergy analysis. In the following, we present the obtained results related to the effect of condensing temperature and the inter stage pressure on the exergy efficiency and the exergy destruction rates of the selected systems.

A. Effect of Condensing Temperature on the Performance of the Cycles

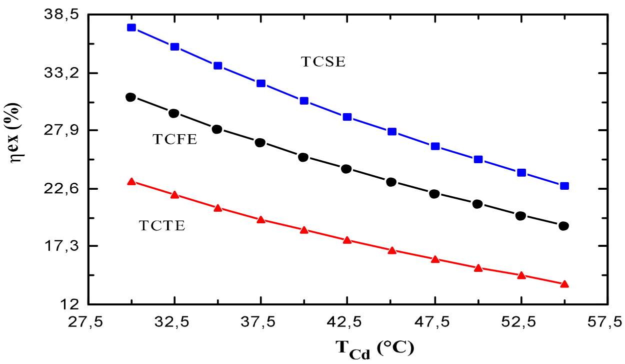

Fig. 4 Exergy efficiencies versus condensing temperature

Exergy efficiencies of the three systems as function of condensing temperature are depicted in Fig. 4. As can be seen, exergy efficiencies decrease with the increase of condenser temperature. This tendency can be explained by the fact that the average thermal gradient between the condenser and the

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environment increases with T_{Cd}. Thus, the condensing temperature must be low to obtain better exergetic efficiency. However, taking into consideration the differential pressure required for good running conditions of the thermostatic expansion valves, the condensing temperature has to be maintained at suitable value.

Among the considered configurations, TCSE cycle shows the best exergetic performance followed by TCFE and TCTE systems.

The effects of condensing temperature on the exergy destruction rate in each component are shown in Fig. 5 for the three cycles.

The exergy destruction rate through the compressors is significantly affected by the condensing temperature for the three systems. In addition, for the TCFE cycle, the DTBP exergy destruction rate is affected by T_{Cd}.

The maximum exergy destruction rates are engendered by the Condensers (about 17 kW) and the high-pressure compressors (average of 15 kW) followed by the evaporators EV2 (about 11 kW).

For TCFE system, the minimum exergy destruction rate is obtained for the DTMP followed by the HEX, DT1 and the flash chamber. While for the TCSE system, the DT1 presents the lower entropy generation rate. For the TCTE configuration, DTMP and HEX have the minimum contribution in exergy destruction. One can see that the exergy destruction within the evaporator EV2 is sensibly higher than that engendered by EV1. This can be explained by the fact that the temperature gradient between the environment temperature T_0 and T_{EV2} is more important than the one taken for the evaporator EV1.

Fig. 6 showed the total exergy destruction within the whole system versus the condensing temperature.

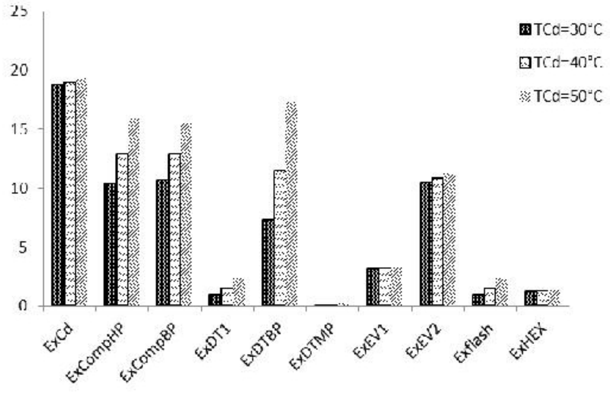
For the three considered systems, the total exergy destruction increases with T_{Cd}.

For the explored condensing temperature range, the TCFE system presents the important total exergy destruction followed by TCSE and TCTE and as shown in Fig. 6. This may constitute a criterion for refrigeration system design.

B. Effect of Inter Stage Pressure on the Performance of the Cycles

It is well known that the value of inter stage pressure MP is one of the most important variables for the optimization of two stage refrigeration machines. This operating parameter is usually calculated using (19) [15]:

$$MP = \sqrt{HP \times BP} \tag{19}$$



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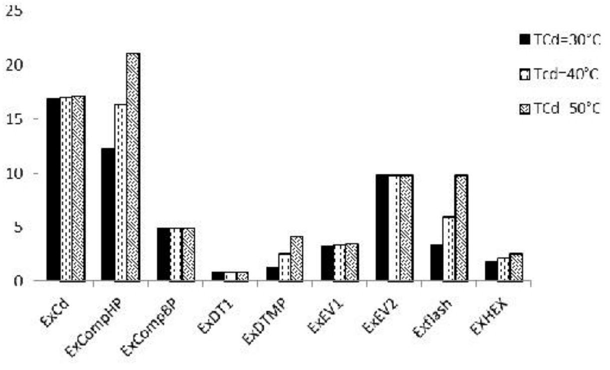

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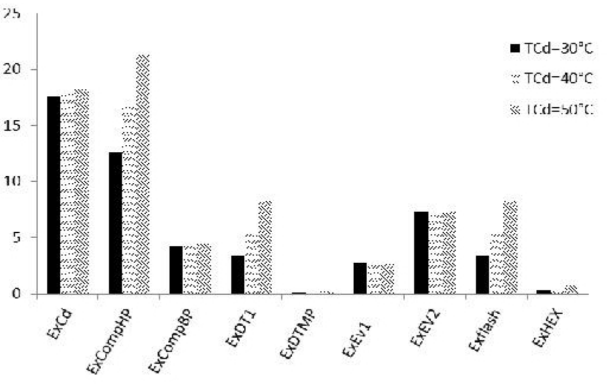

(c)

Fig. 5 (a) Exergy destruction within components as function of condensing temperature for TCFE system (b) Exergy destruction within components as function of condensing temperature for TCSE system (c) Exergy destruction within components as function of condensing temperature for TCTE system

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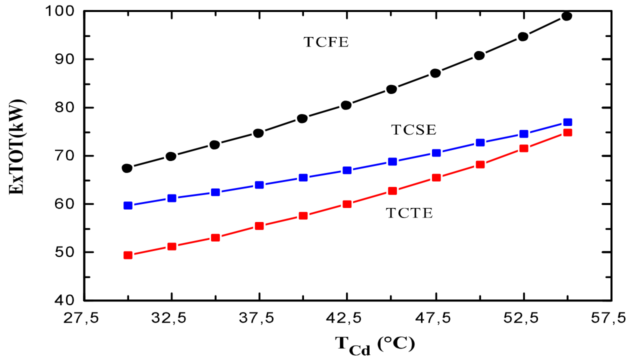

Fig. 6 Total exergy destructions versus condensing temperature

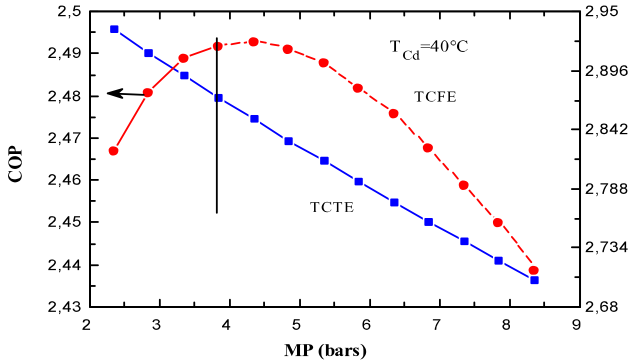

Fig. 7 COP versus inter stage pressure at T_{Cd}=40°C

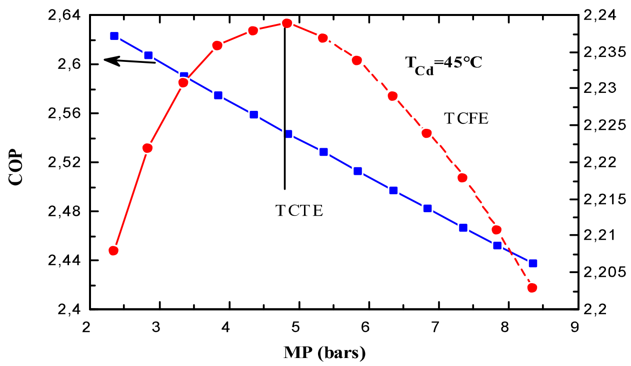

Fig. 8 COP versus inter stage pressure at T_{Cd}=45°C

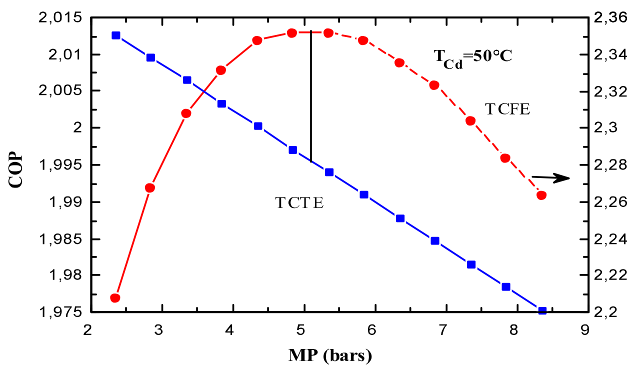

Fig. 9 COP versus inter stage pressure at T_{Cd}=50°C

From an energetic point of view, the optimal inter stage pressure is defined for maximum values of COP as illustrated in Figs. 7-9. For TCFE system, the COP increases with MP to reach a maximum value and then decreases sensibly higher values of MP. This shape is obtained for the considered condenser temperatures (40, 45, and 50°C). The optimum values of MP are obtained for maximum COP as:

- COP_{max} = 2.492 for MP = 3.84 bars and T_{Cd} = 40 °C
- COP_{max} = 2.239 for MP = 4.34 bars and T_{Cd} = 45 °C
- COP_{max} = 2.013 for MP = 5.34 bars and T_{Cd} = 50 °C

However, for TCTE system, linear decrease of COP versus MP is found. Therefore, MP must be taken as minimum as possible to improve the system performance.

TCSE cycle cannot operate with variable MP because it is driven by the EV1 pressure which is defined by the temperature T_{EV1}.

Figs. 10-12 illustrate the variation of exergy efficiencies according to inter stage pressure for TCTE and TCFE systems and different condensing temperatures.

An optimum inter stage pressure value is obtained for the TCFE system where the exergy efficiency reaches a maximum value of about 25.65%. While the exergy efficiency of TCTE cycle decreases linearly with MP.

For the considered condensing temperatures, the optimum values of MP leading to the maximum exergy efficiency are obtained as:

- $\eta_{ex\ max}$ = 25.64% for MP = 4.34 bars and T_{Cd} = 40°C.
- $\eta_{ex\ max}$ = 23.35% for MP = 4.34 bars and T_{Cd} = 45°C.
- $\eta_{ex\ max}$ = 21.28% for MP = 4.84 bars and T_{Cd} = 50°C.

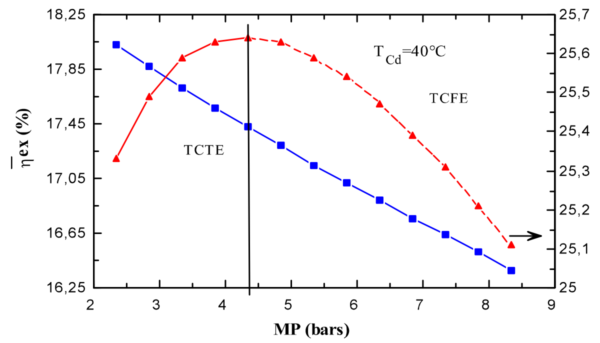

Fig. 10 Exergy efficiencies versus inter stage pressure at T_{Cd}=40°C

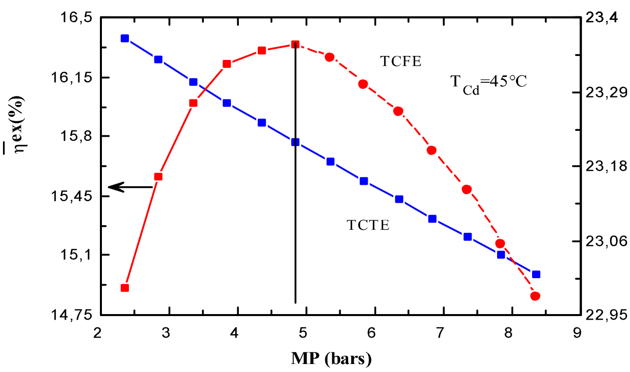

Fig. 11 Exergy efficiencies versus inter stage pressure at T_{Cd}=45°C

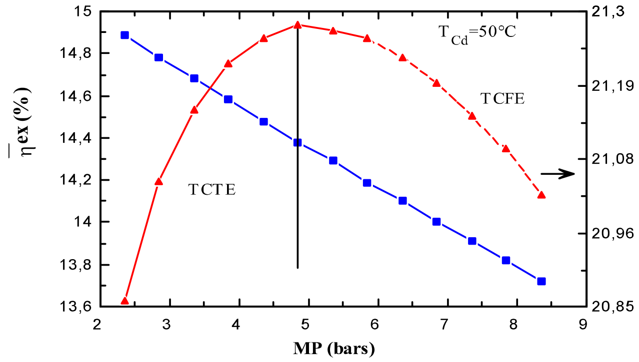

Fig. 12 Exergy efficiencies versus inter stage pressure at T_{Cd}=50°C

IV. Conclusion

An exergy analysis of three industrial configurations of R134a two stage vapor compression refrigeration cycles is presented.

A parametric study is performed to analyze the effects of inter stage pressure and condensing temperature on the energy and exergy performances of the considered systems. The main results obtained can be summarized in the following concluding remarks:

- Condensing temperature affects sensibly the exergy efficiency of all systems and the exergy destruction in most components.
- Condensers and high-pressure compressor CompHP present the maximums exergy destruction rates for the all configurations.
- Optimum inter stage pressure depends strongly on condensing temperature. Its optimum values obtained from energetic optimization are not the same ones obtained from an exergetic approach.
- The best exergy efficiency is obtained for the TCSE system.

Thus, to enhance the system exergetic efficiencies, the design of condensers and compressors has to be carefully improved.

References

[1] R. Yumrutas, M. Kunduz, M. Kanoglu, “Exergy analysis of vapor compression refrigeration systems,” Exergy, vol. 2, 2002, pp. 266–272.
[2] D. Cai, G. He, Q.Tian, W.Tang, “Exergy analysis of a novel air-cooled non-adiabatic absorption refrigeration cycle with NH3–NaSCN and NH3–LiNO3 refrigerant solutions,” Energy Conversion and Management, vol. 88, 2014, pp. 66–78.
[3] G. Yan, J. Chen, J. Yu, “Energy and exergy analysis of a new ejector enhanced auto-cascade refrigeration cycle,” Energy Conversion and Management, vol. 105, 2015, pp. 509–517.
[4] Y. Dai, J. Wang, L. Gao, “Exergy analysis, parametric analysis and optimization for a novel combined power and ejector refrigeration cycle, ” Applied Thermal Engineering, vol. 29, 2009, pp. 1983–1990.
[5] A. Yataganbaba, A. Kilicarslan, I. Kurtbas, “Exergy analysis of R1234yf and R1234ze as R134a replacements in a two evaporator vapour compression refrigeration system,” Refrigeration, 2015, doi:10.1016/j.ijrefrig.2015.08.010
[6] J. Chen, H. Havtun, B. Palm, “Conventional and advanced exergy analysis of an ejector refrigeration system,” Applied Energy, vol. 144 2015, pp. 139–151.
[7] M. H. Yang, R. H. Yeh, “Performance and exergy destruction analyses of optimal subcooling for vapor-compression refrigeration systems,” International Journal of Heat and Mass Transfer, vol. 87, 2015, pp. 1–10.
[8] T. Morosuk, G. Tsatsaronis, C. Zhang, “Conventional thermodynamic and advanced exergetic analysis of a refrigeration machine using a Voorhees’ compression process,” Energy Conversion and Management, vol. 60, 2012, pp. 143–151.
[9] A. Arora, S. C Kaushik, “Energy and exergy analyses of a two stage vapor compression refrigeration system,” International journal of energy research, vol. 34, 2010, pp. 907-923.
[10] J. U. Ahamed, R. Saidur, H. H. Masjuki, “Review on exergy analysis of vapor compression refrigeration system, ” Renewable and Sustainable Energy Reviews, vol. 15, 2011, pp. 1593–1600.
[11] F. Fazelpour, T. Morosuk, “Exergoeconomic analysis of carbon dioxide transcritical refrigeration machines,” International journal of refrigeration, vol. 38, 2014, pp. 128-139.
[12] A. Piacentino, F. Cardona, S. Oriented, “Thermoeconomic analysis of energy systems. Part I: Looking for a non-postulated cost accounting for the dissipative devices of a vapour compression chiller. Is it feasible?,” Applied Energy, vol. 87, 2010, pp. 943–956.
[13] M. W. Halfaoui, T. Khir, A. Ben Brahim, “Thermo analysis and optimization design of industrial vapor compression refrigeration systems,” Journal of Refrigeration, Air Conditioning, Heating and Ventilation, vol. 2, 2015, pp. 30–54p.
[14] R. K. Jassim, T. Khir, “Exergoeconomic optimization of the geometry of continuous fins on array of tubes of a refrigeration air cooled condenser,” Exergy, vol.2, 2005, pp. 146-170.
[15] K. Ahmet, O. Kizilkan, A. K. Yakut, “Performance and exergetic analysis of vapor compression refrigeration system with an internal heat exchanger using a hydrocarbon, isobutane (R600a), ”Energy Research vol.32, 2008, pp. 824–836, doi: 10.1002/er.1396.