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CO_2 AUTOMOTIVE CLIMATIZATION

A. HILAL[1] E. E. VASILESCU[1] A. DOBROVICESCU[1] GH. POPESCU[1]

Abstract: *The use of ecological fluids in the thermal and refrigeration systems represents a priority for the modern research. Thus, a new direction was developed regarding the use of CO_2 in the air conditioning systems of the motor vehicles, especially because these systems represent the most important source of pollutant losses in the atmosphere. The paper reveals the benefits of CO_2 as refrigerant for automotive climatization and presents the energy and exergy analysis of the transcritical cycle with CO_2.*

Key words: *conditioner, vehicle, ecology, natural refrigerant, CO_2.*

1. Introduction

In our time, the directions of research and optimization of the air conditioning systems and equipments are related to the environment protection, energy conservation and a compact design of the installation components. These are important reasons that involved an important scientific effort for the identification of more performant automotive air conditioning systems with less pollutant emissions.

The most used refrigerant in the automotive climatization is R134a but it seems that CO_2 as natural ecologic fluid is the refrigerant of the close future [1], [7-9], [10], [11].

The European Parliament and the countries members of the European Union will impose by law the exclusion of the HFC R134a because it is one thousand times more harmful for the environment than CO_2. The law will be applied starting with 2011 on all the new models of cars.

This action has as purpose the diminution of the emission of gas with green house impact in Europe.

On February 2006, at a conference on the cars air conditioning systems that took place in Saalfelden – Austria, the european cars manufactures and the climatization systems components suppliers proved that natural automotive cooling systems are ready to enter on the european market. The vehicle air conditioner suppliers (as BEHR, OBRIST, VALEO and VISTEON) presented cars with natural CO_2 as working fluid in the conditioners. Also, at this conference, the experts estimated that 2 millions of cars with natural cooling systems will be sold until 2011.

2. The Properties of CO2

The thermodynamics properties of some refrigerants, including R744 (CO2) are shown and compared in the table 1 [6].

One notices that R744 (CO_2) is a natural fluid non-flammable without ODP and with negligible GWP. The reduced pressure of CO_2 is higher than for other refrigerants, imposing a compact and resistant structure of the conditioner but as advantage its volumetric refrigeration power is also 3…10 times higher. A

[1] Dept. of Thermodynamics, Politehnica University of Bucharest.

Refrigerant properties Table 1

	R12	R22	R134a	R407C[a]	R410A[b]	R717	R290	R744
ODP/GWP[c]	1/8500	0.05/1700	0/1300	0/1600	0/1900	0/0	0/3	0/1
Flammability/ toxicity	NO/NO	NO/NO	NO/NO	NO/NO	NO/NO	YES/YES	YES/NO	NO/NO
Molar mass (kg/kmol)	120.9	86.5	102.0	86.2	72.6	17.0	44.1	44.0
Normal boiling temperature[d] (^oC)	-29.8	-40.8	-26.2	-43.8	-52.6	-33.3	-42.1	-78.4
Critical pressure (bar)	41.1	49.7	40.7	46.4	47.9	114.2	42.5	73.8
Critical temperature (^oC)	112.0	96.0	101.1	86.1	70.2	133.0	96.7	31.1
Reduced pressure[e]	0.07	0.10	0.07	0.11	0.16	0.04	0.11	0.47
Reduced temperature[f]	0.71	0.74	0.73	0.76	0.79	0.67	0.74	0.90
Volumetric refrigeration power 0^oC (kJ/m^3)	2734	4356	2868	4029	6763	4382	3907	22545
First commercial use	1931	1936	1990	1998	1998	1859	?	1869

[a] Ternary mixture R32/R125/R134a (23/25/52,%)
[b] Binary mixture R32/R125 (50/50,%)
[c] GLOBAL WARMING POTENTIAL on 100 years, according to Intergovernmental Panel on Climate Change (IPPC)
[d] according to ASRHRAE handbook 2001 fundamentals
[e] Ratio of saturation pressure at 0^oC to critical pressure
[f] Ratio of 273.15 K (0^oC) to critical temperature in Kelvin

particularity of CO_2 is given by its high critical pressure (73,8 bar), and low critical temperature ($31,1^oC$), which means that CO_2 can't transfer heat to the ambient in a condensing process. Thus, the refrigeration cycle with CO_2 has its high side above the critical point and the low side below (transcritical cycle). The high pressure of the cycle can be chosen independently of the low pressure, allowing optimal performances.

3. Components of the Climatization System with CO_2 as Working Fluid

The majority of CO_2 climatization components are constructive and functionally similar to other systems. Thus, the installation has a compressor, an evaporator, an expansion valve, an accumulator of liquid and pipes that connect these apparatus. The difference is given by the existence of one or more gas coolers instead of the classical condenser. An example of conditioner with CO_2 is given in the fig.1. This figure shows the scheme of the climatization system with CO_2 of the hybrid vehicle Toyota FCHV-4 [2].

The manufacturer of the climatization system, DENSO AUTOMOTIVE, specifies that CO_2 has a very good heating capacity that recommends it as working fluid in the heat pump systems, especially for the electrical or hybrid cars that have no other efficient heating source for the cabin. In the fig.1, the heat released by the interior gas cooler is used to warm the cabin in winter while the evaporator cools the cabin in summer time.

The hybrid car fuel (hydrogen)-electric TOYOTA FCHV (fuel cell hybrid vehicle) -4 was officially presented at the 39th Tokyo Motor Show from Makuhari Messe (Nippon Conversion Center) 22 October- 6 November 2005.

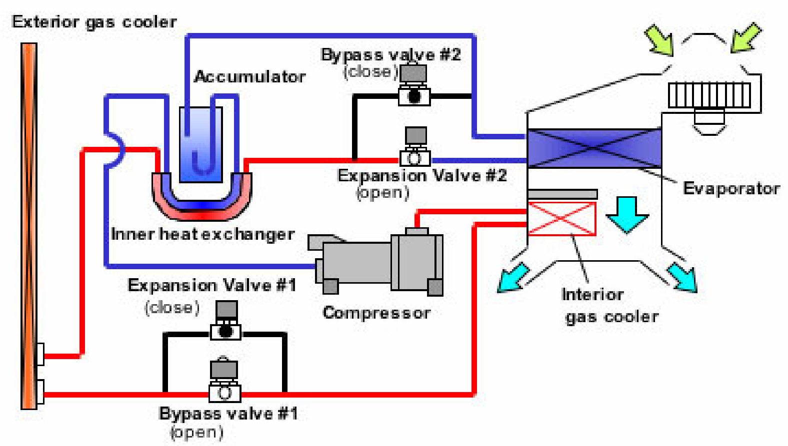

Fig.1. *The CO_2 climatization system of Toyota FCHV-4*

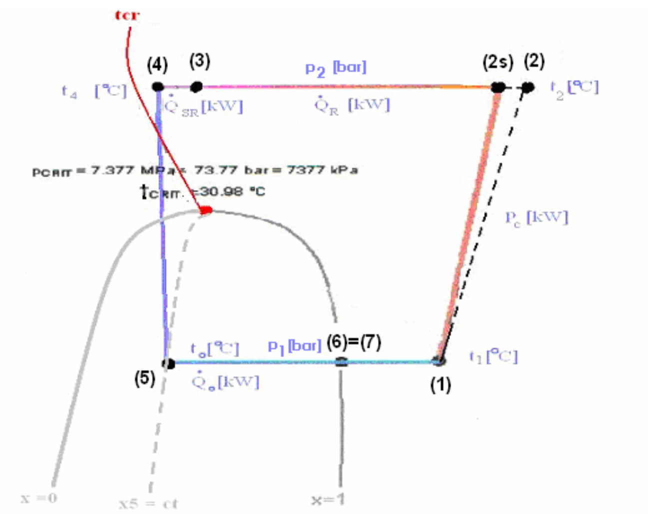

Fig.2. *Trascritical CO_2 cycle with regenerative cooling (with inner heat exchanger)*

4. Transcritical Cycle of CO_2 with Regenerative Sub-Cooling

The general transcritical refrigeration cycle (for the summer time) of CO_2 in a conditioner with inner heat exchanger is according to the fig. 2.

Processes from the cycle:

1 – 2 vapor compression in the compressor

2 – 3 isobar cooling of the vapor of CO_2 in the gas coolers

3 – 4 supplementary cooling (subcooling) of gaseous CO_2 in the inner heat exchanger;

4 – 5 isenthalpic (h=ct.) expansion of the refrigerant in the expansion valve;

5 – 6 isobar-isotherm evaporation of refrigerant (CO_2) in the evaporator

7 - 1 overheating of gaseous CO_2 in the inner heating (in the hypothesis that the accumulator does not change the state of the refrigerant $6 \equiv 7$).

5. Energy and Exergy Analysis of Transcritical Cycle of CO_2 with Regenerative Cooling (with Inner Heat Exchanger in the Scheme); Optimization

The technical literature contains thermodynamic models of analysis and optimization of the thermodynamic cycles with CO_2, as for example [3-5]. Following this direction, this paragraph contains an energy and exergy analysis of the refrigeration cycle (with inner heat exchange) of CO_2 (summer time) and a study of parametric sensitivity in the following conditions:

<u>Parameters of the central point:</u>

- refrigeration load $\dot{Q}_0 = 2$ kW
- evaporation temperature $t_o = 5$ °C
- decrease of the temperature of CO_2 in the inner heat exchanger $\Delta t_{sr} = t_3 - t_4 = 10$ grd
- outlet temperature of the gas coolers $t_3 = 50$ °C (in the summer time when the environment temperature can be higher than 25…30 °C)

<u>Optimization variable:</u>

- compressor exhaust pressure p_2 [bar]

<u>Optimizable function:</u>

- coefficient of performance *COP*
- exergy efficiency η_{ex}

The mathematical model uses the following principal equations:

- refrigeration specific power:

$$q_o = q_{5-6} = h_6 - h_5 \quad \text{[kJ/kg]} \tag{1}$$

- specific condenser load

$$|q_r| = |q_{2-3}| = h_2 - h_3 \quad \text{[kJ/kg]} \tag{2}$$

- specific inner heat exchanger load

$$\begin{aligned} |q_{sr}| &= |q_{3-4}| = h_3 - h_4 = \\ &= q_{si} = q_{7-1} = h_1 - h_7 \end{aligned} \quad \text{[kJ/kg]} \tag{3}$$

- specific mechanical work

$$|l_c| = |l_{t1-2}| = h_2 - h_1 = \frac{|l_{cs}|}{\eta_{ic}} \quad \text{[kJ/kg]} \tag{4}$$

with $|l_{cs}| = |l_{t1-2s}| = h_{2s} - h_1$ reversible adiabatic mechanic work and η_{ic} internal compression efficiency with a chosen value of 0,90.

- mass flow rate

$$\dot{m} = \frac{\dot{Q}_o}{q_o} \quad \text{[kg/s]} \tag{5}$$

- volumetric flow rate in the suction line of the compressor

$$\dot{V}_a = \dot{m} \cdot v_1 \quad \text{[m}^3\text{/s]} \tag{6}$$

- thermal load of the gas coolers

$$\dot{Q}_r = m \cdot |q_r| \quad \text{[kW]} \tag{7}$$

- thermal load of the inner heat exchanger

$$\dot{Q}_{sr} = \dot{m} \cdot |q_{sr}| = \dot{m} \cdot q_{si} \quad \text{[kW]} \tag{8}$$

- mechanical power consumed by the compressor

$$\dot{W}_c = \dot{m} \cdot |l_c| \quad \text{[kW]} \tag{9}$$

- energy balance equation

$$\dot{Q}_o + \dot{W}_c = \dot{Q}_r \tag{10}$$

- coefficient of performance

$$COP = \frac{\dot{Q}_o}{\dot{W}_c} \quad [-] \tag{11}$$

- destruction of exergy in each apparatus of the scheme

$$\dot{\Pi}_j = \sum \left(1 - \frac{T_{amb}}{T_i}\right)\dot{Q}_i - \dot{W} + \\ + \dot{\varepsilon}_{in} - \dot{\varepsilon}_{out} \tag{12}$$

with:

T_{amb} [K] reference temperature in the exergy analysis that one considered as t_{amb} $= 30^{o}\text{C}, \sum Ex(\dot{Q}_i) = \sum \left(1 - \frac{T_{amb}}{T_i}\right)\dot{Q}_i$ [kW] (13)

the exergy of the heat fluxes $\dot{Q}_i$ [kW] exchanged by the refrigerant with the external heat sources with temperature T_i [K], $\dot{W}$ [W] mechanical power produced or consumed.

$$\dot{\varepsilon}_{out} - \dot{\varepsilon}_{in} \\ = \sum_{out} \dot{m}(h - T_{amb}s) - \sum_{in} \dot{m}(h - T_{amb}s) \quad \text{.[kW]} \tag{14}$$

the variation of the exergy of the mass that enter (in) and goes out (out) of the system.

- exergy balance equation

$$Ex(\dot{Q}_o) + Ex(\dot{Q}_r) + \dot{W}_c - \sum \dot{\Pi}_j = 0 \tag{15}$$

with

$$Ex(\dot{Q}_r) = \dot{Q}_r \cdot \left(1 - \frac{T_{amb}}{T_{amb}}\right) = 0 \tag{16}$$

$$Ex(\dot{Q}_o) = \dot{Q}_o \cdot \left(1 - \frac{T_{amb}}{T_f}\right) \text{ with } T_f \text{ [K]} \tag{17}$$

medium temperature of the air cooled by the evaporator of the refrigeration system; one considered $t_f = 15^{o}$C.

Thus, with explicit destruction of exergy on each apparatus (Cp-compressor, R-gas coolers, SR- inner heat exchanger, Vp-evaporator) one results:

$$Ex(\dot{Q}_o) + Ex(\dot{Q}_r) + \dot{W}_c - \\ - \dot{\Pi}_{Cp} - \dot{\Pi}_R - \dot{\Pi}_{SR} - \dot{\Pi}_{Vp} = 0 \tag{18}$$

Exergy efficiency of the system:

$$\eta_{ex} = \frac{|Ex(\dot{Q}_o)|}{\dot{W}_c} = \\ = \frac{\dot{Q}_o \cdot \left(\frac{T_{amb}}{T_f} - 1\right)}{\dot{W}_c} = \frac{COP}{COP_C^{T_f, T_{amb}}} \tag{19}$$

with $COP_C^{T_f, T_{amb}}$ the coefficient of performance of the refrigeration cycle Carnot with extreme temperatures T_f and T_{amb}.

One notices that the exergy efficiency can also be written:

$$\eta_{ex} = \frac{P_c - \dot{\Pi}_{Cp} - \dot{\Pi}_R - \dot{\Pi}_{SR} - \dot{\Pi}_{Vp}}{P_c} = 1 - \frac{\dot{\Pi}_{Cp} + \dot{\Pi}_R + \dot{\Pi}_{SR} + \dot{\Pi}_{Vp}}{P_c} \quad (20)$$

6. Results and Conclusions

The computational made with Engineering Equation Solver leads to the results represented below.

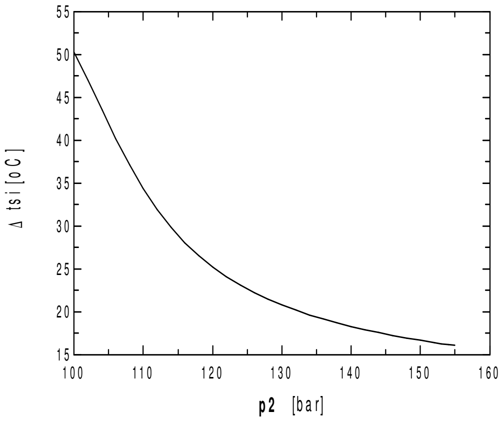

Fig.3. *Increase of the gaseous CO_2 temperature in the inner heat exchanger*

With p_2 as variable, one represented the overheating of gaseous CO_2 in the inner heat exchanger (fig.3), the mass and the volumetric flow rates (fig.4), the specific energy exchange (fig.5), the fluxes of energy (fig.6), the specific loss of exergy (fig.7), the destruction of exergy in each apparatus and in the whole system (fig.8), the coefficient of performance *COP* and the exergy efficiency (fig.9). One notices from the fig. 5 that the augmentation of the pressure p_2 is pursued by the increasing of the specific refrigeration load q_o, of the gas coolers specific load q_r and of the specific mechanical work consumed while the specific load of the inner heat exchange q_{sr} decreases.

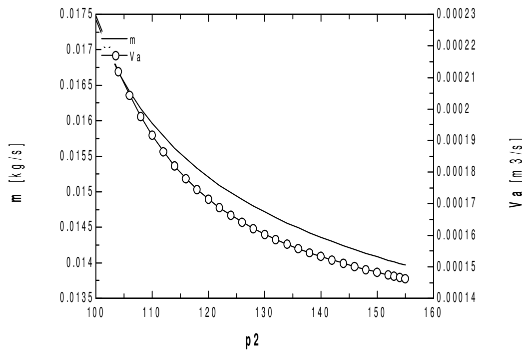

Fig. 4. *Mass and volumetric flow rate*

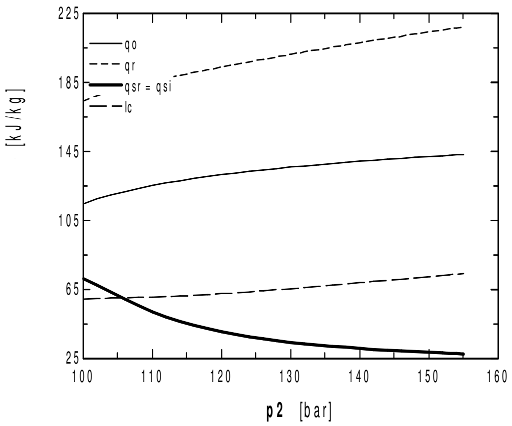

Fig.5. *Specific thermal loads, specific mechanical work*

The mass and volumetric flow rate decrease with p_2 (fig.4). The most important observation is that for an imposed refrigeration load $\dot{Q}_o$, the mechanical power consumed by the compressor has a minimum value at p_2= 119 bar (fig.6) having as consequence maximum values of both coefficient of performance COP and exergy efficiency η_{ex} (fig.9).

The gas coolers and the expansion valve work with the highest destruction of the exergy, while the compressor and the evaporator have almost the same loss. The smallest destruction of exergy appears in the inner heat exchanger (fig.7 and 8).

Fig.8 also shows a minimal value of the exergy total loss at the optimal value of the pressure p_2=119 bar.

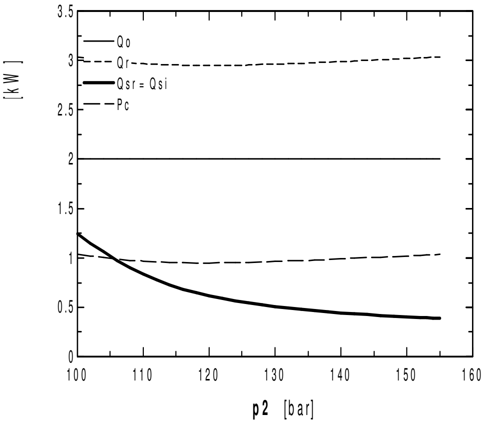

Fig.6. *Thermal loads, mechanical power consumed*

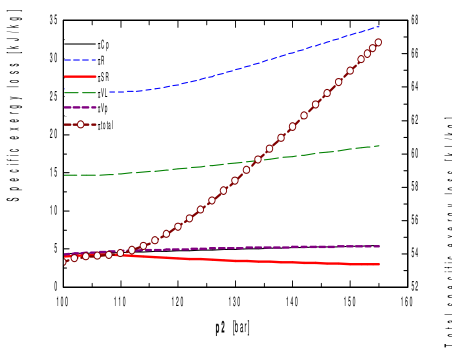

Fig.7. *Specific exergy losses*

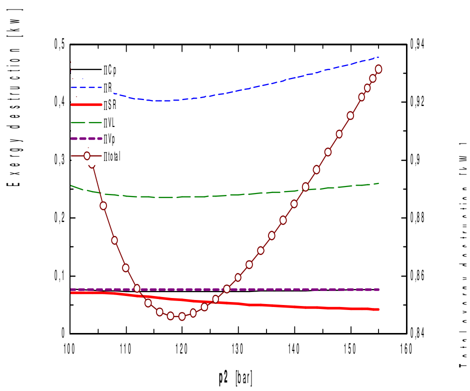

Fig.8. *Fluxes of destroyed exergy*

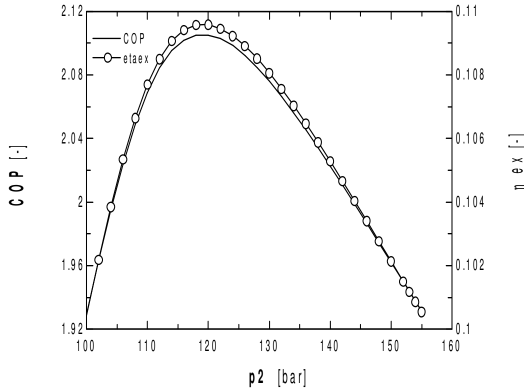

Fig.9. *Coefficient of performance and exergy efficiency*

The fig. 10, 11 and 12 contain the results of a comparative analysis between the performances of the classical, non-ameliorated cycle of CO_2 (without inner heat exchanger) and the ameliorated cycle with regenerative sub-cooling (with inner heat exchanger).

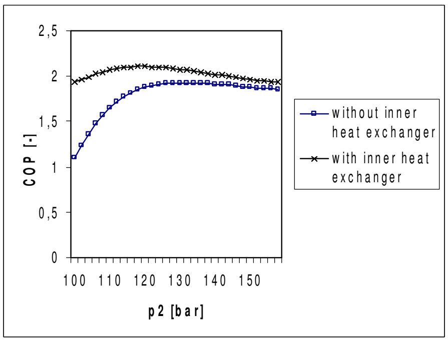

Fig.10. *Comparison of coefficients of performance*

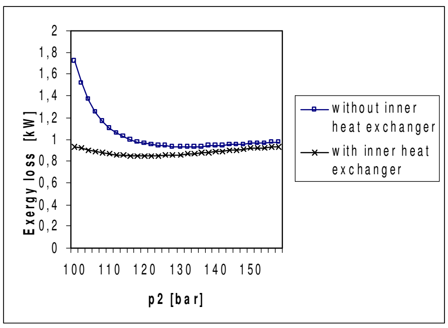

Fig.11. *Comparison of the exergy losses*

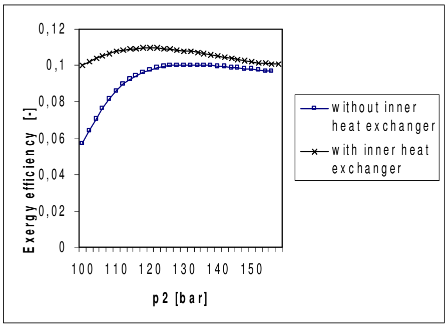

Fig.12. *Comparison of the exergy efficiency*

The fig. 10, 11, 12 show that the inner heat exchanger increases the performance of the system with 40…50 % at values of p_2 lower than the optimal value (119 bar). When the pressure p_2 increases, the coefficient of performance and the exergy efficiency tend to the same value. The both systems (with and without inner heat exchanger) admit optimal values of p_2 with maximal values of *COP* and η_{ex}. One observes that p_2 optimal is lower in the system with inner heat exchanger. This is also a positive aspect meaning that the ameliorated system works to lower pressure than the classical system.

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