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EXERGY ANALYSIS OF THE COUPLING OF TWO CO_2 HEAT PUMP CYCLES

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Keywords: Exergy destruction; Structural optimization; Internal heat exchanger.

The study is looking for the optimal configuration of an air-to-water heat pump capable of heating water in a flow-return system. Carbon dioxide is used as a working agent. In the optimization strategy, exergy analysis is considered. Using exergy analysis, the magnitude and the location of any functional or constructive malfunction can be revealed. First, a standard one-stage heat pump system is considered. Due to the high exergy destruction in the throttling valve, the efficiency of the standard system is improved by coupling it with an auxiliary one. The coupling process is undertaken in an internal subcooler-evaporator-heat exchanger. By subcooling the CO_2 before entering the throttling valve of the standard cycle, the exergy destruction associated with this process diminishes. To increase the efficiency of the globally coupled system, the heat transferred outside in the gas cooler of the auxiliary heat pump is used in the water heating process. The energetic and exergetic efficiencies of the coupled system increased by 19 % and 18.3 %, respectively, compared to the standard heat pump cycle.

1. INTRODUCTION

A lot of heat pump equipment has evolved in the last decade and is becoming more and more common in district heating processes or in industrial applications. The efficient use of energy in heating processes, and the preparation of domestic hot water, is essential for reducing energy consumption and greenhouse gases [1].

The heat pump does not induce any carbon emissions other than that produced at the power plant. Considering that much of the electricity is produced from renewable sources such as solar, wind, or hydro, the positive effect of using heat pumps on reducing greenhouse gas emissions can be quantified. Heat pumps have a super unit efficiency (over 2), returning in the heating mode both the electricity consumed and the thermal energy taken from a cold source which can be a waste heat or heat taken from the environment. Due to the obvious advantages, these pieces of equipment are intensively studied.

Bin HU et al. [2] performed an energy and exergy analysis of heat pump cycles in two and three compression stages, using as working agent R1234ze (Z). The results showed that in the case of the three-stage compression cycle, the exergy destruction in the throttling valve is significantly reduced. In the case of the three-stage compression cycle, the discharge temperature from the compressor is reduced. Alptug Y et al. [3] performed an exergetic analysis of a one-compression stage heat pump cycle using the environmentally friendly working agents R1234yf and R1234ze. The analysis showed that the R1234ze has a very comparable performance to the R134a.

CO_2 heat pumps have been studied extensively [4-13]. Carbon dioxide is a substance that is part of the natural biosphere and an environmentally friendly, safe, and economical refrigerant that can be used for cooling and heating systems. Although many heat pumps are known nowadays, due to the greenhouse gas reduction policy, researchers are focusing on environmentally friendly working fluids [1,14].

Refrigerants for heat pumps are selected based on their impact on global warming [15]. Depending on their characteristics, some promising agents used in heat pump equipment are listed in Table 1.

Table 1

Ecological working agents for heat pumps [16-18].

Working agents	Chemical formula	GWP	Critical temperature [° C]	Critical pressure [bar]
R744	CO2	1	30.98	73.77
R 1233zd (E)	C3ClF3H2	4.5	166.45	36.23
R1234ze (E)	C 3F4H2	7	109.37	36.34
R1234yf	C3F4H2	4	94.71	33.82
R152A	C2F H4	124	113.26	45.20

The present paper uses exergy analysis as a research instrument. It aims to identify the exergy destruction of each one of the components that make up the system, looking for solutions to increase its overall efficiency.

First, a transcritical standard CO_2 heat pump cycle is analyzed.

2. DESCRIPTION OF THE STANDARD SYSTEM

The present study focuses on identifying the optimal configuration of an air-to-water heat pump capable of delivering hot water at 70 °C.

The heat pump shown in Fig. 1 takes heat with low energy potential from the environment. It raises the temperature level of the heat by a mechanical compression process corresponding to the customer demand.

Due to the low critical temperature that characterizes carbon dioxide, the cycle evolves in the transcritical area.

Water is heated in a return system where the inlet and outlet water temperatures correspondingly are t_{10} = 50 °C and t_{11} = 70 °C.

To identify the system malfunctions, an exergetic analysis is used. The exergy analysis results offer a way to follow the optimization procedure.

The main thermodynamic parameters used to determine the performance characteristics of the system are presented in Table 2.

Exergy analysis is used as a research tool.

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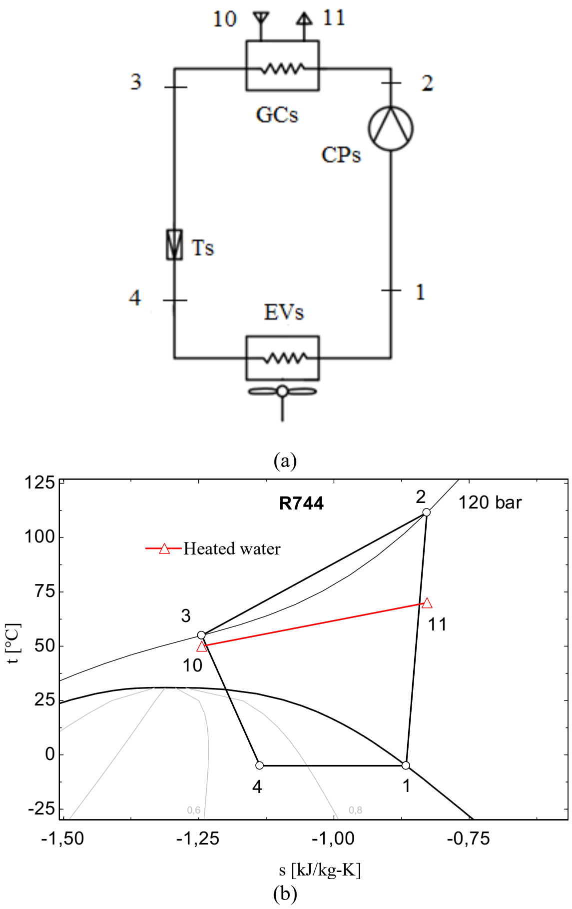


Fig. 1 Standard CO_2 heat pump cycle: (a) Schematic of the system; (b) Representation of the cycle in the T-s diagram

Table 2

Main parameters describing the standard CO_2 heat pump

Description	Notation / unit of measure	Value
Heated water temperature at the inlet	t_{10} [°C]	50
Heated water temperature at the outlet	t_{11} [°C]	70
Ambient temperature	t_0 [°C]	0
Temperature of CO_2 at the outlet of the gas-cooler	t_3 [°C]	55
Evaporation temperature	$t_4 = t_1$ [°C]	-5
Minimum temperature difference (at Pinch point) in heat exchangers	ΔT_P [°C]	5

2.1. MATHEMATICAL MODELING

2.1.1. ENERGETIC ANALYSIS

The energy analysis can be developed by setting the parameters in Table 2.

For 1 kW of heat taken from the environment in the evaporator Evs, the mass flow rate m_s of the refrigerant from the standard cycle can be calculated:

$$\dot{m}_s = \frac{\dot{Q}_{EVs}}{h_1 - h_4} \tag{1}$$

The mechanical work consumed in the compressor is:

$$\dot{W}_s = \dot{m}_s \cdot (h_2 - h_1) \tag{2}$$

The relationship can calculate the heat output in the gas cooler GCs:

$$\dot{Q}_{GCs} = \dot{m}_s \cdot (h_2 - h_3) \tag{3}$$

After the gas cooling process, the CO_2 is throttled in the throttling valve Ts, where:

$$h_3 = h_4. \tag{4}$$

The energy performance coefficient of the standard cycle can be assessed as:

$$COP_s = \frac{\dot{Q}_{GCs}}{\dot{W}_s} = \frac{h_2 - h_3}{h_2 - h_1}. \tag{5}$$

2.1.2. ENERGETIC ANALYSIS

Exergetic analysis is used to locate the malfunctions in the system. The exergetic balance equation corresponding to a closed system is:

$$\sum \dot{Ex}_Q^T = \sum \dot{W} + \sum \dot{I}. \tag{6}$$

Applying equation (6) to the heat pump cycle shown in Figure 1, the exergetic balance equation becomes:

$$\dot{Ex}_{QEVs}^{TEVs} + \dot{Ex}_{QGCs}^{TGCs} = \dot{W}_s + \dot{I}_{CPs} + \dot{I}_{Ts}. \tag{7}$$

Identifying the fuel and the product, equation (7) can be written at the level of the working agent:

$$\left|\dot{W}_s\right| = \left|\dot{Ex}_{QEVs}^{TEVs}\right| + \left|\dot{Ex}_{QGCs}^{TGCs}\right| + \dot{I}_{CPs} + \dot{I}_{Ts}. \tag{8}$$

Equation (8) written at the user level becomes:

$$\left|\dot{W}_s\right| = \left|\dot{Ex}_{QGCs}^{Thw}\right| + \dot{I}_{\Delta tEVs} + \dot{I}_{CPs} + \dot{I}_{\Delta tGCs} + \dot{I}_{Ts}. \tag{9}$$

The exergy destructions are calculated based on the Gouy-Stodola theorem [19]:

$$I = T_0 \cdot S_{gen}. \tag{10}$$

The exergetic performance coefficient of the heat pump is:

$$COP_{ex} = \frac{\dot{Ex}_{QGCs}^{Thw}}{\dot{W}_s}. \tag{11}$$

2.2. RESULTS AND DISCUSSIONS

A parametric sensibility analysis of the variation of the compression pressure p_2 is performed. Exergy destruction shares in the compression mechanical work input, and energetic and exergetic efficiencies were calculated.

The calculation algorithm was written based on the thermodynamic model described above and the thermophysical properties of the working fluid using the Engineering Equation Solvers program [20].

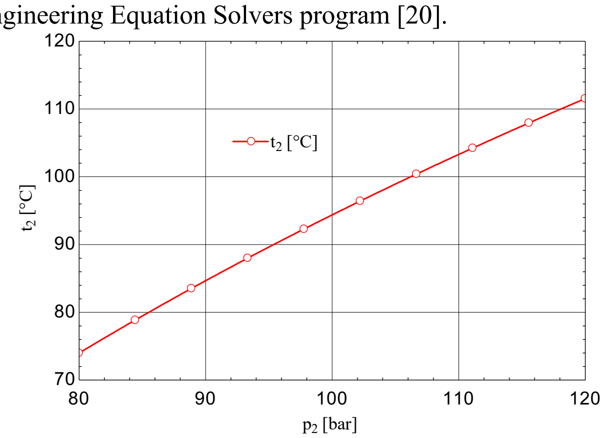


Fig. 2 – Standard cycle: compressor discharge temperature against the compression pressure.

Figure 2 presents the increase in the compressor discharge temperature with the compression pressure. As the temperature of the heated water remains constant, when temperature t_2 increases, a rise in the exergy destruction Ψ_{GCs} in the gas-cooler, associated with heat transfer across a finite temperature difference, is expected (Fig. 3,a).

Figure 3,a shows the exergy destruction shares in the compression mechanical work input for the key parts of the system. The highest exergy destruction is associated with the throttling process Ψ_{Ts} (Fig. 3,a). The increase in the compression pressure diminishes the exergy destruction Ψ_{Ts} in the throttling process, increasing both the exergetic COP_{ex} and energetic COP_s coefficients of performance (Fig.3).

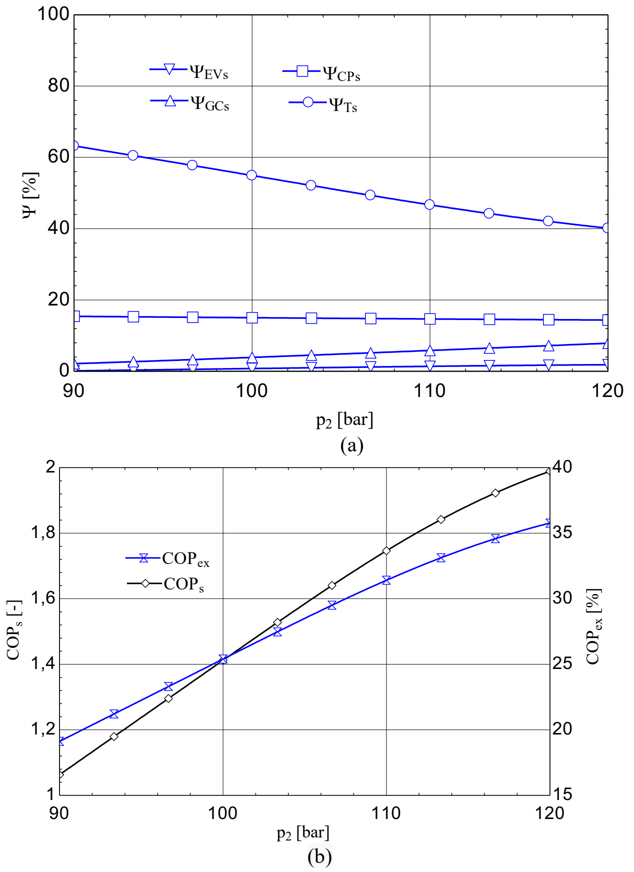


Fig. 3 – Standard cycle: (a) Shares of exergy destructions in the compression mechanical work input against the compression pressure; (b) Exergetic and energetic coefficients of performance against the compression pressure.

The optimization strategy must focus on decreasing the exergy destruction associated with the throttling process to increase the coefficients of performance of the standard heat pump cycle.

To continue to reduce the exergy destruction in the throttling process, besides the increase in the compression pressure (Fig. 3,a), a change in the inlet parameters of the throttling valve must be made.

To decrease the temperature of CO_2 at the inlet of the throttling valve, a subcooling of the gas after the gas-cooler using an external subcooler can be imagined. That could be done by coupling the standard cycle with an auxiliary one.

3. CO2 -CO2 COUPLED HEAT PUMP CYCLES

The flow chart of the two coupled heat pump cycles is presented in Fig. 4.

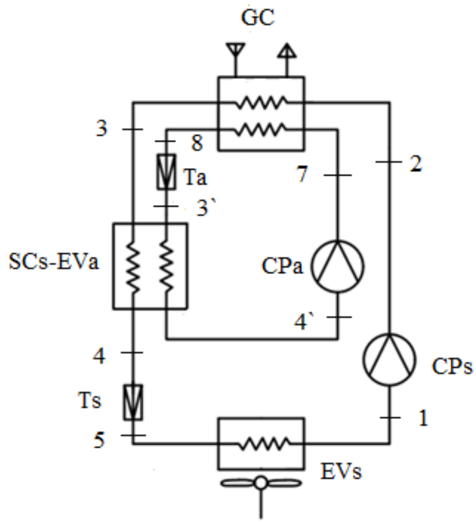


Fig. 4 – Flow chart of the coupled heat pump cycles.

The CO_2 gas leaving the gas-cooler of the standard cycle (state 3, Fig. 4) can be subcooled in the evaporator of CO_2 gas leaving the gas-cooler of the standard cycle (state 3, Fig. 4) can be subcooled in the evaporator of the auxiliary refrigerating cycle (process 3-4, Fig. 4). The evaporator of the auxiliary refrigerating cycle plays the part of a subcooler for the standard cycle. The heat of condensation of the auxiliary cycle (process 7-8, Fig. 4) can be used in the heating process of domestic hot water. In this way, the inverse auxiliary Rankine cycle plays the part of an auxiliary heat pump, and the two heat pumps (the standard and the auxiliary one) are coupled through the subcooler-evaporator (SCs – EVa). The main parameters that characterize the coupled heat pump are presented in Table 2.

Table 2.

Main parameters of the CO_2-CO_2 coupled heat pump

Description	Notation / unit of measure	Value
Temperature of CO_2 at the inlet of the subcooling process in the standard cycle	t_4 [°C]	25
Temperature of CO_2 at the outlet from the gas-cooler of the auxiliary cycle	t_8 [°C]	55
Auxiliary cycle vaporization temperature	$t_{3'} = t_{4'}$ [°C]	20

3.1 MATHEMATICAL MODELING OF THE CO_2-CO_2 COUPLED CYCLES

The two cycles are coupled via the SCs-EVa heat exchanger, which ensures the subcooling of the standard cycle working fluid. The energy balance of the SCs-Eva gives the mass flow rate of CO_2 in the auxiliary heat pump cycle.

The mechanical work input of the auxiliary cycle compressor is:

$$\dot{m}_a = \frac{\dot{Q}_{SCs-EVa}}{w_a} \tag{12}$$

The mechanical work input of the auxiliary cycle compressor is:

$$W_a = \dot{m}_a \cdot (h_7 - h_{4'}) \tag{13}$$

The absolute value of the auxiliary heat flow in the GCa gas cooler is:

$$\dot{Q}_{GCa} = \dot{m}_a \cdot (h_7 - h_8). \tag{14}$$

After rejecting heat, the working agent is throttled in the throttling valve Ta, where:

$$h_8 = h_{3`}. \quad (15)$$

The coefficient of performance of the auxiliary cycle becomes:

$$COP_a = \frac{\dot Q_{GCa}}{\dot W_a}. \quad (16)$$

The global energy performance coefficient in the case of coupled cycles becomes:

$$COP_c = \frac{\dot Q_{GCs} + \dot Q_{GCa}}{\dot W_s + \dot W_a}. \quad (17)$$

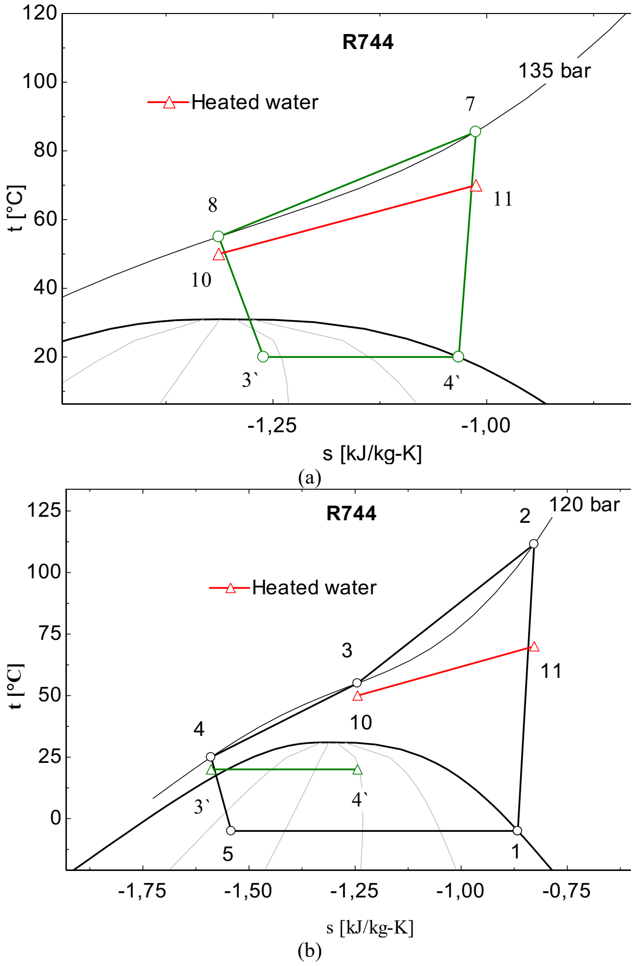


Fig. 5 – Coupled heat pump cycle -representation in T-s diagram: (a) Standard cycle; (b) Auxiliary cycle.

Applying eq. (6) for the coupled system shown in Fig. 4 and Fig. 5, the exergetic balance equation written at the level of the working fluid becomes:

$$\dot{Ex}_{\dot Q_{EVs}}^{T_{EVs}} + \dot{Ex}_{\dot Q_{GCs}}^{T_{GCs}} + \dot{Ex}_{\dot Q_{GCa}}^{T_{GCa}} = \dot W_{CPs} + \dot W_{CPa} + \sum \dot I. \quad (18)$$

To be able to highlight the exergy destruction at the level of the heat consumer and of the environment (the cold source heat supplier) eq. 21 becomes:

$$\left|\dot W_{CPs}\right| + \left|\dot W_{CPa}\right| = \left|\dot{Ex}_{\dot Q_{GCs}}^{T_{hw}}\right| + \left|\dot{Ex}_{\dot Q_{GCa}}^{T_{hw}}\right| + \left|\dot{Ex}_{\dot Q_{EVs}}^{T_{EVs}}\right| + \dot I_{\Delta T_{GCs}} + \dot I_{\Delta T_{GCa}} + \dot I_{CPs} + \dot I_{CPa} + \dot I_{Ts} + \dot I_{Ta} + \dot I_{\Delta T_{SCs-EVa}} \quad (19)$$

The relationship gives the total fuel for the coupled system:

$$F_c = \left|\dot W_s\right| + \left|\dot W_a\right|. \quad (20)$$

The energy product and the exergetic product of the coupled system are:

$$P_c^{en} = \left|\dot Q_{GCs}\right| + \left|\dot Q_{GCa}\right|, \quad (21)$$

$$P_c^{ex} = \left|\dot{Ex}_{\dot Q_{GCs}}^{T_{hw}}\right| + \left|\dot{Ex}_{\dot Q_{GCa}}^{T_{hw}}\right|. \quad (22)$$

Substituting in the relationship (17) relationships (20), (21), COP_c^{en} becomes:

$$COP_c^{en} = \frac{P_c^{en}}{F_c}. \quad (23)$$

The exergetic coefficient of performance of coupled cycles can be written:

$$COP_c^{ex} = \frac{P_c^{ex}}{F_c} = \frac{\left|\dot{Ex}_{\dot Q_{GCs}}^{T_{hw}}\right| + \left|\dot{Ex}_{\dot Q_{GCa}}^{T_{hw}}\right|}{\dot W_s + \dot W_a}. \quad (24)$$

Share of exergy destruction of each component:

$$\Psi[\%] = \frac{\dot I_i}{F_c} \cdot 100. \quad (25)$$

3.2. RESULTS AND DISCUSSIONS FOR THE CO_2-CO_2 COUPLED CYCLES

T-s diagrams of the standard and auxiliary cycles, part of the coupled cycle (Fig. 4), are presented in Fig. 5.

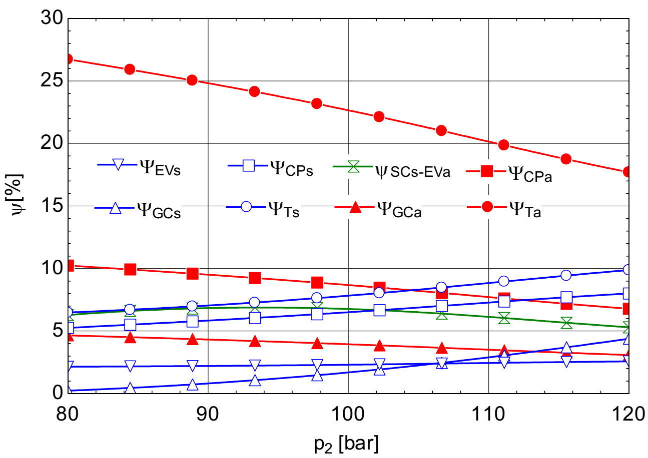
The heated water receives heat between temperatures t_2 and t_3 from the standard cycle and from t_7 to t_8 from the auxiliary one (Fig. 4).

The shares of the exergy destruction in the total mechanical work input in the coupled system and the variation of the energetic and exergetic coefficients of performance are presented in Fig. 6.

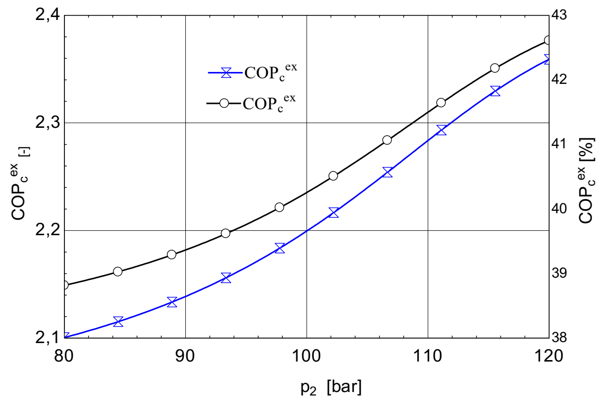
Comparing the performance coefficients (COP_s and COP_{ex}) from Fig. 3,b for the standard heat pump and Fig. 6,b, for the case of the coupled heat pump, an improvement in the global energetic and exergetic coefficients of performance is observed. This global improvement in the performance indices comes through subcooling, which leads to a substantial decrease in the exergy destruction in the throttling process of the standard cycle.

This improvement is possible because the exergy analysis shows the structural and functional dysfunctions and imperfections at the level of each device and process.

Exergy analysis only can provide information inside the system and offer optimization strategies and priorities.



(a)



(b)

Fig 6. – Coupled cycles: (a) Shares of exergy destructions in the total compression mechanical work input against the compression pressure in the standard cycle; (b) Exergetic and energetic coefficients of performance of the coupled cycles against the compression pressure in the standard cycle.

4. **CONCLUSIONS**

Increasing the performance coefficients is possible by tracking exergy destructions in each piece of equipment and process. Based on the exergetic analysis that identifies and locates the malfunctions of the system, a structural and functional optimization procedure can be established.

The standard transcritical CO_2 heat pump cycle shows low exergy destructions in the gas cooler, compressor, and evaporator but records a high exergy consumption in the throttling process.

The optimization strategy must focus on decreasing the exergy destruction associated with the throttling process to increase the coefficients of performance of the standard heat pump cycle.

To continue to reduce the exergy destruction in the throttling process, besides the increase in the compression pressure, a change in the inlet parameters of the throttling valve must be made.

To decrease the temperature of CO_2 at the inlet of the throttling valve, a subcooling of the gas after the gas-cooler through an external subcooler can be imagined. That has been done by coupling the standard cycle with an auxiliary one.

To increase the efficiency of the globally coupled system, the heat transferred outside in the gas cooler of the auxiliary heat pump is used in the water heating process. The energetic and exergetic efficiencies of the coupled system increased by 19 % and 18.3 %, respectively, compared to the standard heat pump cycle.

NOMENCLATURE

COP- *coefficient of performance*
CP- *compressor*
$\dot{E}x$- *exergy [kW]*
EV-*evaporator*
GC- *gas cooler*
h- *specific enthalpy [kJ/kg]*
$\dot{I}$- *destruction of exergy [kW]*
$\dot{m}$- *mass flow [kg/s]*
t- *temperature* $[^0C]$
T- *throttling valve*
$\dot{Q}$- *heat flow [kW]*
$\dot{W}$- *power [kW]*
Ψ[%]- *destruction of percentage exergy*
Subscripts
a- *auxiliary*
c- *coupled*
en- *energy*
ex- *exergy*
s-*standard*
Superscripts
hw- *heated water*

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