Analysis of a dual-purpose refrigerating and heat pump system

D Dima¹, C Ionita¹, E E Vasilescu¹, A T Gheorghian¹ and A Dobrovicescu¹⁺

¹Faculty of Mechanical Engineering and Mechatronics, University Politehnica of Bucharest

⁺E-mail: adobrovicescu@yahoo.com

Abstract. The study is looking for the optimum configuration of a system that requires refrigeration and heating. An ammonia refrigerator separated from a R152a heat pump are first considered. Water is the hot carrying fluid. Based on the exergy analysis, that points out the location and magnitude of each system operational malfunction, an optimization strategy is considered. Following the optimization procedure the two initial separated systems are coupled by using the condenser of the refrigerating unit to preheat water. Due to the exergy destruction in the water preheating condenser, ammonia is replaced by CO₂. The transcritical CO₂ heater diminish the exergy destruction with the heat transfer across a finite temperature difference. Solutions for further improving the efficiency of the dual-purpose system focus on the decrease in the throttling valve destruction in the CO₂ refrigerating unit.

1. Introduction
Heat pump systems represent an efficient and ecological alternative to heat recovery from sources with low energy potential to be used at a higher energy potential, in different residential, industrial and commercial applications.

Although in the year 2020 there are many working agents for heat pumps, due to the greenhouse gas reduction policy [1,2] the researchers focus on green working agents.

Heat pumps are a widely studied subject. X. Cao et al. [3] performs a theoretical analysis of the main heat pumps cycles used for residential heating purposes. The working agent used is R152a refrigerant. The analysis is focused on identifying the highest efficiency. Based on the exergetic analysis, the most efficient cycle has been found, this being the cycle in two compression stages with incomplete intermediate cooling.

D. Yang et al. [4] performs a theoretical and experimental analysis of a combined CO₂ and R134a cycle of a heat pump, used for heating domestic water. The analysis is focused on identifying the maximum efficiency, for the optimal parameters of the system functioning. The comparative analysis indicates an increase of up to 22% of the coefficient of performance compared to the classic compression system in one step.

A. Yataganbaba et al. [5] performs a theoretical analysis focused on the replacement of the R134a refrigerant in the existing heat pump systems and refrigeration installations with mechanical vapor compression, with the R1234yf and R1234ze refrigerants. Exergetic analysis indicates that the R1234ze has similar performance to the R134a refrigerant.

M. Pitarch et al. [6] performs the exergetic analysis of a subcritical heat pump cycle in which the R290 (propane) working agent evolves. The analysis is focused on identifying exergy destructions of
the components that make up the system. The results indicate the highest exergy destruction rates were
found in the throttling valve and condenser.

CO₂ heat pumps [7-9] are intensively analyzed due to the unique temperature slip feature during
the heat rejection process from the gas cooler. Many research studies [10-13] have been conducted to
improve the overall performance of the CO₂ heat pump cycle.

The aim of the present paper is finding the optimal configuration of a system that is capable of
generating heat and cold. In the first case, authors have considered two separate systems: one for the
cold production and one for the heat production. The paper aims to identify the exergy destruction of
the components that make up the systems, in order to find solutions to diminish them to increase the
overall efficiency of the system.

In order to reduce the exergy destruction of the system, both functional and constructive solutions
are used, finally a combined system is used which simultaneously produces cold and heat with a
higher efficiency than the classical system.

2. System description

Figure 1 schematically shows the separate systems, consisting of two independent cycles that do not
exchange mechanical work or heat between them. The two devices are composed of: Ev - evaporators,
Cp - compressors, Cd - condensers and T - throttling valves.

![Figure 1. Independent systems: 1) Refrigeration system; 2) Heat pump.](image)

The main thermodynamic parameters used to determine the performance of the described
equipment are specified in table 1.

| Description                                      | Parameter | Value |
|--------------------------------------------------|-----------|-------|
| The amount of cold                               | \( Q_v \) [kW] | 750   |
| The amount of heat                               | \( Q_c \) [kW] | 2500  |
| Cooling water temperature \( Cd1 \)              | \( t_9 \) [°C] | 20    |
| Inlet temperature of the heated water            | \( t_{11} \) [°C] | 20    |
| Outlet temperature of the heated water           | \( t_{13} \) [°C] | 100   |
| Inlet temperature of waste water carrying heat   | \( t_{14} \) [°C] | 50    |
| Cooling water inlet temperature                  | \( t_{16} \) [°C] | 12    |
| Cooling water outlet temperature                 | \( t_{17} \) [°C] | 7     |
| Minimum temperature difference in heat exchangers| \( \Delta t_P \) [°C] | 5     |

Table 1. The main parameters used in the thermodynamic analysis of the equipment.
Cycle 1 is used to achieve the cooling effect, by taking the heat from the water that cools from \( t_{16} \) to \( t_{17} \), in the \( Ev1 \) evaporator. The heat is then passed to the cooling water at the level of the \( Cd1 \) condenser, where it is discharged into the environment.

Cycle 2 is used for heating the water from \( t_{11} \) to \( t_{13} \) at the level of the \( Cd2 \) condenser, with the help of the heat pump cycle. The heat source at the level of the evaporator \( Ev2 \) is a waste heat source from an industrial process.

### 2.1. Energy analysis

By fixing the parameters specified in table 1, the thermodynamic model can be further developed in order to identify the other parameters.

In the evaporator \( Ev1 \) the working agent absorbs the heat from the water loop which cools from \( t_{16} \) to \( t_{17} \) to achieve the cooling effect, the flow rate of the agent in the cycle 1, \( m_1 \), can be calculated as follows:

\[
m_1 = \frac{Q_{Ev1}}{(h_1 - h_4)} = \frac{Q_v}{(h_1 - h_4)}.
\]

where, \( h \) represents enthalpy in the characteristic points.

Subsequently, the working agent is compressed using the \( Cp1 \) compressor which consumes the mechanical work:

\[
W_{Cp1} = m_1 \cdot (h_2 - h_1).
\]

The heat discharged from the condenser \( Cd1 \) can be calculated with the relation:

\[
Q_{Cd1} = m_1 \cdot (h_2 - h_3).
\]

After condensation the working agent is laminated in the throttling valve \( T1 \), where:

\[
h_4 = h_3.
\]

Cycle 2 is carried by the working agent with the flow rate \( m_2 \), which can be appreciated by the relation:

\[
m_2 = \frac{Q_{Cd2}}{(h_6 - h_7)} = \frac{Q_c}{(h_6 - h_7)}.
\]

In the evaporator \( Ev2 \) the working agent takes the heat from the waste heat source. The amount of heat taken to the \( Ev2 \) evaporator can be calculated as:

\[
Q_{Ev2} = m_2 \cdot (h_5 - h_8).
\]

The energy consumption required for driving the \( Cp2 \) compressor is:

\[
W_{Cp2} = m_2 \cdot (h_6 - h_8).
\]

The amount of heat transferred to the water that is heated from \( t_{11} \) to \( t_{13} \) is:

\[
Q_{Cd2} = m_2 \cdot (h_6 - h_7) = m_{w2} \cdot c_p \cdot (t_{13} - t_{11}) = Q_c,
\]

where: \( m_{w2} \) is the mass flow rate of heating water and \( c_p \) is the specific heat of the water.

Subsequently, the working agent is laminated into the throttling valve \( T2 \), and returns to the evaporator \( Ev2 \) level, where:

\[
h_8 = h_7.
\]
The total power required to drive the two compressors is:
\[ W = W_{Cp1} + W_{Cp2}. \] (10)

The coefficient of energy performance is:
\[ COP_{en} = \frac{Q_c + Q_c}{W}. \] (11)

2.2. Exergy analysis

For a cycle the exergy balance gives:
\[ \sum E_{xQ} = \sum W + \sum \dot{I}. \] (12)

For the refrigeration cycle working alone, equation (12), written at the refrigerant side becomes:
\[ \dot{E}_{xQ1} + \dot{E}_{xQ1} = W_{Cp1} + I_{Cp1} + \dot{I}_{T1}. \] (13)

Accounting for the sign of each exergy transfer from equation (13) one gets:
\[ \dot{W}_{Cp1} = \dot{E}_{xQ1} + \dot{E}_{xQ1} + I_{Cp1} + \dot{I}_{T1}. \] (14)

\( \dot{E}_{xQ} \) is the exergy of the heat rejected to the environment in the condenser \( Cd1 \); it is a loss
\( \left[ \dot{E}_{xQ} \right] = L_{Cd1}. \)

Considering the customer side (the chilled water) equation (13) can be rewritten such as:
\[ \left[ \dot{W}_{Cp1} \right] = \dot{E}_{xQ} + I_{\Delta T,Ev1} + I_{Cp1} + \dot{I}_{T1} + L_{Cd1}. \] (15)

\( \dot{E}_{xQ} \) = the exergy of the heat stream taken out from the chilled water at the temperature level of the chilled water;
\( I_{\Delta T,Ev1} \) = stream of exergy destruction due to heat transfer across a finite temperature difference in the evaporator;
\( I_{Cp1} \) = stream of exergy destruction due to the irreversible compression process in compressor \( Cp1 \);
\( \dot{I}_{T1} \) = stream of exergy destruction in the throttling process.

For the heat pump supplied, as a cold source, with heat at a higher temperature than the environmental one the exergy balance equation (equation 12) written on the recovered heating source side (in the evaporator) and the customer (hot water in the condenser) one, becomes:
\[ \left[ \dot{W}_{Cp2} + \dot{E}_{xQ} \right] = \dot{E}_{xQ} + I_{\Delta T,Ev2} + I_{Cp2} + I_{\Delta T,Cd2} + \dot{I}_{T2}. \] (16)

If one adds equation (15) and (16) gets the exergy balance of the global system:
\[ W_{cp1} + W_{cp2} + E_{Twh}^T + E_{Q}^T + \sum \dot{i}_1 + \sum \dot{i}_2 + L_{cd1}. \quad (17) \]

Superscripts \( Twh, Tch, w, Thw, Tc, Tv \) refer in order to the thermodynamic temperatures of the waste heat, chilled water, hot water, condensation temperature, vaporization temperature.

\[ F_{GS} = W_{cp1} + W_{cp2}. \quad (18) \]

The energetical and exergetical Products of the global systems (GS) are:

\[ P_{GS}^{en} = Q_{Ev1} + Q_{cd2}. \quad (19) \]

\[ P_{GS}^{ex} = E_{Tch,w}^T + E_{Thw}^T. \quad (20) \]

According to the energetic or exergetic approach, the coefficient of performance of the global system becomes:

\[ COP_{GS}^{en} = \frac{P_{GS}^{en}}{F_{GS}} = \frac{Q_{Ev1} + Q_{cd2}}{W_{cp1} + W_{cp2}}. \quad (21) \]

and, correspondingly

\[ COP_{GS}^{ex} = \frac{P_{GS}^{ex}}{F_{GS}} = \frac{E_{Tch,w}^T + E_{Thw}^T}{W_{cp1} + W_{cp2}}. \quad (22) \]

The share of a destruction in the Fuel consume is calculated such that:

\[ \Psi \left[ \% \right] = \frac{\dot{i}}{F_{GS}} \cdot 100. \quad (23) \]

With the definition of the \( COP_{GS}^{ex} \) from equation 22 it follows that:

\[ COP_{GS}^{ex} = 1 + \frac{E_{Thw}^T}{W_{cp1} + W_{cp2}} - \sum \Psi_i. \quad (24) \]

As \( E_{Thw}^T \) is fixed, the increase in the \( COP_{GS}^{ex} \) depends on the diminishing of the exergy destruction shares \( \Psi \).

The optimization procedure will chase the apparatuses and processes that consume large quantities of exergy (destruct exergy) and tries to find ways to diminish them.
3. Results and discussion
The calculation program was written based on the thermodynamic model described above, and the thermophysical properties of the working fluids were determined using the Engineering Equation Solvers program [14].

3.1. Separate cycles R717 and R152a
In a first instance, according to the basic model described in figure 1, for cooling the water from \( t_{16} \) to \( t_{17} \), ammonia (R717) is used in cycle 1; in cycle 2, the one responsible for heating water from \( t_{11} \) to \( t_{13} \) evolves freon R152a.

![Figure 2](image2.png)

**Figure 2.** The shares of the exergy destructions for cycles with R717 and R152a working separately.

Figure 2 shows that the largest exergy destruction \( \Psi \) is due to the loss of heat exergy in the condenser of the refrigeration system. This destruction increases with the increase in the condensation temperature \( t_{cR717} \). The diminishing of this exergy destruction represents a target in the optimization procedure.

![Figure 3](image3.png)

**Figure 3.** Exergies at the heat exchangers level for cycles with R717 and R152a working separately.
Due to the fact that both the heating and refrigerating loads are fixed, but also the temperature of the industrial water (waste) used as the source of heat extraction for cycle 2, the functional optimization can be done at the condenser level of cycle 1. Thus, the variation of the condensation temperature of cycle 1 was obtained.

![Figure 4. COP<sub>en</sub> and COP<sub>ex</sub> for the global system with separate cycles with R717 and R152a.](image)

Following figure 3 it is observed that the exergy at the level of condenser 2 has a useful potential. To recover this exergy the two cycles are coupled at the condensers $Cd1$ and $Cd2$ level as in figure 5.

Figure 4 gives the variation of the performance coefficients against the condensation temperature $t_{cR717}$. The behavior of the refrigeration cycle is determinant.

3.2. Cycles connected by condenses R717 and R152a

In order to improve the overall performance, the two cycles are connected by condensers according to figure 5. In this case cycle 1 has a dual purpose. The feed water is heated in two stages, thus cycle 1 preheats the feed water from $t_{11}$ to $t_{12}$, while at the vaporizer level 1 cold water loop is cooled from $t_{16}$ to $t_{17}$. Preheated water with intermediate temperature $t_{12}$ is introduced into the condenser $Cd2$ where it is heated to temperature $t_{13}$.

![Figure 5. Systems coupled with condensers.](image)
Due to the fact that the supply water has fixed parameters (flow rate, inlet temperature, outlet temperature), a new operating condition occurs which forces the temperature at condenser level 1 to be at least 50 [°C]. This temperature is required to ensure a minimum temperature of 5 [K] in the heat exchanger.

The T-s diagrams for the two units are presented in figures 6 and 7.

**Figure 6.** The T-s diagram for the refrigeration cycle with R717.

**Figure 7.** The T-s diagram for the heat pump cycle with R152a.
Figure 8. Shares of exergy destructions for cycles with R717 and R152a coupled through condensers.

One can observe a substantial decrease in the exergy destruction in the condenser when using R744 compared to R717 (figures 8 and 11).

The coefficient of performance reaches a maximum for the compression pressure $p_3$ in the gas cooler, of the R744 refrigeration cycle, of $p_3 = 77$ bar.

Figure 9. $\text{COP}_{\text{en}}$ and $\text{COP}_{\text{ex}}$ for cycles with R717 and R152a coupled through condensers.

Comparing the performance indices (COP and $\eta_{\text{ex}}$) of the separate cycles (figure 4) and the cycles coupled by condensers (figure 9), a considerable improvement is observed for the cycles coupled by condensers. This improvement comes through the use of exergy from condenser 1 to preheat the supply water. This preheating also reduces the exergy destruction in cycle 2 because the heat transfer at the condenser 2 level is due to a lower temperature difference.
Following the exergy destructions (figure 8) of the components that make up the system coupled by condensers, it is observed that the exergy destruction in the \( \text{Cd2} \) condenser with R717 shows a significant value due to the large temperature difference in this heat exchanger.

### 3.3. Cycles with R744 and R152a connected by condensers

In order to reduce this destruction, the working agent in cycle 1 ammonia is replaced with CO\(_2\) while keeping the same construction scheme, and in the second cycle, the same working agent R152a. The CO\(_2\) refrigeration cycle evolves in the transcritical domain.

The T-s diagrams for the refrigeration cycle operating with R744 and for the heat pump with R152a are presented in figures 7 and 10.

**Figure 10.** The T-s diagram for the refrigerating cycle with R744.

**Figure 11.** Shares of exergy destructions for cycles with R744 and R152a coupled through condensers.
Figure 12 shows that there is an optimum compression pressure in the CO₂ refrigeration cycle (76 bar) for which the overall coefficients of performance reach a maximum.

Unfortunately the exergy destruction drop in the R744 gas cooler (cg) is accompanied by a rise in the exergy destruction associated with the throttling process in the CO₂ cycle (figures 8 and 11).

A reduction in the exergy destruction in the refrigeration cycle with CO₂ can be obtained by the use of an internal heat exchanger to reduce the temperature at the inlet of the throttling valve.

4. Conclusions

A dual purposes system that offers refrigeration and heating is discussed.

First the global system operates with two separate units. The refrigerator uses ammonia while the heat pump operates with R152a.

The exergy analysis gives the strategy for the improvement of the global system.

The large exergy loss in the refrigeration unit operating alone suggest the use of the heat rejected in the condenser for preheating the hot water.

The exergetic analysis of the global system with the condenser completed makes evidence of the large exergy destruction in the condenser of the R717 refrigerating unit due to heat transfer across a large temperature difference.

The replacement of R717 with R744 is suggested due to the glide of the CO₂ temperature in the gas cooler.

The use of the CO₂ transcritical cycle reduces the exergy destruction in the gas cooler but introduces an increase in the exergy destruction in the throttling process.

For a future study the use of an internal heat exchanger in the CO₂ refrigeration cycle is suggested in the aim of reducing the temperature at the throttling valve inlet.

5. References

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