CO₂ high temperature heat pump – a promising solution

O Talaba¹, D Dima¹*, V Buiuc¹, C Ionita¹, M F Stefanescu¹, A Serban¹ and A Dobrovicescu¹

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

*E-mail: dimadaniel08@yahoo.com

Abstract. The study focuses on obtaining hot water above 80°C for industrial purposes. Compared to the electrical heating the use of a heat pump appears much more efficient. A comparative analysis using different refrigerants is carried out. The cold source is waste heat from an industrial process. The exergetic analysis reveals the advantages and weaknesses of using CO₂ as refrigerant. Based on the exergy destruction concept several schematics of two-stage CO₂ Heat Pumps are considered. For high temperature water heating, a CO₂ Heat Pump system represents a promising solution.

1. Introduction

High temperature (over 65°C) heat pumps (HTHP) deliver useful heat at 66°C – 150°C.

Thousands of units are operating in Europe, North America and Asia to supply heat up to 95°C [1].

The present study focuses on heat delivery at 90°C. The refrigerant is CO₂ whose use in heat pumps is mainly recommended for cold regions of the globe characterized by an annual average temperature not exceeding t₀=15°C. Such systems are advised when a waste source of heat is available [2],[3].

Heat pumps are a good energy solution with low carbon negative impact. Electricity to run such heat pumps could be provided from renewable sources.

As classic applications of HTHP one can mention the food industry for drying and washing purposes [4].

Refrigerants for industrial heat pumps are selected based on their global warming impact [5]. Some promising refrigerants with their characteristics are listed in table 1.

| Refrigerant | Chemical formula | GWP | Flammability | Tc [°C] | Pc [MPA] | NBP [°C] |
|-------------|------------------|-----|--------------|--------|---------|---------|
| R-601       | CH₃CH2CH2CH2CH₃  | 5   | yes          | 196.6  | 3.37    | 36.1    |
| R-717       | NH₃              | 0   | yes          | 132.25 | 11.33   | -33.33  |
| R-744       | CO₂              | 1   | none         | 30.98  | 7.3773  | -78.50  |
| R-1233zd(E) | (E)CF₃-CH=CCIH   | 1   | weak         | 166.5  | 3.62    | 18      |
| R-1224yd(Z) | CF₃CH₂F         | <1  | weak         | 155.5  | 3.33    | 14      |
| R-1234ze(Z) | CF₃CH=CHF(Z)    | <1  | weak         | 150.1  | 3.53    | 9.8     |
| R-245fa     | CF₃CH₂CHF₂      | 858 | none         | 154.01 | 3.651   | 14.9    |

CO₂ among other refrigerants is a natural substance with harmless impact on the environment. In addition CO₂ is fireproof, not toxic and can be freely released to the environment.
The problem of low critical temperature of carbon dioxide is overcome by operating the system in the transcritical region [7], [8], [9].

Among the properties that recommend CO$_2$ as refrigerant in HTHP are the low swept volume per kW of heat delivered in the gas cooler and the low compression ratio in the compressors even if it works at high pressures.

The interest in using CO$_2$ in heat pumps is well known. At every two years the International Institute of Refrigeration organizes an International Conference on Ammonia and CO$_2$ Refrigeration Technologies [10] and a Gustav Lorenzen Conference on Natural Refrigerants [11] where CO$_2$ heat pump systems represent a main topic.

The optimization of the operating and design of CO$_2$ heat pumps represents a major preoccupation for researchers. Changhyun Baek et al. [12] experimentally analyze the performance characteristics of a heat pump for district water heating in low temperature climate. The paper points out that compared to other HFC refrigerants the CO$_2$ temperature difference in the gas cooler is practically balanced offering the possibility of reaching high temperature levels for the hot water.

Wang D. et al. [13] analyze comparatively the performance of a single and two-stage vapor compression transcritical heat pump cycles operating with CO$_2$ or an azeotropy mixture of CO$_2$/ethane. The use of a two-stage system improves substantially the coefficient of performance of the system.

2. Material and methods

The one stage system is presented in figure 1.

The mathematical model is based on the CO$_2$ real fluid behavior given by the fundamental equation of state developed by R. Span and W. Wagner [14].

The simulating of the system operation was performed with the Engineering Equation Solver (EES) [15].

The considered CO$_2$ heat pump operates in a colder climate where the average annual temperature is around $t_0=15^\circ$C.

A waste heat from an industrial process at $t_w=30^\circ$C is available.

The heat pump supplies heat and hot water at up to 90$^\circ$C.

Figures 1, 2 and 3 show the above-mentioned schematics.

![Figure 1. One stage transcritical cycle. a) the flow chart; b) representation of the cycle in the T-s diagram.](image)
Three schematics are considered: figure 1 a one stage transcritical system considered as the reference one; figure 2 a two-stage system with the injection of a cold stream between the two compression stages [16]; figure 3 a two-stage cycle with intermediary external cooling by preheating a stream of water.

The analysis and the choice of the structural evolution of the CO$_2$ heat pump cycle is based on the exergy analysis that can reveal the way in which the electrical energy input to run the heat pump is consumed in each part of the system [17], [18], [19].

![Figure 2](image1.png)

**Figure 2.** Two-stage system with the injection of a cold stream between the two compression stages. a) the flow chart; b) representation of the cycle in the p-h diagram.

![Figure 3](image2.png)

**Figure 3.** Two-stage cycle with intermediary external cooling by preheating a stream of water. a) the flow chart; b) representation of the cycle in the p-h diagram.
3. Mathematical formulation

3.1. Modeling of the reference system

The one stage system is presented in figure 1. The analysis is done for 1kW of heat supplied in the gas cooler.

For a cycle the exergy balance gives:

\[ \sum \dot{E}_{x} = \sum \dot{W} + \sum \dot{I}. \]  (1)

Following the one stage heat pump schematic, equation (1) on the refrigerant side becomes:

\[ \dot{E}_{x_{V}} + \dot{E}_{x_{gc}} = \dot{W}_{cp} + \dot{I}_{cp} + \dot{I}_{t}. \]  (2)

Accounting for the sign of each exergy transfer equation (2) becomes:

\[ |\dot{W}_{cp}| + |\dot{E}_{x_{V}}| = |\dot{E}_{x_{gc}}| + \dot{I}_{cp} + \dot{I}_{t}. \]  (3)

Considering the temperature differences in the evaporator and gas-cooler, equation (3) written on the customer side gives:

\[ |\dot{W}_{cp}| + |\dot{E}_{x_{Q}}| = |\dot{E}_{x_{Q}}| + \dot{I}_{\Delta T,Ev} + \dot{I}_{\Delta T,gc} + \dot{I}_{t}. \]  (4)

where the exergy destructions refer in order to the irreversibilities associated with heat transfer across a temperature difference in the evaporator, compressor, heat transfer across a temperature difference in the gas cooler and throttling valve.

The energetical and exergetical coefficients of performance are:

\[ \text{COP}_{en} = \frac{\dot{Q}_{gc}}{|\dot{W}_{cp}| + |\dot{E}_{x_{Q}}|}. \quad (5) \]

\[ \text{COP}_{ex} = \frac{|\dot{E}_{x_{Q}}|}{|\dot{W}_{cp}| + |\dot{E}_{x_{Q}}|}. \quad (6) \]

The coefficient of exergy destruction is defined as the ratio between exergy destruction and the total exergy input:

\[ \Psi_{j} = \frac{\dot{I}_{j}}{|\dot{W}_{cp}| + |\dot{E}_{x_{Q}}|} \times 100. \quad (7) \]

3.2. Modeling of the two-stage system with the injection of a cold stream between the two compression stages

The heat pump cycle is presented in figure 2. A parametric study at the variation of the intermediary pressure is performed.

3.3. Modeling of the two-stage cycle with intermediary external cooling

The flow chart of the installation is presented in figure 3. The intermediary cooling is performed by heating a stream of water.

This two-stage cycle is characterized by a unique mass flow rate and single throttling valve.
4. Results and discussion

4.1. The reference cycle

In figures 4 and 5 the coefficients of performance and exergies destructions as part of the electrical energy consumption are shown for the one stage cycle taken as reference.

The decisional parameter is the compression ratio $\pi$.

**Figure 4.** One stage system (figure 1). Coefficients of performance against de compression ratio.

**Figure 5.** One stage system (figure 1). Shares of exergy destruction in the compressor power input against de compression ratio.
Figure 5 shows a very high value for the exergy destruction in the throttling valve $\Psi_t$ followed by the destruction in the compressor $\Psi_{cp}$ and gas-cooler $\Psi_t$.

The exergetic coefficient of performance $\text{COP}_{ex}$ increases with the increase in the compression ratio $\pi$ (figure 4) following the decrease in the throttling process destruction (figure 5) which is the preponderant destruction.

Even if for CO$_2$ in the gas-cooler there is a temperature glide that makes almost uniform the temperature difference, the high temperature of CO$_2$ at the discharge from the compressor is responsible for the high exergy destruction $\Psi_{\Delta T_{gc}}$.

To reduce the high temperature in the compressor discharge and reduce the irreversibility with heat transfer across a temperature difference in the gas-cooler, a two-stage cycle is suggested.

### 4.2. The two-stage system with the injection of a cold stream between the two compression stages

Figure 2 shows the schematic of a two-stage system with the injection of a cold stream between the two compression stages. The temperature of the gas at the inlet of the high-pressure compression stage decreases and so does the temperature at the gas cooler inlet.

Figures 6 and 7 show the variation of the COP and exergy destruction coefficients $\Psi$ at the variation of the intermediary pressure.

![Figure 6](image)

**Figure 6.** The two-stage system with the injection of a cold stream (figure 2). Coefficients of performance against the intermediary pressure.

As expected, the share of the exergy destruction in the gas cooler has decreased, but due to the existence of two throttling processes the destruction associated to these processes has increased overcoming the gain in the gas cooler. The share of the exergy destruction in the compressor has slightly increased. The energetical and exergetical COPs have decreased.
Figure 7. The two-stage system with the injection of a cold stream (figure 2). Shares of exergy destruction in the compressor power input against the intermediary pressure.

Table 2 shows comparatively the simulation results for the one stage and the two-stage system with intercooling by the injection of a stream of cold refrigerant in the gas discharged by the low-pressure compressor.

For the two schematics the compared operating regimes correspond to the highest values for the COPex.

Table 2. Exergetic and energetic characteristics for the one stage HP and the two-stage HP with injection.

| Schematic                | $\Psi_{\Delta T,gc}$ [%] | $\Psi_{cp}$ [%] | $\Psi_t$ [%] | $\Sigma \Psi$ [%] | COPex | COPen |
|--------------------------|--------------------------|-----------------|-------------|-------------------|-------|-------|
| One stage HP             | 7.99                     | 14.12           | 45.87       | 67.98             | 0.29  | 1.66  |
| Two-stage HP with injection| 6.7                      | 14.59           | 48.29       | 69.58             | 0.28  | 1.58  |

4.3. Two-stage cycle with intermediary external cooling

To keep the advantage brought by the reducing of the inlet gas cooler temperature but to reduce the exergy destruction due to the throttling process a schematic with external intermediary cooling figure 3 will be considered.

This flow pipe is characterized by just one throttling.

Figures 8 and 9 show the behavior of this schematic at the variation of the intermediary pressure.

This schematic operates with the highest coefficient of performance. It has kept the same exergy destruction in the throttling valve as the one stage system but with a lower exergy destruction in the gas-cooler.
Figure 8. Two-stage cycle with intermediary external cooling (figure 3). Coefficients of performance versus the intermediary pressure.

Figure 9. Two-stage cycle with intermediary external cooling (figure 3). Shares of exergy destruction in the compressor power input versus the intermediary pressure.
Table 3 shows the exergetic characteristics of the schematic with external intermediary cooling. The comparative analysis of the operating of the three schematics (figures 1, 2 and 3) can be followed by the inspection of table 2 and table 3.

Table 3. Exergetic and energetic characteristics of the two-stage heat pump with external intermediary cooling.

| $p_{int}$ [bar] | $\Psi_{\Delta T,gc}$ [%] | $\Psi_{cp}$ [%] | $\Psi_{t}$ [%] | $\Sigma \Psi$ [%] | COP$_{ex}$ | COP$_{en}$ |
|-----------------|---------------------|----------------|----------------|-----------------|-----------|-----------|
| 55              | 7.76                | 14.4           | 45.6           | 67.76           | 0.303     | 1.627     |

5. Conclusions
Carbon dioxide is a natural refrigerant, nontoxic and fire free. These qualities recommend it for the use in a large field of activities such that food industry.

Due to its low critical temperature the use of carbon dioxide in heat pumps is mainly advised in regions with cold climate.

Different one stage and two-stage configurations have been modeled and parametrically tested. The exergetic analysis has been used in the optimum structure and operating regime search.

Trying to diminish the exergy destrucions associated to heat transfer across a finite temperature difference in the gas cooler and the throttling process a two-stage schematic has been considered.

For the two-stage system the existence of two throttling processes will cancel the positive effect of the exergy destruction decrease in the gas-cooler.

The exergetic analysis gives close results for the two-stage with external precooling and the reference schematic in one stage.

Observing that the comparative analysis has been performed for 1 kW of heat transferred in the gas-cooler the increase in the exergetic coefficient of performance for the two-stage cycle with external intermediary cooling is due to the additional heat transferred to the heated water in the intermediary cooler.

6. Nomenclature

Cp compressor
Ev evaporator
$\dot{E}x_Q$ exergy current of heat
$\dot{E}x_{T_v}Q$ exergy current of the heat transferred at temperature $T_v$ to the refrigerant in the evaporator
$\dot{E}x_{T_{gc}}Q$ exergy current of the heat transferred outside, from the refrigerant, at temperature $T_{gc}$, in the gas-cooler
$\dot{E}x_{T_{5-6}}Q$ exergy current of the heat at the thermodynamic temperature level of the waste source of heat
$\dot{E}x_{T_{7-8}}Q$ exergy current of the heat at the thermodynamic temperature level of the hot water
GC gas-cooler
HP heat pump
\( \dot{I} \) current of exergy destruction
\( \dot{I}_{cp} \) current of exergy destruction in the compressor
\( \dot{I}_t \) current of exergy destruction in the throttling valve
\( \dot{I}_{\Delta T,Ev} \) current of exergy destruction due to heat transfer across a temperature difference in the evaporator
\( \dot{I}_{\Delta T,gc} \) current of exergy destruction due to heat transfer across a temperature difference in the gas-cooler
\( T_V \) thermodynamic temperature of the evaporation process
\( T_{gc} \) thermodynamic temperature of the gas-cooling process
\( \dot{W} \) mechanical power exchange
\( |\dot{W}_{cp}| \) compressor electrical power input

Greek symbols
\( \Psi \) exergy destruction factor defined as the ratio between exergy destruction and the total exergy input

7. References
[1] High Temperature Heat Pumps for the Australian food industry August 2017 Opportunities and assessment https://www.airah.org.au/Content_Files/Industryresearch/19-09-17_A2EP_HT_Heat_pump_report.pdf
[2] Saikawa M and Koyama S 2016 Thermodynamic analysis of vapor compression heat pump cycle for tap water heating and development of CO\(_2\) heat pump water heater for residential use Appl. Therm. Eng. 106 1236-43
[3] Hakkaki-Fard A, Aidoun Z and Ouzzane M 2015 Improving cold climate air-source heat pump performance with refrigerant mixtures Appl. Therm. Eng. 78 695-703
[4] Klocke R, Schmidt E L and Steimle F 2001 Carbon dioxide as a working fluid in drying heat pumps Int. J. Refrig. 24 100-7
[5] Bolaji B O and Huan Z 2013 Ozone depletion and global warming: Case for the use of natural refrigerant – A review Renew. Sustain. Energy Rev. 18 49-54
[6] Arpagaus C Kuster R Prinz M Bless F Uhlmann M Buchel E Frei S Schiffmann J and Bertsch S High temperature heat pump using HFO and HCFO refrigerants – system design and experimental results. https://cm.icr2019.org/files/lecture/eposter_pdf/242/242.pdf
[7] Cavallini A, Cecchinato L, Corradi M, Fornasieri E and Zilio C 2005 Two-stage transcritical carbon dioxide cycle optimization: A theoretical and experimental analysis Int. J. Refrig. 28 1274-83
[8] Cecchinato L, Chiarello M, Corradi M, Fornasieri E, Minetto S, Stringari P and Zilio C 2009 Thermodynamic analysis of different two-stage transcritical carbon dioxide cycles Int. J. Refrig. 32 1058-67
[9] Jiang Y, Ma Y, Li M and Fu L 2013 An experimental study of trans-critical CO\(_2\) water-water heat pump using compact tube-in-tube heat exchangers Energy Convers. Manag 76 92-100
[10] Institut international du froid 2019 8th Conference on Ammonia and CO2 Refrigeration Technology (Paris: IIF-IIR) Online: https://iifiir.org/en/fridoc/6077
[11] Institut international du froid 2018 13th IIR Gustav Lorentzen Conference on Natural Refrigerants (GL2018) (Paris: IIF-IIR) Online: https://iifiir.org/en/fridoc/6066
[12] Changhyun B, Jaehyeok H, Jongho J, Honghyun C and Yongchan K 2014 Performance characteristics of a two-stage CO$_2$ heat pump water heater adopting a sub-cooler vapor injection cycle at various conditions Energy 77 570-578

[13] Wang D, Chen Z, Gu Z P, Liu Y R, Kou Z L and Tao L R 2020 Performance analysis and comprehensive comparison between CO$_2$ and CO$_2$/ethane azeotropy mixture as a refrigerant used in single stage and two-stage vapor compression transcritical cycles International Journal of Refrigeration 115 39-47

[14] Span R and Wagner W 1996 A New Equation of State for Carbon Dioxide Covering the Fluid Region from the Triple-Point Temperature to 1100 K at Pressures up to 800 MPa J. Phys. Chem, Ref. Data 25 Issue 6

[15] Engineering Equation Solver (EES) http://fchartsoftware.com/ees/

[16] Cho H, Baek C, Park C and Kim Y 2009 Performance evaluation of a two-stage CO$_2$ cycle with gas injection in the cooling mode operation Int. J. Refrig. 32 40-46

[17] Sarkar J, Bhattacharyya S and Gopal M R 2005 Transcritical CO$_2$ heat pump systems: Exergy analysis including heat transfer and fluid flow effects Energy Convers. Manag. 46 2053-67

[18] Ghazizade-Ahsae H and Ameri M 2018 Energy and exergy investigation of carbon dioxide direct-expansion geothermal heat pump Appl. Therm. Eng 129 165-78

[19] White S D, Yarrall M G, Cleland D J and Hedley R A 2002 Modelling the performance of a transcritical CO2 heat pump for high temperature heating Int. J. Refrig 25 479-86