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INTERNATIONAL ADVANCED RESEARCHES and ENGINEERING JOURNAL

Journal homepage: www.dergipark.org.tr/en/pub/iarej

e-ISSN: 2618-575X

International Open Access

Volume 06
Issue 03

December, 2022

Research Article

Experimental analysis of a transcritical heat pump system with CO_2 refrigerant

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ARTICLE INFO

Article history:
Received 20 June 2022
Accepted 23 November 2022
Published 15 December 2022

Keywords:
Capillary tube
CO_2
Energy efficiency
EPC
Exergy efficiency
Heat pump
Transcritical

ABSTRACT

Today, it is seen that increasing environmental pollution is getting ahead of the increasing energy need. Therefore, more environmentally friendly and more economical refrigerants are needed. In this context, carbon dioxide appears as a natural refrigerant in cooling systems and heat pump (HP) systems, and it has been widely used in recent years. In this study, a single-stage heat pump system with a CO_2 refrigerant, with a transcritical cycle, has been experimentally studied. The system is designed as a water-to-water heat pump. The performance of the system has been determined experimentally. In the system, capillary pipes with a diameter of 2.00 mm and two different lengths are used. It is aimed to create different evaporation pressures with two capillary tubes. The first capillary tube is 2.40 m long and the second is 1.20 m long. Gas cooler pressures, gas cooler and evaporator cooling water mass flow rates were kept the same for both cases. A certain gas charge was made and measurements were made for both cases. Thermodynamic analysis and comparison of the system were made. In the short capillary tube system, it was observed that the COP_{HP} value was 7.2% higher, the CO_2 mass flow rate increased by 9.1% to achieve the same gas refrigerant pressure value, and the power consumption in the compressor decreased by 1.8%. In addition, the gas cooler outlet temperature, the evaporator outlet temperature and the change in ambient temperatures, as well as the exergetic destruction and exergetic efficiencies in the system and system components are presented in figures with EES.

1. Introduction

Heat pump systems, which have become very competitive in domestic uses, needed alternative, innovative and most importantly environmentally friendly refrigerants that do not harm the ozone layer. In line with these needs, the depletion potential of the ozone layer (ODP=0) and global warming potential (GWP=1), and carbon dioxide (CO_2), a natural refrigerant, have inspired our studies in recent years. In our studies, a capillary tube was used for the expansion valve, which is one of the main components used in a heat pump system. Many elements such as electronic valve, thermostatic valve and capillary pipe are used in the expansion process. The most important reason why we chose the capillary tube in this study is that it is cheap and easy to find. In this context, it is important to determine the operating temperature and conditions of the systems to be installed and to choose the appropriate capillary tube according to the need.

Some studies in the literature; Elbir et al., They presented with graphics and pictures how the test results will change according to the thermodynamic rules in the same system and with which elements they will change when the gas cooler pressure of 75 bar is increased to 100 bar gas cooler pressure and the mass flow rate of the water providing cooling from water to water is increased [1]. Jadhav and Agrawal., calculated the effect of various geometric parameters such as tube diameter, length, roughness and slope on mass flow rate, cooling capacity and COP [2]. Jadhav and Agrawal., found that the percentage decrease in mass flow rate increased as the pipe diameter and the length of the helical capillary pipe increased [3]. Rocha et al., investigated the effect of surface roughness on mass flow rate from adiabatic capillary tubes in their study [4]. Wang et al., stated that

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DOI: 10.35860/iarej.1132994

the refrigerant charge plays an important role in optimizing the system performance [5]. Andres et al., experimentally investigated the optimum operating conditions of the COP value of the subcooled transcritical CO_2 system at different operating pressures. From the experimental data, only two general expressions as a function of the evaporation level and the gas cooler outlet temperature are specified to determine the optimum pressure and subcooling [6]. Song et al., In their work, they experimentally incorporated the variation of the gas refrigerant outlet in a transcritical CO_2 heat pump system using an electronic expansion valve (EEV) and a capillary tube. In terms of performance, they said that the transcritical CO_2 heat pump using capillary tube is promising. [7]. Agrawal et al**.**, studied optimum diameter length and diameter correlation. They said the pressure gradient is significantly higher for CO_2 than with conventional refrigerants, with a shorter capillary tube length [8]. Wang et al., They calculated the optimal combination of capillary tube size and refrigerant charge and made the experimental comparison in their study. In the experiment, they said that the length of the capillary tube was shortened by 8.77% and the optimum refrigerant charge increased by about 5%. In the experimental study, they focused on how the reduction of refrigerant charge can affect the coefficient of heating performance [9]. Agrawal and Bhattacharyya, Instead of an expansion valve, a capillary tube was proposed, and optimum operating conditions were investigated. Optimal conditions of gas refrigerant pressure were investigated, and their effect on COP value was investigated. They made recommendations on the most appropriate diameter and length [10]. Song et al., In the system they designed, they compared the transcritical CO_2 heat pump system, which is dependent on the gas cooler outlet temperature, using both an electronic expansion valve (EEV) and a capillary tube. They said that the results obtained with the capillary tube obtained close results in the heat pump system using EEV[11]. Agrawal and Bhattacharyya, Parameters that affect system performance have been identified. They emphasized that the temperature of the water entering the gas cooler as a coolant source is more important than the flow rate. They emphasized the effect on system performance by determining the optimum gas charge and optimum capillary tube [12]. Wang and Lu, In their work, they conducted experimental and theoretical studies on the optimization of capillary tube dimensions with gas charge. In the results they determined, it was emphasized that the shortening of the capillary tube length increased the amount of refrigerant. In addition, they made calculations on how the decrease in the amount of refrigerant could reduce the performance of the system [13]. Date et al., In their study, they presented studies on optimum system pressure values and refrigerant amounts in the system using EEV [14]. Jadhav and Agrawal, Thermodynamic analyses of an adiabatic spiral and a helical capillary tube in a CO_2 transcritical system were performed with the subcritical R22 fluid. They said that the decrease in the mass flow rate of the refrigerant is more pronounced in the spiral capillary tube than in the spiral capillary tube effect [15]. Rocha et al., They developed an algebraic equation and dimensionless correlation for straight capillaries. An algebraic solution for helical capillary tubes in the transcritical CO_2 cycle is highlighted in the literature [16]. Anka et al., They developed alternative methods by using the average of the refrigerant charge amounts in optimum summer and winter conditions and the outdoor temperature that gives the same COP of the IDC (internet data center) cooling system in the refrigerant charge amounts in optimum summer and winter conditions, and made calculations to determine the annual optimum charge of the system [17]. Jadhav et al., In their study, a thermodynamic analysis was carried out for a system with CO_2 refrigerant using straight, spiral, and spiral capillary tubes. It is said that the reduction in mass in the helical capillary tube is less than that in the straight capillary tube. They show that there is a mass reduction in the spiral tube compared to the straight capillary tube. calculated the reduction in mass in a spiral tube compared to a helical capillary tube. The reduction in length in a straight capillary tube and a spiral tube gave the percentage changes presented the relationships between reductions in length in a spiral tube compared to a helical capillary tube [18]. Freegah et al., In their study, they investigated the effect of capillary tube diameter and mass flow rate of the refrigerant on the physical properties of the refrigerant in the capillary tube. They stated that there was a very good agreement between their experimental and numerical results. The results of the first comparison between the physical properties of the capillary tube and the mass flow are emphasized. An increase in the diameter will result in an increase in the length of the capillary tube, and an increase in the mass flow rate will result in a decrease in the length [19]. El Achkar et al., In their work, they designed a system for the analysis of a cooling system working with conventional throttling and capillary injection. They stated that the cooling system is more stable in capillary injection mode, which will reduce the steady-state time and increase the COP value [20]. The purpose of the study is to replace an expansion valve with a capillary tube. It is to reveal how the capillary tubes of different lengths change in the thermodynamic side of the system at constant gas refrigerant pressure.

2. Transcritical Cycle

The refrigerants used today have high critical temperatures, but some of the fluids we use frequently (such as R404A 72°C, R410A 72.13°C and R407C 86.74°C) are higher than that of Carbon Dioxide (R744

31°C). For example, the critical temperature for R134A is 101.1°C, which means that condensation, as well as heat removal from the system, can be at 101.1°C. This temperature is higher than that required for heat dissipation to the atmosphere for almost any refrigeration application. For R744 it means that condensation heat removal can only occur at temperatures up to 31°C. This temperature is much lower than required to evacuate heat to the atmosphere for many refrigeration applications [22].

In conventional heat pump cycles, the heat rejection occurs below the critical point. Cycles that occur below the critical point are called subcritical cycles. It is provided with a condenser in subcritical systems. Due to the critical temperature of CO_2, the heat dissipation process takes place above the critical point. CO_2 performs the heating process above the critical point with a gas cooler. Since the evaporation process takes place below the critical point and the condensation process takes place above the critical point, these cycles are called transcritical cycles. Subcritical and transcritical cycles are shown in Figure 1 [21].

3. Assumptions and Equations

Since the second law of thermodynamics examines the quality of energy as well as its quantity, it is known that it generally gives reliable results in energy analysis. The increase in the efficient use of energy has been made possible by the transition to the concept of exergy. The main purpose of exergy analysis is to use energy in the most efficient way and to see where and how losses occur on system components.

The following assumptions were taken into account while making the thermodynamic analysis of the system.

•Pure substance is used in the system.

•The compression in the compressor is adiabatic.

•The pipes between the system components are well insulated and the internal and external loss of heat transfer from the evaporator and gas cooler is neglected.

•Pressure drops in system components and on the pipeline and the heat transfer process are also neglected.

•The heat exchangers used as evaporators and gas coolers are counter flowing.

•System performance is assumed to be constant and regular.

•It is assumed that the enthalpy is constant throughout the pressure reduction process in the capillary tube.

•The electrical frequency in the system is 50 Hz and the voltage is 220 volts.

•The kg/minute in the refrigerant flow meter is converted to kg/second.

•The electrical power shown in the watt meter was measured by including the circulation pumps. The circulation pump drew 100 watts of power.

•Gravitational potential energy and kinetic energy are not taken into account.

After obtaining the necessary tools and equipment for the establishment of the test system, high pressure endurance tests of the system were carried out and the detection phase of the leaks in the system was completed. In order to remove the moisture formed in the system, the system was vacuumed and kept under vacuum. Finally, the system was made ready by gradually pressing gas, and it was matched with the software of the Danfoss devices used in the system and installed on the computer used for measurement. The obtained results were recorded on the computer and Store View Desktop for Danfoss branded devices and EES engineering software programs were used for energy and exergy analysis of the system [22].

In Figure 2, the drawing of the experimental system for which thermodynamic analysis was made is given [23].

How efficiently the energy in the system is used is determined by Equation (1) COP (heat pump performance coefficient) calculation, $\dot{Q}_{gc}$, $\dot{W}_c$ and its equation is;

$$COP_{HP} = \frac{\dot{Q}_{gc}}{\dot{W}_c} \quad (1)$$

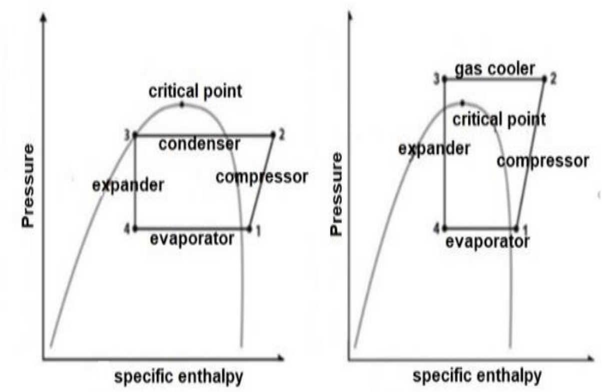

Figure 1. Subcritical and transcritical refrigeration cycle processes [11]

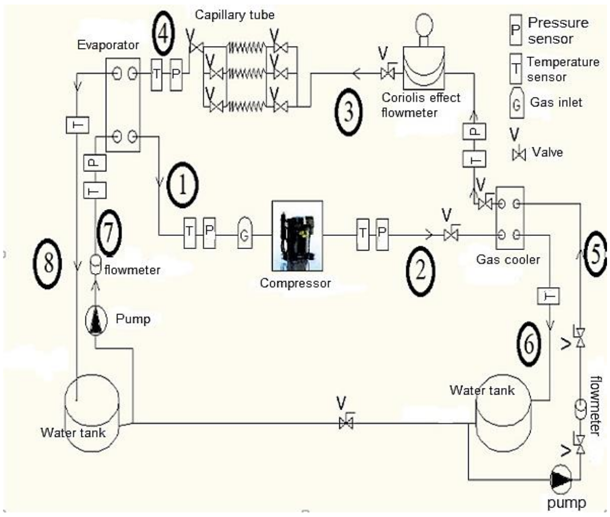

Figure 2. Experiment system with thermodynamic analysis

The temperature values in the system are indicated by T (K), the instantaneous temperature values in the evaporator and the gas cooler in Equation (2) are the temperature values found from the $T_{instantaneous\ temperature}$ ($T_{ins.temp.}$) equation. He calculated separately for carbon dioxide gas and cooling water circulating in the system. It is given in Table 1.

$$T_{ins.temp.} = \frac{h_{in} - h_{out}}{s_{in} - s_{out}} \tag{2}$$

The main purpose of exergy analysis is to use energy in the most efficient way and to see where and how losses occur on system components. Exergy analysis of pure substance;

$$ex=(h-h_0)-T_0(s-s_0) \tag{3}$$

The total exergy destruction for the components is;

$$\dot{E}x = \dot{m} * ex \tag{4}$$

Exergy destruction equation of the total system;

$$\dot{E}x_{total} = \dot{W}_c - \dot{E}x_{ev} \tag{5}$$

Or

$$\dot{E}x_{total} = \dot{E}x_c + \dot{E}x_{gc} + \dot{E}x_v + \dot{E}x_{ev} \tag{6}$$

General exergy efficiency equation;

$$\psi = \frac{\sum useful\ output\ exergy}{\sum input\ exergy} = 1 - \frac{\sum exergy\ loss}{\sum input\ exergy} \tag{7}$$

Heating exergetic efficiency;

$$\psi_{II.} = \frac{\dot{E}x_{gc}}{\dot{W}_c} \tag{8}$$

General exergetic coefficient of performance equation for heating; [24]

$$EPC_{HP} = \frac{\psi_{II.}}{(1-\psi_{II.})} \tag{9}$$

Also, the mass-energy and exergy equations for each part are given in Table 1.

In Figure 3, the closed cycle thermodynamic equations of the system parts are presented in the form of a Table 2. Engineering Equation Solver (EES) program [25] was used for shaping. Variable values of temperature values in their drawings;

- Ambient temperature between 290 K and 305 K
- Evaporator outlet temperature between 290 K and 305 K.
- Gas cooler outlet temperature between 290 K and 305 K.

Table 1. Table representation of system parts in closed cycle thermodynamic equations

Parts	Compressor	Gas cooler	Expansion Valve	Evaporator
Mass Balance	$\dot{m}_1 = \dot{m}_2 = \dot{m}_{CO_2}$	$\dot{m}_2 = \dot{m}_3 = \dot{m}_{CO_2}$ $\dot{m}_5 = \dot{m}_6 = \dot{m}_{H_2O}gs$	$\dot{m}_3 = \dot{m}_4 = \dot{m}_{CO_2}$	$\dot{m}_4 = \dot{m}_1 = \dot{m}_{CO_2}$ $\dot{m}_7 = \dot{m}_8 = \dot{m}_{H_2O}ev$
Energy Balance	$\dot{W}_{C=} \dot{m}_{CO_2}(h_2 - h_1)$	$\dot{Q}_{gc} = \dot{m}_{CO_2}(h_3 - h_2)$ $\dot{Q}_{gs} = \dot{m}_{H_2O}c_{p_{H_2O}}(T_6 - T_5)$	$h_3 = h_4$	$\dot{Q}_e = \dot{m}_{CO_2}(h_3 - h_2)$ $\dot{Q}_e = \dot{m}_{H_2O}c_{p_{H_2O}}(T_7 - T_8)$
Entropy Balance	$\dot{S}_{gen,C} = \dot{m}_{CO_2}(s_2 - s_1)$	$\dot{S}_{gen,gc} = \dot{m}_{CO_2}(s_3 - s_2) + \dot{m}_{H_2O}(s_6 - s_5)$	$\dot{S}_{gen,V} = \dot{m}_{CO_2}(s_4 - s_3)$	$\dot{S}_{gen,ev} = \dot{m}_{H_2O}(s_7 - s_8) + \dot{m}_{CO_2}(s_1 - s_4)$
Exergy Balance	$\dot{E}x_{D,C} = \dot{m}_{CO_2}(ex_1 - ex_2) + \dot{W}_K$	$\dot{E}x_{D,gc} = \dot{m}_{CO_2}(ex_2 - ex_3) + \dot{m}_{H_2O}(ex_5 - ex_6)$	$\dot{E}x_{D,V} = \dot{m}_{CO_2}(ex_3 - ex_4)$	$\dot{E}x_{D,ev} = \dot{m}_{CO_2}(ex_4 - ex_1) + \dot{m}_{H_2O}(ex_7 - ex_8)$
Incoming Exergy	$ex_1 = (h_1 - h_0) - T_0(s_1 - s_0)$	$ex_2 = (h_2 - h_0) - T_0(s_2 - s_0)$ $ex_5 = (h_5 - h_0) - T_0(s_5 - s_0)$	$ex_3 = (h_3 - h_0) - T_0(s_3 - s_0)$	$ex_4 = (h_4 - h_0) - T_0(s_4 - s_0)$ $ex_7 = (h_7 - h_0) - T_0(s_7 - s_0)$
Outgoing Exergy	$ex_2 = (h_2 - h_0) - T_0(s_2 - s_0)$	$ex_3 = (h_3 - h_0) - T_0(s_3 - s_0)$; $ex_6 = (h_6 - h_0) - T_0(s_6 - s_0)$	$ex_4 = (h_4 - h_0) - T_0(s_4 - s_0)$	$ex_1 = (h_1 - h_0) - T_0(s_1 - s_0)$; $ex_8 = (h_8 - h_0) - T_0(s_8 - s_0)$
Exergy Efficiency	$\psi_c = \frac{\dot{m}_{CO_2}(ex_2 - ex_1)}{\dot{W}_C}$	$\psi_{gc} = \frac{\dot{m}_{H_2O}(ex_6 - ex_5)}{\dot{m}_{CO_2}(ex_2 - ex_3)}$	$\psi_v = \frac{ex_4}{ex_3}$	$\psi_{ev} = \frac{\dot{m}_{CO_2}(ex_1 - ex_4)}{\dot{m}_{H_2O}(ex_8 - ex_7)}$

4. Results and Discussion

For the first test part in Table 2 below, a capillary tube with a length of 2.40 m and a diameter of 2.00 mm was used. For the second part, the thermodynamic data obtained in the systems with a capillary tube with a length of 1.20 m and a diameter of 2.00 mm are presented. Actual powers are given in the table after deducting the compressor powers and the powers drawn by the circulation pumps (100W).

Figure 3. Pressure, mass CO2 flow and total power consumption in the test system (ISUBÜ CO2 Laboratory)

Table 2. Thermodynamic comparison in the first and second experiments

1. Experiment		2. Experiment	
Compressor power (kW)	0.719	Compressor power (kW)	0.707
Gas cooler pressure (bar)	100.0	Gas cooler pressure (bar)	100.0
Evaporator pressure (bar)	48.3	Evaporator pressure (bar)	50.8
COP_{HP}	3.72	COP_{HP}	4.01
EPC_{HP}	0.435	EPC_{HP}	0.484
Ex_{total}	0.572	Ex_{total}	0.537
Ψ_c	0.556	Ψ_c	0.577
Ψ_v	0.965	Ψ_v	0.967
Ψ_{gc}	0.509	Ψ_{gc}	0.613
Ψ_{ev}	0.398	Ψ_{ev}	0.353
$\Psi_{II\ HP}$	0.303	$\Psi_{II\ HP}$	0.326
m_{CO2} (Kg/sec)	0.01227	m_{CO2} (Kg/sec)	0.0135
$\dot{m}_{gc}$ H_2O (Kg/sec)	0.0263	$\dot{m}_{gc}$ H_2O (Kg/sec)	0.0263
$\dot{m}_{ev}$ H_2O (Kg/sec)	0.0372	$\dot{m}_{ev}$ H_2O (Kg/sec)	0.0372
T_{gcCO2} (K)	325.9	T_{gcCO2} (K)	325.5
T_{evCO2} (K)	286.1	T_{evCO2} (K)	288.1

In Table 2, a thermodynamic comparison of capillary tubes with the same diameter but different lengths has been made, and the change in some cases is presented at levels of thousandths. The shortening of the capillary tube brought an increase of 7.2% to the COP_{HP} value in the system, and the mass flow rate of the refrigerant increased by 9.1%.

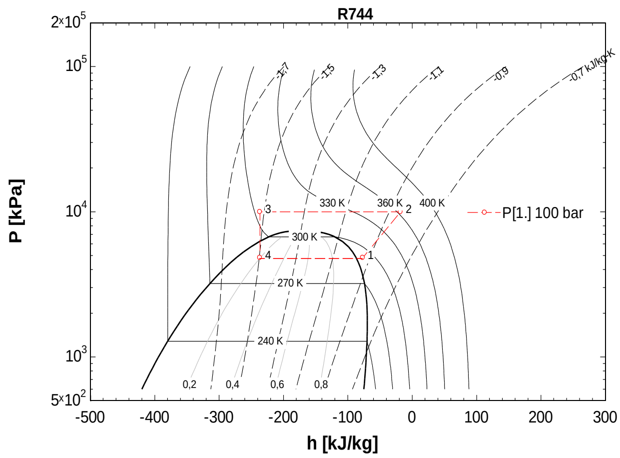

Figure 4. The change of the ln P-h graph of the transcritical cycle with CO_2 is given (1. Capillary)

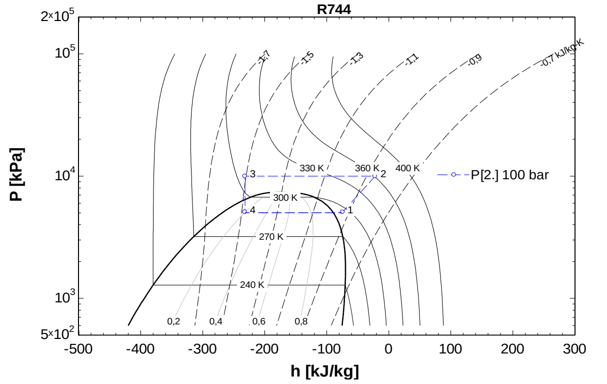

Figure 5. The variation of the Ln P-h graph of the transcritical cycle with CO_2 is given (2. Capillary)

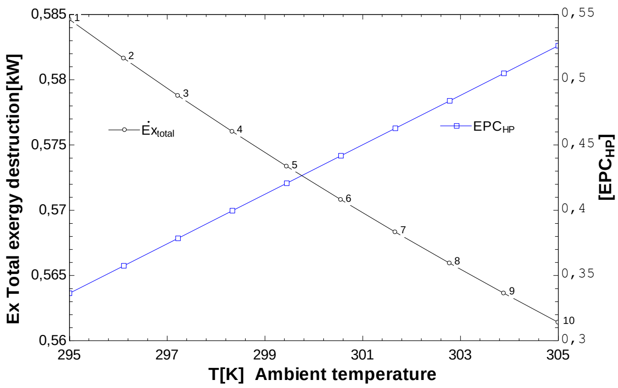

Figure 6. Change of 1. Capillary Ambient temperature in EPC_{HP} and Total exergy destruction

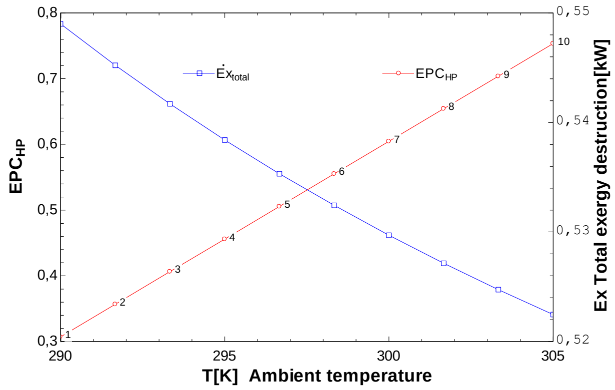

Figure 7. Change of 2. Capillary Ambient temperature in EPC_{HP} and Total exergy destruction

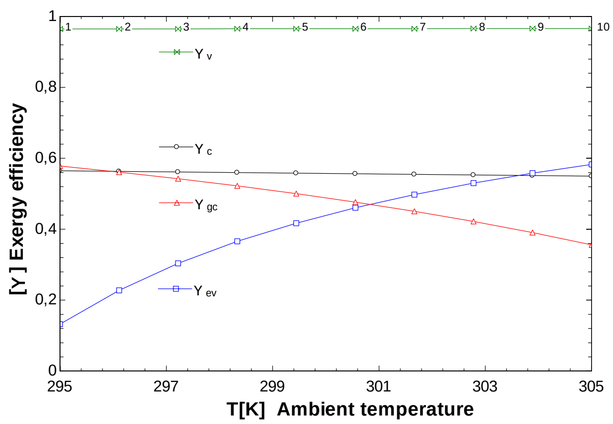

Figure 8. 1st Capillary The effect of change in ambient temperature on exergy efficiency

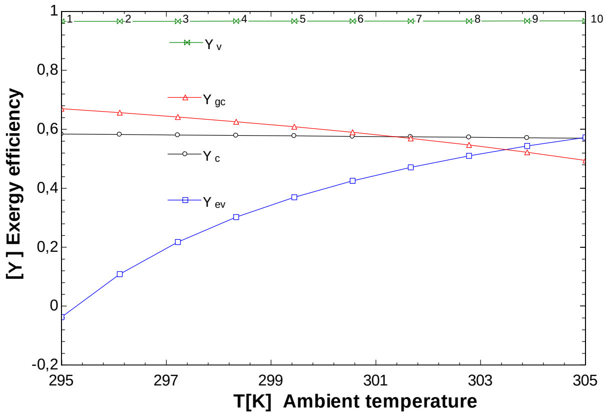

Figure 9. 2nd Capillary the effect of change in ambient temperature on exergy efficiency

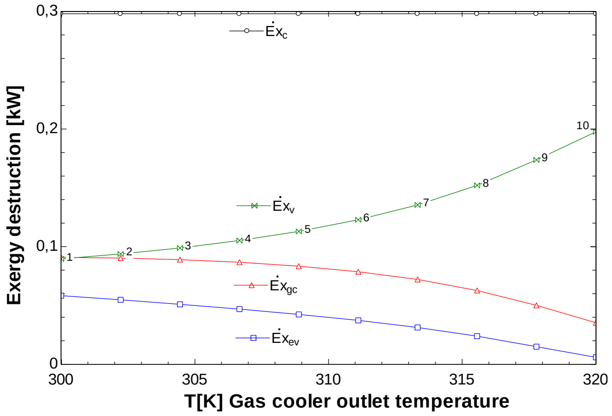

Figure 13. Change in exergy destruction with increase in 2nd capillary gas cooler outlet temperature

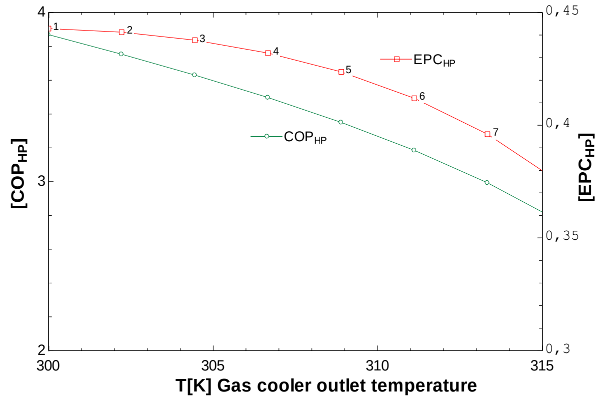

Figure 10. COP_{HP} and EPC_{HP} Heater change with the increase of 1st Capillary Gas cooler outlet temperature

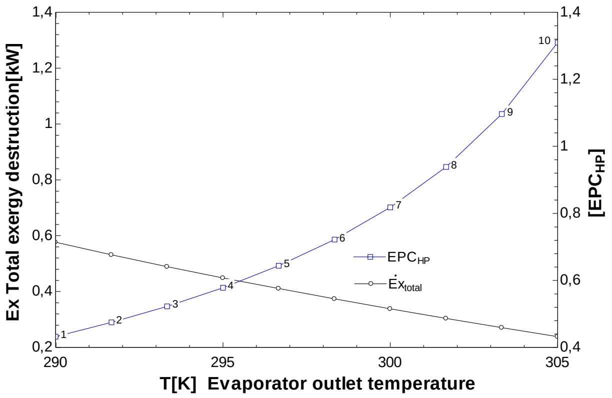

Figure 14. Change of 1st Capillary Evaporator outlet temperature in EPC_{HP} and destruction in Total exergy

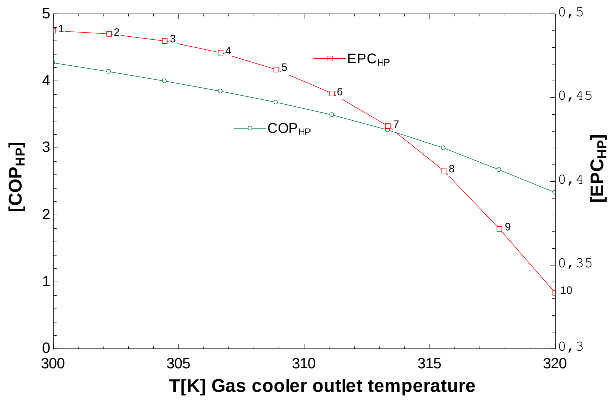

Figure 11. COP_{HP} and EPC_{HP} Heater change with the increase of 2nd Capillary Gas cooler outlet temperature

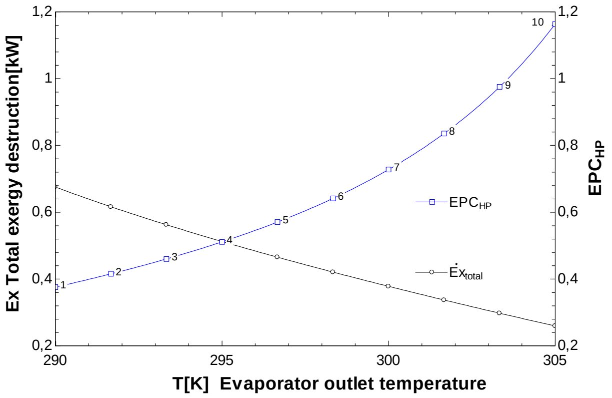

Figure 15. Change of 2nd Capillary Evaporator outlet temperature in EPC_{HP} and destruction in Total exergy

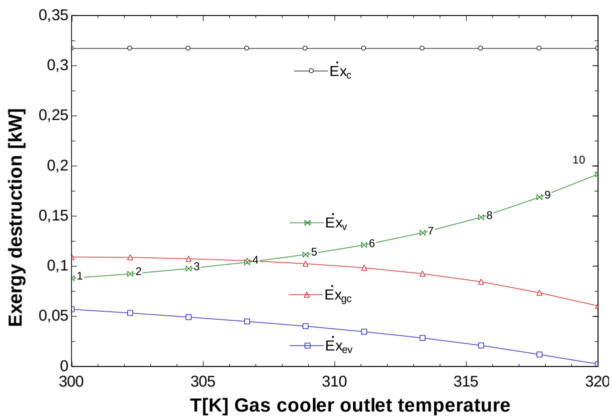

Figure 12. Change in exergy destruction with increase in 1st capillary gas cooler outlet temperature

While the increase in ambient temperature increases the EPC_{HP} value, there is a decrease in the total exergy destruction. In addition (Figure 6-7), while the increase in ambient temperature decreases the exergy efficiency of the gas cooler, it also increases the exergy efficiency of the evaporator (Figure 8-9). Increasing the gas cooler outlet temperature decreases the EPC_{HP} and COP_{HP} coefficients. As a result of the exergy destruction (Figure 10-11), the increase in the gas cooler outlet temperature, the highest compressor, valve, gas cooler and then the evaporator have been reached (Figure 12-13). The increase in the evaporator outlet temperature is seen as a factor that increases the

COP_{HP} and reduces the total exergy destruction (Figure 14-15).

5. Conclusion

In this study, a thermodynamic comparison of the water-to-water heat transfer of a single-stage heat pump system with transcritical CO_2 refrigerant and the capillary tubes of different lengths of a water-to-water cooling heat pump system operating at different operating pressures has been made. In the system, capillary pipes with a diameter of 2.00 mm and two different lengths are used. It is aimed to create the same gas cooler pressures with the two capillary pipes and to keep the cooling waters passing through the gas cooler and the evaporator at the same mass flow rate. The capillary tube in the first experiment is 2.40 m long, and 1.20 m in the second experiment. Thermodynamic analysis and comparison of the system were made. In the system with short capillary pipe (Experiment 2), it was observed that the COP_{HP} value was 7.2% higher, the mass flow rate of the refrigerant increased by 9.1% under the conditions of the short capillary pipe (according to the 1st Experiment), and on the other hand (Experiment 2) In the short capillary tube system, 1.8% less electrical power is used from the compressor power. In terms of exergetics, Exergetic coefficient of performance (EPC_{HP}) is 10%, compressor exergy efficiency (Ψc) is 3.6%, gas cooler exergy efficiency (Ψgc) is 16.9%, capillary tube exergy efficiency (Ψv) is 10%. There is a 7% increase in 0.2 and second law heating efficiency ($\Psi_{II.HP}$). In the latter case, the total exergy destruction increased by 6.5% and the evaporator exergy efficiency (Ψev) increased by 12.7%.

General expressions that do not change for both systems;

• While the increase in ambient temperature increases the EPC_{HP} value, there is a decrease in the total exergy destruction. In addition, while the increase in ambient temperature decreases the exergy efficiency of the gas cooler, it also increases the exergy efficiency of the evaporator.

• Increasing the gas cooler outlet temperature decreases the EPC_{HP} and COP_{HP} coefficients. As a result of the exergy destruction, the increase in the gas cooler outlet temperature, the highest compressor, valve, gas cooler and then the evaporator have been reached.

• The increase in the evaporator outlet temperature is seen as a factor that increases the COP_{HP} and reduces the total exergy destruction.

There are many comments on this subject in the literature. For example, Agrawal et al., reported that it provides COP_{HP} with a shorter capillary tube length. The indicators in this study were consistent with the literature [8].

Transcritical CO_2 heat pump systems are systems that can easily release heat at high pressure. These systems are extremely important in determining the optimum operating conditions, determining the capillary tube sizes and determining the evaporator temperature. Transcritical heat pumps require high pressure control to minimize power consumption. This control is a function of the optimal pressure, the pressure (or temperature) of the evaporating fluid, and the CO_2 gas cooler outlet temperature [10]. With the change in capillary tube length, the evaporator temperatures will be determined, as a result, determining the optimum working intervals will help the users to determine the working conditions.

Declaration

The authors declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article. The authors also declared that this article is original, was prepared in accordance with international publication and research ethics, and ethical committee permission or any special permission is not required.

Author Contributions

A. Elbir conducted an experiment, C. H. Bayrakçı provided consultancy, A. E. Özgür provided the installation of the system, Ö. Deniz developed the measuring system. A. Elbir proofread the manuscript.

Nomenclature

R744	: Carbon dioxide (CO_2)
COP	: Coefficient of Performance
EPC	: Exergetic Coefficient Of Performance
EEV	: electronic expansion valve
ΨII	: Second law exergy efficiency
T_0	: Ambient temperature (299,2K)
$\dot{m}$	: Mass flow rate (kg/s)
$\dot{E}x_{total}$	: Total exergy destruction
s	: specific entropy
h	: Specific enthalpy
K	: temperature (Kelvin)
mm	: millimeters
HP	: heat pump
c	: compressor
ev	: evaporator
gen.	: generation
Sec	: second
P	: bar
Q	: heat (kW)
m	: meters
Ψ	: exergy efficiency
v	: valve
gc	: gas cooler
D	: destruction

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