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Energy Analysis of Two-Stage Cascade Refrigeration System Using R-1234yf and R-404a Refrigerant Couple

Ali Kilicarslan[1]*, Tahir Kaplan[1]

[1] Hitit University, Faculty of Engineering and Natural Sciences, Department of Mechanical Engineering, 19030 Çevre Yolu, Çorum, Türkiye

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ABSTRACT

Vapor compression cascade refrigeration systems consist of a low-temperature refrigeration cycle (LTC) and a high-temperature refrigeration cycle (HTC). Two-stage vapor-compression cascade refrigeration cycles (CRCs) are commonly used in low-temperature applications. In this study, R-404a is used in the LTC, and R-1234yf is used in the HTC. The effects of the outdoor air temperature on the suction and discharge pressures of HTC and LTC, powers drawn by the compressors of HTC and LTC, heat rejection capacity of the condenser of CRC, heat absorption capacity of the evaporator of CRC, coefficient of performance (COP) of CRC, the mass flow rates of refrigerants circulating through HTC and LTC were experimentally investigated in terms of the 1st Law of thermodynamics. The results show that increasing the outlet air temperature from 28 °C to 43.5 °C results in significant changes in system parameters. As the outdoor air temperature increased, the suction and discharge pressures of HTC and LTC, the powers drawn by the compressors of HTC and LTC, the heat rejection capacity of the condenser of CRC, the mass flow rates of refrigerants circulating through HTC and LTC increased. In contrast, the heat absorption capacity of the evaporator of CRC and the COP of CRC decreased. The largest increase was observed in the investigated parameters, with HTC discharge and suction pressures increasing by 41.94% and 31.11%, respectively.

KEYWORDS Cascade refrigeration, Two-stage, Energy analysis, Outdoor air temperature, R1234yf, R404a

1. Introduction

The inability of single-stage vapor-compression refrigeration systems to meet low-temperature requirements economically and efficiently has accelerated the development of two-stage vapor-compression cascade refrigeration systems as an alternative. Cascade refrigeration systems are widely preferred, especially for low-temperature applications (−50 °C to −100 °C) (Yılmaz *et al.* 2020). The main application areas of these systems include blood and medical product storage, gas liquefaction, and heat treatment (hardening) of special alloy steels (Gupta 1985). A two-stage cascade refrigeration system consists of two compressors, a condenser, an evaporator, two expansion valves, and a cascade heat exchanger connecting the cycles. This heat exchanger serves as the condenser for the low-temperature circuit (LTC) and simultaneously functions as the evaporator for the high-temperature circuit (HTC), facilitating heat transfer between the two cycles. There are studies related to the vapor compression cascade refrigeration cycles in the literature. The most relevant ones are summarized below.

In a theoretical study, a two-stage cascade system using the R1150/R717 refrigerant pair was investigated as an alternative to a three-stage system with the R1150/R41/R717 refrigerant group, and it was determined that the total compressor power of the two-stage system was 1.45–1.61 kW lower. The coefficient of performance (COP) was 7.0% to 11.7% higher (Sun and Wang 2022a). In a study where thermodynamic simulations of six different cycle architectures (single-stage with/without heat

* Corresponding author. Email address: alikilicarslan@hitit.edu.tr (Ali Kilicarslan)

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exchanger, two-stage cascade with vapor and liquid injection, parallel compression cascade with/without economizer) were performed at very low temperatures, R290 was used in the upper cycle and R170 in the lower cycle. In the study, it was observed that the optimum intermediate-stage cascade temperature ranged from 20 °C to 37 °C, depending on the cycle type, and that this directly affected energy performance. In addition, it has been determined that the energy performance of parallel compression cascade cycles is higher than that of single-stage cycles but 20% to 30% lower than that of stage compression cascade cycles (Udroiu *et al.* 2022).

In an experimental study in which R452a was used in the upper circuit and a conventional R23 or CO2 sublimation cycle in the lower-temperature cycle, it was reported that the preference for CO2 sublimation in the lower-temperature cycle positively affected system performance (Xu *et al.* 2021). In a study comparing the energy and exergy parameters of different refrigerant pairs (R152a–R23, R290–R23, R507–R23, R234a–R23, R717–R23, and R404a–R23); it was observed that the COP value increased in all refrigerant pairs depending on the increasing polytropic efficiency and evaporator temperature, while the exergy destruction due to irreversibilities decreased. Among the alternatives examined, the R717–R23 pair yielded the best results, with the maximum COP and minimum exergy destruction rate (Kilicarslan and Hosoz 2010).

Symmetric cascade systems (R125/R125, R1270/R1270, R143a/R143a, R404a/R404a, R407a/R407a, R410a/R410a, and R507a/R507a) using the same fluid in the upper and lower circuits were compared with the conventional R23/R404a cascade system. Performance improvement rates were calculated as 19.48%, 8.08%, 21.54%, 14.33%, 11.00%, 15.64%, and 12.11%, respectively (Sun *et al.* 2022b). In an experimental study, low-GWP R513a, R1234yf, and R436a were used as alternatives to R134a in the upper circuit, and R744 (CO2) was used in the lower circuit. In contrast, the use of R436a resulted in a 3.1% increase in system performance; the use of R1234yf and R513a decreased performance by 3.7% and 4.4%, respectively (Blanco Ojeda *et al.* 2022).

In another theoretical study conducted at different evaporation and condensation temperatures, the fluids R32, R1234yf, R1234ze, R161, R1270, R290, and R717 in the temperature cycle and R23, R41, and R170 in the low-temperature cycle were examined in terms of energy and exergy. The analysis results showed that the combination of R161 in the upper cycle and R41 and R170 in the lower cycle was the most suitable option (Sun *et al.* 2019).

In a modeling study comparing traditional R404a and R134a refrigerants with low GWP natural alternatives (R717, R744, R290) and synthetic refrigerants (R1234ze and R152a) in terms of energy and TEWI (Total Equivalent Environmental Impact) parameters, it was predicted that future cooling solutions will rely on low GWP refrigerants. However, it was emphasized that these refrigerants pose risks, including toxicity and flammability, and that the most stable and promising refrigerant for the subcycle of central cascade systems is CO2 (R744) (Llopis *et al.* 2015). The refrigerant pairs R41-R161, R41-R1234yf, R41-R1234ze, R744-R161, R744-R1234yf, and R744-R1234ze. In a study in which energy, exergy, economy, and environment (4E) analyses were performed, the maximum COP (2.09) and the highest exergy efficiency (35.32%) were obtained at condensation temperatures of 40 °C and evaporation temperatures of −30 °C. In the study, R41-R161 and R41-R1234ze stood out as optimum fluid pairs (Deymi-Dashtebayaz *et al.* 2021).

When the literature summarized above is examined, it is evident that most studies on two-stage vapor-compression cascade refrigeration systems are theoretical and simulation-based. The limited number of experimental studies stems from the difficulties in setting up test setups, high costs, and operational risks. In particular, when research on the use of R-1234yf refrigerant, a current and environmentally friendly alternative with low global warming potential (GWP), in cascade systems is reviewed, it is observed that existing studies remain almost entirely at the theoretical level, and parametric evaluations are generally limited only to evaporator and condenser temperatures. This situation indicates a significant gap in the literature regarding the experimental validation of the performance of new-generation refrigerants under real operating conditions.

This experimental study differs from studies in the literature: the environmentally friendly refrigerant R-1234yf will be used in the upper circuit of a two-stage vapor-compression cascade refrigeration system, and the refrigerant R-404a, which performs well at low temperatures, will be used in the lower circuit. Another novelty of this study is that the effect of outside air temperature up to 45 °C is investigated in a two-stage vapor-compression cascade refrigeration system. The effect of outside air temperature on the inlet and outlet pressures of the compressors, the power consumed by the compressors, the flow rates of the refrigerant filling the system, the evaporator and condenser capacity, and the cooling efficiency will be investigated experimentally. The results will be obtained by developing a computer code using Engineering Equation Solver, EES (Klein 2020).

2. Materials and Method

Vapor compression cascade refrigeration systems are used to maintain media, materials, or fluids at very low temperatures. Depending on the desired low temperature, vapor-compression cascade refrigeration systems can have 2, 3, or more cascade stages. The method used in this study will be based on the two-stage vapor compression cascade refrigeration system shown in Figure 1 below. The method consists of three stages. In the first stage, a thermodynamic model incorporating energy analysis will be developed. In the second stage, a computer program based on the thermodynamic model and accounting for the system's operating points will be developed using the Engineering Equation Solver (EES). In the third stage, experimental studies will be conducted.

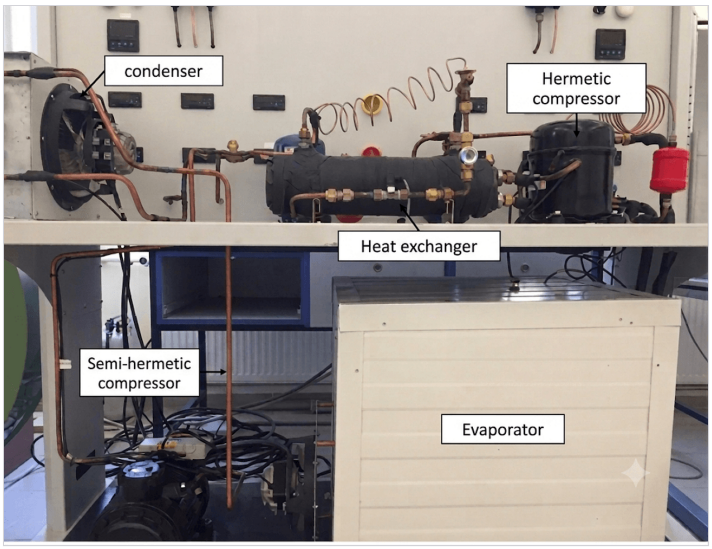

Figure 1. Experimental setup of a two-stage vapor compression cascade refrigeration system.

The condenser of the cascade refrigeration system will be placed inside a duct through which air is passed, with electric heaters located there. After the high-temperature cycle (HTC) of the cascade system is vacuumed, R1234yf refrigerant will be charged, and after the low-temperature cycle (LTC) is vacuumed, R-404a refrigerant will be charged. The cascade refrigeration system will be started and allowed to reach steady-state conditions. The heaters installed in the condenser will be activated to maintain air temperatures between 25 °C and 35 °C. Temperature and pressure measurements will be taken at the operating points of the cascade refrigeration system under varying air temperatures, and mass flow rate measurements will be taken in the liquid line. These measurements will be repeated at least three times under the same conditions. Thermophysical properties of the selected refrigerants for HTC and LTC are given in Table 1.

Table 1. Thermo-physical properties of the selected refrigerants adapted from (Klein 2020; Wikipedia no date).

Refrigerant	ODP	GWP	Safety group	Boiling point (°C)	Critical Temperature (°C)	Critical Pressure (kPa)
R-1234yf	0	<1	A2L	−29.48	94.7	3382
R-404a	0	3922	A1	−46.22	72.12	3735

In this study, a two-stage vapor compression cascade refrigeration system will be experimentally investigated. Figure 2 schematically shows the experimental setup of the two-stage vapor compression cascade refrigeration system. This refrigeration system consists of two compressors, two expansion valves, a condenser, an evaporator, and a heat exchanger. As seen in Figure 1, the cascade refrigeration system consists of a low-temperature cycle and a high-temperature cycle.

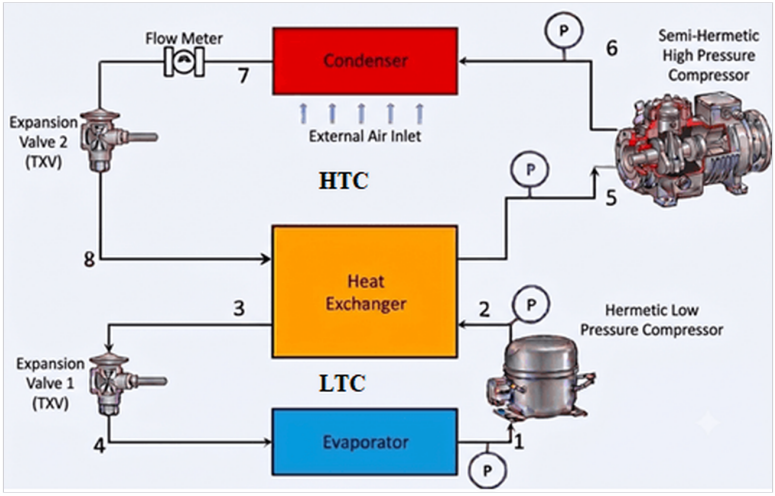

Figure 2. Schematic representation of the experimental setup of a two-stage vapor compression cascade refrigeration cycle.

3. Thermodynamic Analysis

In a two-stage vapor compression cascade refrigeration system, it is assumed that the thermodynamic properties of the fluid do not change with time at any point in the system (steady regime), that changes in kinetic and potential energies between the inlet and outlet of the elements forming the cascade system are neglected, that pressure losses are neglected, and that the compressors are adiabatic. Applying the assumptions stated in the paragraph above, the energy, entropy, and second law equations for the elements forming the cascade refrigeration system are written as follows,

$$\text{Compressor (LTC):} \quad \dot{W}_{LTC} = \dot{m}_{LTC}(h_1 - h_2) \tag{1}$$

$$\text{Compressor (HTC):} \quad \dot{W}_{HTC} = \dot{m}_{HTC}(h_6 - h_5) \tag{2}$$

$$\text{Condenser (CRC):} \quad \dot{Q}_c = \dot{m}_{HTC}(h_7 - h_6) \tag{3}$$

$$\text{Evaporator (CRC):} \quad \dot{Q}_e = \dot{m}_{LTC}(h_1 - h_4) \tag{4}$$

$$\text{Heat exchanger (CRC):} \quad \dot{m}_1(h_2 - h_3) = \dot{m}_2(h_5 - h_8) \tag{5}$$

$$\text{Expansion valve (LTC):} \quad h_3 = h_4 \tag{6}$$

$$\text{Expansion valve (HTC):} \quad h_7 = h_8 \tag{7}$$

$$\text{COP} = \frac{\dot{Q}_e}{\dot{W}_{LTC} + \dot{W}_{HTC}} \tag{8}$$

Where:

- h = the specific enthalpy at the operating points (1, 2, ..., 8) (kJ/kg)
- $\dot{m}_{LTC}$ = the mass flow rates in the low-temperature in LTC (kg/s)
- $\dot{m}_{HTC}$ = the mass flow rates in the high-temperature in HTC (kg/s)
- $\dot{W}_{LTC}$ = the power consumed by the compressor in the low-temperature (kW)
- $\dot{W}_{HTC}$ = the power consumed by the compressor in the high-temperature (kW)
- $\dot{Q}e$ = the refrigeration capacity of the two-stage vapor compression cascade refrigeration (CRC) (kW)
- $\dot{Q}c$ = the amount of heat released to the environment by the CRC condenser (kW)
- COP = the coefficient of performance of the CRC.

4. Uncertainty Analysis

Holman (1984) presents uncertainty analysis in detail. Assume that the experimental result G is a function of n measured variables $x_1, x_2, x_3, \ldots, x_n$. Therefore,

$$G = G(x_1,\ x_2,\ x_3,\ \ldots,\ x_n) \tag{9}$$

Suppose that Y_R is the uncertainty in the experimental result and Y_1, Y_2, Y_3,..., Y_n are the uncertainties in the measured variables. If the uncertainties in the measured variables are all specified with the same probability, the uncertainty in the experimental result with these probabilities can be expressed by (Holman 1984) as

$$Y_R = \left[\left(\frac{\partial G}{\partial x_1} Y_1\right)^2 + \left(\frac{\partial G}{\partial x_2} Y_2\right)^2 + \left(\frac{\partial G}{\partial x_3} Y_3\right)^2 + \cdots + \left(\frac{\partial G}{\partial x_n} Y_n\right)^2\right]^{1/2} \tag{10}$$

The uncertainties of the measuring devices that are used to calculate the uncertainties associated with $\dot{W}_{LTC}$, $\dot{W}_{HTC}$, $\dot{Q}_c$, $\dot{Q}_e$ and COP are ± 1 % for turbine-type mass flow meter, ± 1 % for bourdon-type pressure gauge, and ± 1.5 % for K-type thermocouple. Maximum and minimum uncertainties of the parameters are determined by EES (Klein 2020). Minimum and maximum uncertainties of the parameters are given in Table 2.

Table 2. Minimum, maximum, and uncertainties in this experimental study.

Parameters	Minimum uncertainties	Maximum uncertainties
$\dot{W}_{LTC}$	1.589	2.039
$\dot{W}_{HTC}$	5.058	7.133
$\dot{Q}_c$	8.375	10.79
$\dot{Q}_e$	4.337	5.593
COP	0.02199	0.02905

5. Results and Discussion

Experimentally measured average values were entered into a computer program developed with Engineering Equation Solver (EES) software, and parameters related to the First Law of Thermodynamics were obtained as a function of the outside air temperature passing through the condenser. R404a is used as the refrigerant in LTC, while R1234yf is used in HTC.

In the following section, the results depicting the effects of the outdoor air temperatures ranging from 28 °C to 43.5 °C, on the suction and discharge pressures of HTC and LTC, powers drawn by the compressors of HTC and LTC, heat rejection capacity of the condenser of CRC, heat absorption capacity of the evaporator of CRC, coefficient of performance (COP) of CRC, the mass flow rates of refrigerants circulating through HTC and LTC are presented and discussed in detail.

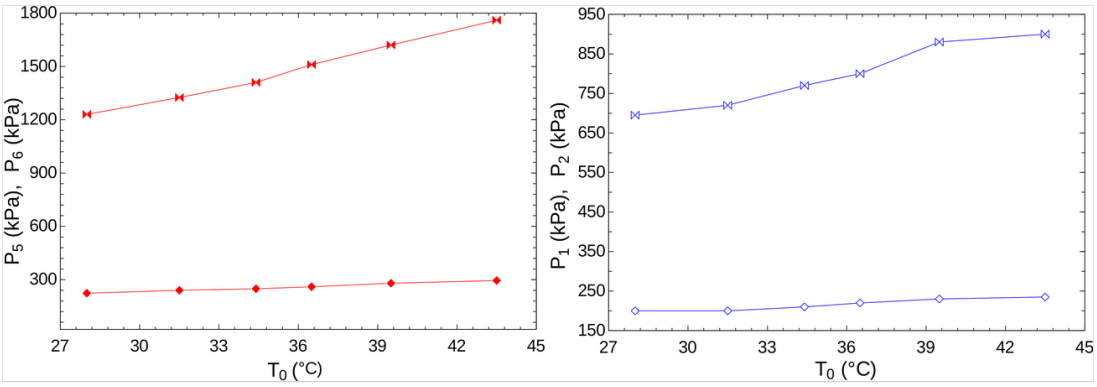

Figure 3. Changes of compressor suction and discharge pressures as a function of outlet air temperature: (a) high temperature cycle (HTC); (b) low temperature cycle (LTC).

Figure 3 depicts the variation of the suction and discharge pressures of the high- and low-temperature cycles of a two-stage vapor compression cascade refrigeration cycle (CRC) with respect to the outlet air temperatures. All pressures in the cycle increase with increasing outlet air temperature. As outdoor air temperature increases, the compressor of an HTC tries to raise the refrigerant's saturation temperature in the condenser (condenser temperature) to facilitate effective heat transfer to the environment. The increase in the condenser temperature is directly proportional to the discharge pressure of the HTC. As shown in Figure 3a, the greatest increase in pressure is observed in the discharge pressure of HTC (P_6). The increase in discharge pressure increases the pressure ratio of the HTC compressor, thereby increasing the suction pressure of the HTC (P_6), as shown in Figure 3a. The discharge pressure of LTC (P_2) is indirectly affected by changes in outlet air temperature. As the suction pressure of HTC increases, the evaporator temperature also increases. In the heat exchanger, the condenser temperature of LTC increases to provide the necessary heat transfer to the evaporator as the HTC evaporator temperature increases. The increase in the condenser temperature of LTC causes the discharge pressure of LTC (P_2) to increase, as the condenser temperature of LTC is directly related to its discharge pressure (P_2), as depicted in Figure 3b. The suction pressure of LTC (P_1) is indirectly affected by changes in outlet air temperature. However, it is

directly affected by changes in the evaporator temperature of the two-stage vapor-compression cascade refrigeration cycle (CRC). As the outlet air temperature increases, the evaporator temperature of CRC increases, thereby increasing the suction pressure of LTC (P_1), as shown in Figure 3b. As the outlet air temperature changes from 28 °C to 43.5 °C, the percent increases in P_6, P_5, P_2, and P_1 are 41.94%, 31.11%, 29.50%, and 17.50%, respectively.

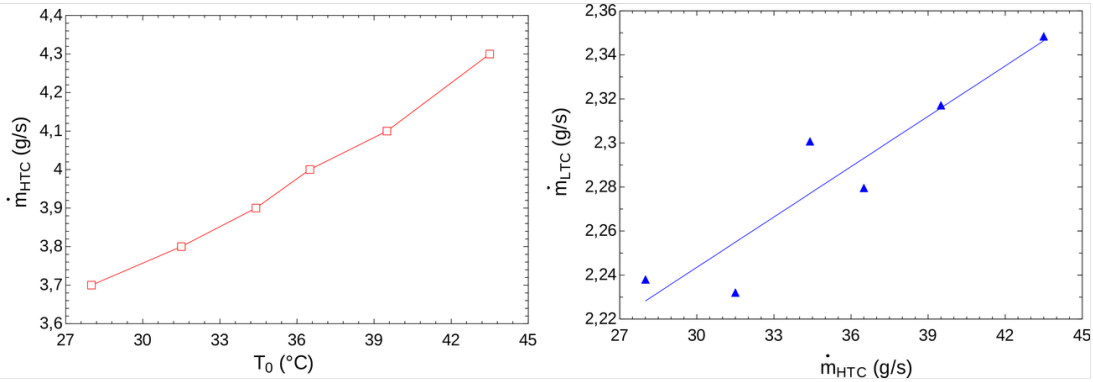

Figure 4. Changes of mass flow rates of HTC and LTC as a function of outlet air temperature: (a) high temperature cycle (HTC); (b) low temperature cycle (LTC).

Figure 4 depicts the variation of the mass flow rates of HTC and LTC with respect to the outlet air temperatures. The mass flow rates of HTC and LTC increase with increasing outlet air temperature. As outdoor air temperature increases, the compressor in HTC consumes more power than normal to reach the CRC condenser temperature and pressure. To provide extra power to the HTC, more refrigerant should be circulated through it. As the outdoor air temperature increases from 28 °C to 43.5 °C, the mass flow rate of the refrigerant flowing through HTC increases from 3.7 g/s to 4.3 g/s, as shown in Figure 4a. To maintain heat transfer balance in the heat exchanger, the refrigerant mass flow rate through LTC should also increase, as depicted in Figure 4b. The mass flow rate of LTC changes from 2.23 g/s to 2.38 g/s. When the percent increases in the mass flow rates are compared, the percent increases for HTC and LTC are 16.22% and 4.91%, respectively. The mass flow rate of refrigerant through HTC is much higher than that through LTC because HTC is directly affected by changes in outdoor air temperature, and the latent heat of vaporization of R-1234yf is lower than that of R-404a at the operating pressures in the experiments.

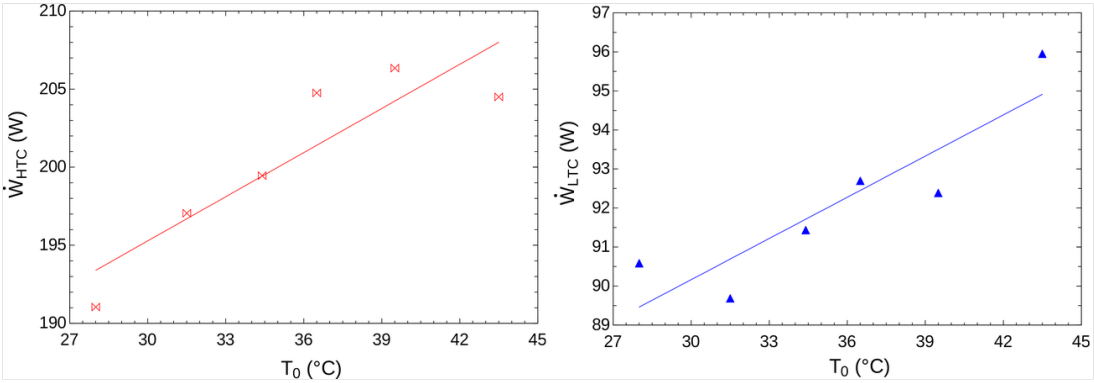

Figure 5. Changes of compressor power consumed by HTC and LTC as a function of outlet air temperature: (a) high temperature cycle (HTC); (b) low temperature cycle (LTC).

Figure 5 shows the variation in power consumption of the compressors for HTC and LTC with respect to outside air temperature. R1234yf is used as a refrigerant in the HTC, and R404a is used as a refrigerant in LTC. As the outside air temperature ranges from 28 °C to 43.5 °C, the compressor power consumed in both cycles increases. When the temperature of the outside air passing through the

condenser increases, the temperature difference between the condenser of the cascade system (also the condenser of the HTC) and the outside air (ambient air) will decrease, and since the condenser of the cascade system cannot release enough heat to the ambient air. To increase the condenser temperature for necessary heat transfer to the environment, the discharge pressure of the HTC will increase, and, as a natural consequence, the compressor power of the HTC will increase, as seen in Figure 5a. The pressure ratio of the HTC also increases as the compressor's power consumption increases. The increase in the pressure ratio of the HTC compressor increases its suction pressure, thereby increasing the discharge pressure of the LTC compressor and enabling effective heat exchange in the heat exchanger. As the discharge pressure of the LTC compressor increases, the compressor draws more power, as shown in Figure 5b. Another physical reason for the increase in the compressor power of HTC and LTC is the increase in the circulating mass flow rate of HTC and LTC. As the outside air temperature changes by 15.5 °C, the power of the compressors of HTC and LTC increases by 7.66 % and 6.03 %, respectively.

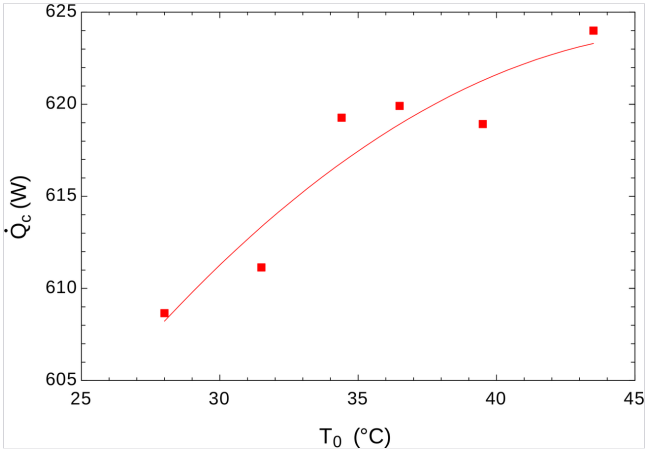

Figure 6. Change of capacity of the condenser of CRC as a function of outlet air temperature.

The variation in the capacity of the condenser in the two-stage vapor compression cascade refrigeration system (CRC) with respect to outside air temperature is shown in Figure 6. As the outside air temperature increases from 28 °C to 43.5 °C, the condenser temperature also increases from 47.62 °C to 63.11 °C. Two parameters affect the condenser capacity in CRC. These are the temperature difference between the condenser temperature and the outside air temperature, and the mass flow rate circulating through the HTC. The temperature difference between the condenser temperature and the outside air temperature is almost constant, around 19.5 °C, at outlet air temperatures between 28 °C and 43.5 °C, but the mass flow rate increases by 16–22% in these temperature ranges, as mentioned before. The most dominant factor affecting the condenser capacity of CRC is the refrigerant mass flow rate in HTC. As shown in Figure 6, the condenser capacity increases from 608.7 W to 624 W. The capacity of the condenser increases 0.987 W for every increase of 1 °C.

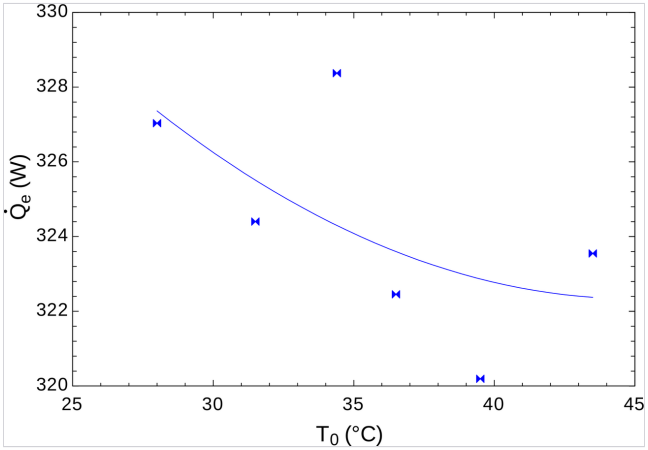

Figure 7. Change of capacity of the evaporator of CRC as a function of outlet air temperature.

Figure 7 shows the variation in the evaporator capacity of the CRC with respect to outside air temperature. As the outside air temperature increases from 28 °C to 43.5 °C, the temperature of the room to be refrigerated increases from −27 °C to −21 °C because the evaporator capacity of the CRC is insufficient to reduce the room temperature to the desired level. In other words, the evaporator capacity decreases as outside air temperature increases. The average decrease in evaporator capacity is around 1%. The evaporator of CRC is indirectly affected by changes in outdoor air temperature.

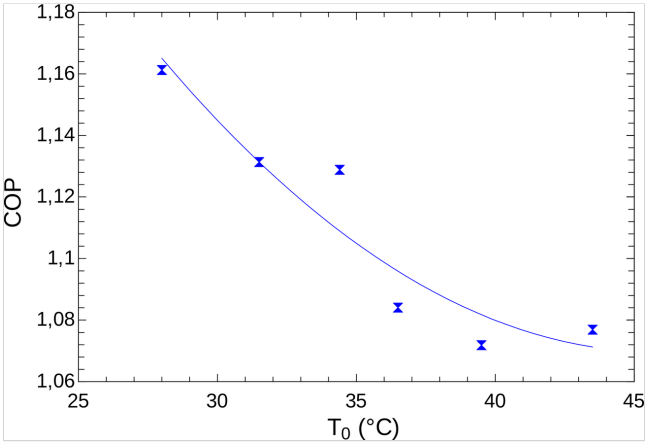

Figure 8. Change of coefficient of performance (COP) of CRC as a function of outlet air temperature.

The variation in the coefficient of performance (COP) of the two-stage vapor compression cascade refrigeration system (CRC) with respect to outdoor air temperature is shown in Figure 8. As the outdoor air temperature increases from 28 °C to 43.5 °C, the COP of CRC decreases. COP is mainly a function of the total power consumed by the compressors of HTC and LTC and the evaporator capacity of CRC. As outdoor air temperatures increase from 28 °C to 43.5 °C, the evaporator capacity decreases while the total power consumed by the compressors of HTC and LTC increases. As a result, the COP of CRC decreases, as shown in Figure 8. The COP of CRC decreases 7.23 % at the outdoor air temperatures between 28 °C and 43.5 °C.

At higher outdoor air temperature, in order to prevent the coefficient of performance (COP) of the two-stage vapor compression cascade refrigeration system (CRC) from decreasing a certain value, the HTC compressor and cascade condenser with higher capacities should be selected to decrease the saturation pressure of the cascade condenser, variable speed fans for the condenser and inverter compressor for HTC should be employed to control the performance of two stage cascade refrigeration system, suitable environment friendly refrigeration couple having low discharge

pressures at higher outdoor air temperature should be selected and intermediate temperature of the CRC should be optimized.

6. Conclusion

The effects of outdoor air temperature on the energy parameters of the two-stage vapor-compression cascade refrigeration cycle (CRC), including the suction and discharge pressures of the compressors, the power consumed by the compressors, the mass flow rate, the evaporator capacity, the heat rejected by the condenser, and the COP were experimentally observed. The main conclusions drawn are as follows;

- At the increasing values of the outlet air temperatures between 28 °C and 43.5 °C, suction and discharge pressures of low temperature and high temperature cycle increased, the maximum increase in the pressure occurred at the discharge of the compressor of HTC as 41.94 %, while the minimum occurred at the suction of the compressor of LTC.
- The power consumption of the compressors in CRC increased with outdoor air temperature. The increase in power is 7.66% for the HTC compressor and 6.03% for the LTC compressor.
- The increase in outdoor air temperature also caused the mass flow rates of the refrigerants circulating through HTC and LTC to increase to meet the increased compressor power. The maximum increase in the mass flow rate occurred in HTC at 16.22 %, while the minimum occurred in LTC at 4.91 %.
- The heat rejection capacity of the condenser increased by 2.5%, while the evaporator capacity decreased by 1%, as the outlet air temperature increased from 28 °C to 43.5 °C.

Declarations

Author contributions. Ali Kilicarslan: Conceptualization, Methodology, Investigation, Formal analysis, Writing (original draft), Writing (review and editing), Supervision. Tahir Kaplan: Writing (original draft), Investigation.

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Conflicts of interest. The authors declare no competing interests.

Data availability. Not applicable. This study did not generate or analyse any datasets.

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