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ENERGY, ENVIRONMENTAL, AND EXERGOECONOMIC (3E) ANALYSIS OF TRANSCRITICAL CO_2 BOOSTER AND PARALLEL COMPRESSION SUPERMARKET REFRIGERATION CYCLES IN CLIMATE ZONES OF TÜRKİYE

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Highlights

- Two CO_2 supermarket refrigeration cycle configurations were investigated.
- Bin-hour data of 11 provinces in Türkiye were derived using hourly temperatures.
- Up to 5.6% energy consumption reduction was obtained using parallel compression.
- The contribution of CO_2 to direct emissions is negligible.
- Parallel compression cycle has up to 18% less unit product cost.

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ABSTRACT: Legal restrictions on high-GWP refrigerants lead to the widespread use of carbon dioxide in commercial refrigeration, where there is a high energy consumption. Although CO_2 has many benefits, its lower critical temperature and higher operation pressure compared to other refrigerants lead to performance reduction. For this reason, studies have been conducted by researchers for performance enhancement. This paper presents energy, environmental impact, and exergoeconomic (3E) analysis of transcritical CO_2 booster and parallel compression supermarket refrigeration cycles based on meteorological data of 11 provinces in Türkiye as samples of different climatic regions. Parallel compression cycle achieved up to 18.4% higher coefficient of performance than booster cycle between the investigated ambient temperatures. Up to 5.6% annual energy consumption and environmental impact reduction were obtained using parallel compression. Unit product costs of the parallel compression cycles were calculated between 8.2% and 18% lower than booster cycle in investigated provinces. Developing energy-efficient systems that use environmentally friendly refrigerants will contribute to a sustainable future.

Keywords: *Bin-hour, Carbon dioxide, Environmental impact, Exergoeconomic analysis, Supermarket*

1. INTRODUCTION

Global warming is a serious issue around the world and therefore, legal authorities are taking action to avoid it. Commercial refrigerators contribute to 25-60% of the electricity used in stores and the highest CO_2 emission of all refrigerators [1], [2]. Because the European Commission has prohibited using refrigerants with global warming potential (GWP) values higher than 150 for newly installed central refrigeration systems starting from 2022 [3], using low-GWP refrigerants has gained importance. Carbon dioxide (CO_2) is a non-toxic, non-flammable, and cheap refrigerant with a GWP value of only 1 [4]. However, CO_2 has a lower critical temperature and a higher operation pressure compared to other common refrigerants. This case leads to lower performance. There have been improvements made to increase the performance of transcritical CO_2 refrigeration cycles including parallel compression, adiabatic gas cooling, subcooling, and ejector expansion [5]. There are also studies using CO_2 in cascade and secondary loop cycles in the literature [6], [7].

Sharma et al. [8] conducted a theoretical comparison of various supermarket refrigeration systems including transcritical CO_2 and cascade/secondary loop for various climate zones in the US resulting in that transcritical CO_2 system with parallel compressor is the most efficient in northern and central regions among investigated systems. Fritschi et al. [9] determined that a low evaporation temperature and high gas cooler outlet temperature have a positive effect on the parallel compression circuit. Chesi et al. [10] found that an ideal cycle with parallel compressor may improve the coefficient of performance (COP) and cooling capacity by over 30% and 65%, respectively, provided that there is no pressure loss along piping, superheat is constant, and liquid-vapor separation is perfect in the flash tank.

The use of transcritical CO_2 systems in commercial refrigeration is spreading worldwide. International supermarket chain Carrefour has installed two CO_2 systems in İstanbul. The one installed

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in Bahçelievler is an R404A/CO_2 cascade system while the other one in Kurtköy is transcritical CO_2 booster system with adiabatic gas cooler [11], [12]. The company continues to install transcritical CO_2 refrigeration systems around the world such as Poland, Romania, Italy, and Belgium [13]–[15]. UK-based grocery retailer Tesco has installed transcritical CO_2 systems at about 1000 stores, with plans to be HFC-free by 2035 [16]. Longo Brothers Fruit Markets in Canada uses transcritical CO_2 systems in 44% of its grocery stores, with plans to install CO_2 systems in all new stores [17]. It is reported that 6960 locations in Japan (including 300 large retail stores, 6330 convenience stores, and 330 industrial facilities) use transcritical CO_2 refrigeration systems as of the end of December 2022 [18].

This paper presents energy, environmental impact, and exergoeconomic analysis of transcritical booster and parallel compression CO_2 supermarket refrigeration cycles based on meteorological data of 11 provinces in Türkiye.

2. MATHEMATICAL MODELS OF THE CYCLES

Transcritical CO_2 booster refrigeration cycle (BRC) is a baseline for supermarket refrigeration systems as shown in Figure 1. The cycle consists of two evaporators for frozen and fresh food. No phase change occurs above the critical point in the gas cooler, contrary to the condenser. High-pressure refrigerant coming from the gas cooler is expanded to intermediate pressure in the high-pressure expansion valve (HPXV). In the flash tank, liquid and vapor separation occurs. Vapor is expanded to chiller pressure via the flash-gas-bypass (FGB) valve while liquid is sent to the low-pressure expansion valve (LPXV) and medium-pressure expansion valve (MPXV). Refrigerant gaining heat in the freezer is compressed to chiller pressure via the low-pressure compressor (LPC). Three refrigerant streams are mixed in the suction line of the high-pressure compressor (HPC) and compressed to the gas cooler pressure.

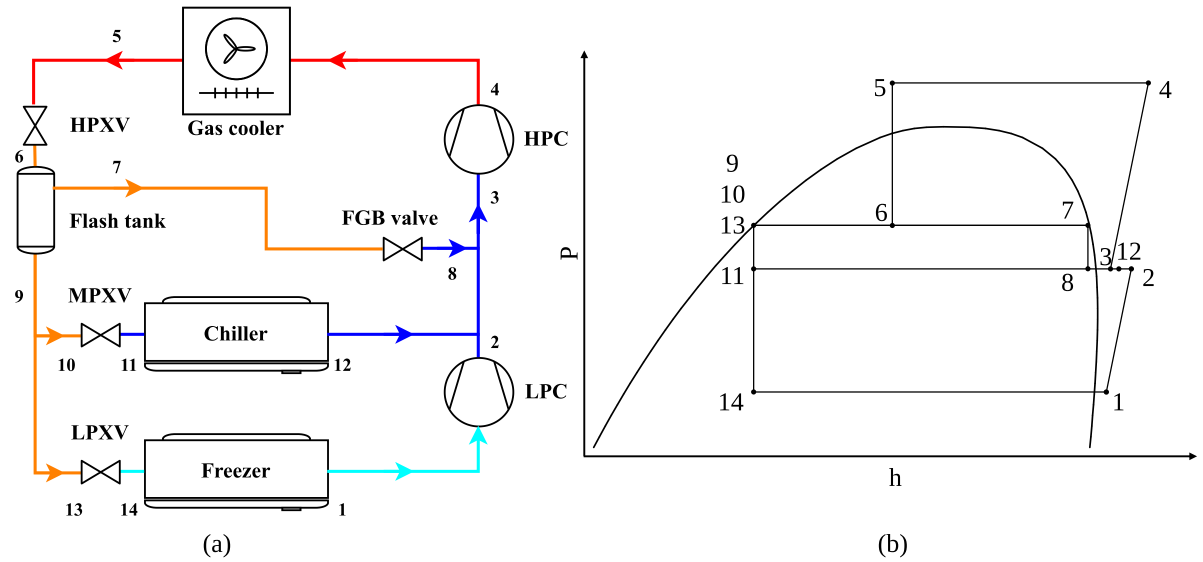

Figure 1. Transcritical CO_2 booster refrigeration cycle (BRC) (a) Plant layout, (b) P-h diagram

Transcritical CO_2 parallel compression refrigeration cycle (PRC) is an improvement made to BRC as shown in Figure 2. Vapor coming from the flash tank is separately compressed to the gas cooler pressure via a separate parallel compressor (PC). There is no expansion loss caused by the flash vapor. There is also a bypass circuit in case of insufficient mass flow rate in the PC at lower ambient temperatures. It works exactly the same as BRC in this case.

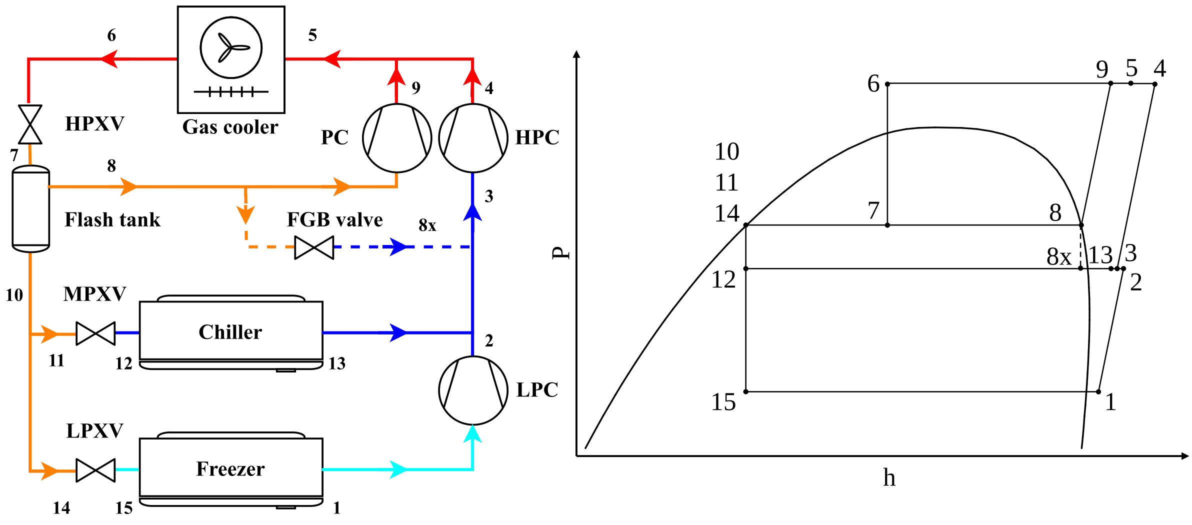

Figure 2. Transcritical CO_2 parallel compression refrigeration cycle (PRC) (a) Plant layout, (b) P-h diagram

The cycles were modeled in Engineering Equation Solver (EES) software [19] and the following assumptions were made:

- Cycles are operating in steady-state conditions.
- Expansion processes in the valves were assumed to be isenthalpic.
- Heat losses in the piping were neglected.
- Pressure losses in the piping and evaporators were neglected.
- Flash tank phase separation efficiency was taken as 100%.
- Freezer temperature and capacity were taken as -35 °C and 25 kW, respectively [20], [21].
- Chiller temperature and capacity were taken as -10 °C and 120 kW, respectively [20], [21].
- 5 K of useful superheat was applied to evaporators [21].
- Power consumptions of the evaporator and gas cooler fans were taken as 3% of the heat transfer ratio of the respective component [21], [22].
- Eq. (1) for used for compressor isentropic efficiency ($\eta_{comp,is}$) calculations [23] while global efficiency correlations were used for energy consumption calculations [21].

$$\eta_{comp,is} = -0.04478\frac{P_{comp,out}}{P_{comp,in}} + 0.9343 \tag{1}$$

Energy analysis of the cycles was made using Eqs. (2-4) [21].

$$\sum \dot{Q}_{in} + \sum \dot{m}_{in} h_{in} + \sum \dot{W}_{in} = \sum \dot{Q}_{out} + \sum \dot{m}_{out} h_{out} + \sum \dot{W}_{out} \tag{2}$$

$$COP_{BRC} = \frac{\dot{Q}_{LT}+\dot{Q}_{MT}}{\dot{W}_{HPC}+\dot{W}_{LPC}} \tag{3}$$

$$COP_{PRC} = \frac{\dot{Q}_{LT}+\dot{Q}_{MT}}{\dot{W}_{HPC}+\dot{W}_{LPC}+\dot{W}_{PC}+\dot{W}_{fan,LT}+\dot{W}_{fan,MT}} \tag{4}$$

Where $\dot{Q}$ is the heat transfer ratio, $\dot{m}$ is the mass flow rate of the refrigerant, h is the specific enthalpy, and $\dot{W}$ is the power consumption. Condenser/gas cooler conditions were determined based on Table 1. The cycles operate under transcritical conditions at ambient temperatures above 27 °C.

Table 1. Evaporator/gas cooler conditions [24]

T_{amb} (°C)	$T_{cond/GC,out}$ (°C)	$P_{cond/GC}$ (bar)
$T_{amb} \leq 5$	11	$P_{sat}@(13\ °C)$
$5 < T_{amb} \leq 14$	$T_{amb} + 6$	$P_{sat}@(T_{amb} + 8\ °C)$
$14 < T_{amb} \leq 27$	$0.7692T_{amb} + 9.23$	$1.397T_{GC,out} + 32.09$
$T_{amb} > 27$	$T_{amb} + 3$	Optimized

Evaporator loads vary between the minimum and design values based on the ambient temperature as shown in Table 2. This is because when ambient temperature deviates from the design point, the indoor air temperature and relative humidity will change and this will cause refrigeration loads to derivate from the design loads defined by the store refrigeration schedule [25].

Table 2. Evaporator load fractions [25]

T_{amb} (°C)	$\dot{Q}_{MT}$ (kW)	$\dot{Q}_{LT}$ (kW)
$T_{amb} < 5$	$0.66\dot{Q}_{MT,design}$	$0.80\dot{Q}_{LT,design}$
$5 \leq T_{amb} \leq 30$	$\left[1-(1-0.66)\left(\frac{30-T_{amb}}{30-5}\right)\right]\dot{Q}_{MT,design}$	$\left[1-(1-0.80)\left(\frac{30-T_{amb}}{30-5}\right)\right]\dot{Q}_{LT,design}$
$T_{amb} > 30$	$\dot{Q}_{MT,design}$	$\dot{Q}_{LT,design}$

The Total Equivalent Warming Impact (TEWI) method based on the EN378 standard was used for environmental impact calculations. Direct, indirect, and total TEWI were calculated using Eqs. (5-7). Emission caused by refrigerant leakage during the system lifetime and disposal contributes to direct TEWI while the one caused by energy production to operate the system contributes to indirect TEWI [26].

$$TEWI_{Direct} = GWP \times m \times [L \times n + (1-\alpha)] \tag{5}$$

$$TEWI_{Indirect} = E \times K \times n \tag{6}$$

$$TEWI_{tot} = TEWI_{Direct} + TEWI_{Indirect} \tag{7}$$

Where E is the annual energy consumption. GWP, annual leakage loss (L), operation lifetime (n), recovery factor (α), refrigerant charge (m), and CO_2 emission coefficient (K) were taken as 1, 15%, 10 years, 90%, 435 kg, and 0.551 kg CO_2/kWh, respectively [22], [27]–[30].

Exergy calculations of the cycles were made using Eqs. (8-12) [31].

Exergy rate of the mass flow:

$$\dot{E}_{mass} = \dot{m}[h - h_0 - T_0(s - s_0)] \tag{8}$$

Exergy transfer rates from the evaporators and gas cooler:

$$\dot{E}_{MT}^{Q} = \dot{Q}_{MT}\left(\frac{T_0}{T_{L,MT}} - 1\right) \tag{9}$$

$$\dot{E}_{LT}^{Q} = \dot{Q}_{LT}\left(\frac{T_0}{T_{L,LT}} - 1\right) \tag{10}$$

$$\dot{E}_{GC}^{Q} = \dot{Q}_{GC}\left(1 - \frac{T_0}{T_{H,GC}}\right) \tag{11}$$

Exergy balance throughout the cycle:

$$\sum \dot{E}_{mass,in} + \sum \dot{W} = \sum \dot{E}_{mass,out} + \sum \dot{E}^{Q}_{out} + \dot{E}_D \quad (12)$$

Dead state temperature and pressure were taken as T_{amb}+273 K, and 1.01325 bar, respectively. Source temperatures for freezer, chiller, and gas cooler were taken as 249 K, 272 K, and T_{amb} + 275.5 K, respectively while air inlet and outlet temperatures were taken as -23 °C and -25 °C for freezer, 2 °C and 0 °C for chiller, T_{amb} and T_{amb}+5 °C for gas cooler, respectively [32], [24], [30].

The Specific Exergy Costing (SPECO) method was used for exergoeconomic calculations. Capital investment and operation and maintenance costs are the non-exergy costs. Hourly capital investment $(\dot{Z}_k^{CI})$ and operation and maintenance $(\dot{Z}_k^{OM})$ costs of the components were calculated using Eqs. (13-15) considering purchased equipment cost (PEC), lifetime (n), operation hours (H), and interest rate of money (i_{eff}) [33], [34]. Table 3 presents PEC calculations of each component.

$$\dot{Z}_k = \dot{Z}_k^{CI} + \dot{Z}_k^{OM} \quad (13)$$

$$\dot{Z}_k^{CI} = PEC \times \frac{CRF}{H} \quad (14)$$

$$CRF = \frac{i_{eff}\left(1 + i_{eff}\right)^n}{\left(1 + i_{eff}\right)^n - 1} \quad (15)$$

Taxes, operation, and maintenance costs were not considered. 15% was added to the capital investment cost of each component for piping, automation, control instruments, and installation [35]. Annual interest rate and operation lifetime were taken as 2% and 10 years, respectively [27], [28], [36].

Table 3. Purchased equipment cost (PEC) values of the components

Component	Purchased equipment cost (€)	Reference
Compressor	$10167.5\dot{W}_{el,comp}^{0.46}$	[35], [37]
Gas cooler	$1397A_{GC}^{0.89} + 629.05\dot{W}_{fan,GC}^{0.76}$	[35], [38]
	$A_{GC} = \frac{\dot{Q}_{GC}}{U \times LMTD}$	
	$LMTD = \frac{\left(T_{GC,in} - T_{air,out}\right) - \left(T_{GC,out} - T_{air,in}\right)}{\ln\left(\frac{T_{GC,in} - T_{air,out}}{T_{GC,out} - T_{air,in}}\right)}$	
	$U = 0.575\, kW/m^2K$	
Evaporator	19226 € for chiller	[39]
	3243 € for freezer	
Expansion valve	$114.5\dot{m}_{exp}$	[40]
Flash tank	$280.3\dot{m}_{flash}{}^{0.67}$	[40]

Unit electricity price was multiplied by the constant escalation levelization factor $(CELF)$

considering inflation (r_n) and interest (i_{eff}) rates using Eqs. (16-18) [33], [34].

$$c_{el} = Electricity\ cost\ per\ kWh \times CELF \tag{16}$$

$$CELF = \frac{k(1-k^n)}{1-k} \times CRF \tag{17}$$

$$k = \frac{1+r_n}{1+i_{eff}} \tag{18}$$

Where k is the cost correction factor. Annual inflation rate (r_n) and electricity price (c_{el}) were taken as 9.2% and 0.2104 €/kWh, respectively according to Eurostat [41], [42].

Eqs. (19-23) were used as cost balance equations of the components [33], [34].

$$c_{comp,in}\dot{E}_{comp,in} + c_{el}\dot{W}_{comp,el} + \dot{Z}_{comp} = c_{comp,out}\dot{E}_{comp,out} \tag{19}$$

$$c_{ev,in}\dot{E}_{ev,in} + c_{Q_{ev}}\dot{Q}_{ev}\left(1-\frac{T_0}{T_L}\right) + c_{el}\dot{W}_{fan,ev} + \dot{Z}_{ev} = c_{ev,out}\dot{E}_{ev,out} \tag{20}$$

$$c_{GC,in}\dot{E}_{GC,in} + c_{el}\dot{W}_{fan,GC} + \dot{Z}_{GC} = c_{GC,out}\dot{E}_{GC,out} + c_{Q_{GC}}\dot{Q}_{GC}\left(1-\frac{T_0}{T_H}\right) \tag{21}$$

$$c_{exp,in}\dot{E}_{exp,in} + \dot{Z}_{exp} = c_{exp,out}\dot{E}_{exp,out} \tag{22}$$

$$c_{flash,in}\dot{E}_{flash,in} + \dot{Z}_{flash} = c_{flash,out,l}\dot{E}_{flash,out,l} + c_{flash,out,g}\dot{E}_{flash,out,g} \tag{23}$$

Where c is the hourly exergy cost, $\dot{E}$ is the exergy rate, $\dot{Z}$ is the hourly non-exergy cost of the component. Eqs. (24-26) were used as auxiliary equations [43].

$$c_{ev,in} = c_{ev,out} \tag{24}$$

$$c_{GC,in} = c_{GC,out} \tag{25}$$

$$c_{flash,out,l} = c_{flash,out,g} \tag{26}$$

3. RESULTS AND DISCUSSION

As temperature and pressure are independent of each other above the critical point, gas cooler outlet pressure needs to be optimized for maximum performance. Optimum gas cooler pressures for each gas cooler outlet temperature were obtained using the Golden Section Search method and Eqs. (27-28) were derived with curve-fitting. Intermediate pressure was taken as P_{MT} + 5 bar for BRC, and 55 bar under transcritical conditions for PRC [30]. Intermediate pressure for PRC, however, was taken as 45 bar under subcritical conditions.

$$P_{GC,opt,BRC} = 2.6529T_{GC,out} - 4.80083 \tag{27}$$

$$P_{GC,opt,PRC} = 2.35073T_{GC,out} + 1.85568 \tag{28}$$

Figure 3 shows COP comparison of the modeled cycles with the study done by Gullo et al. [21] under the same evaporator and gas cooler conditions. The results exhibited a satisfactory agreement with the literature.

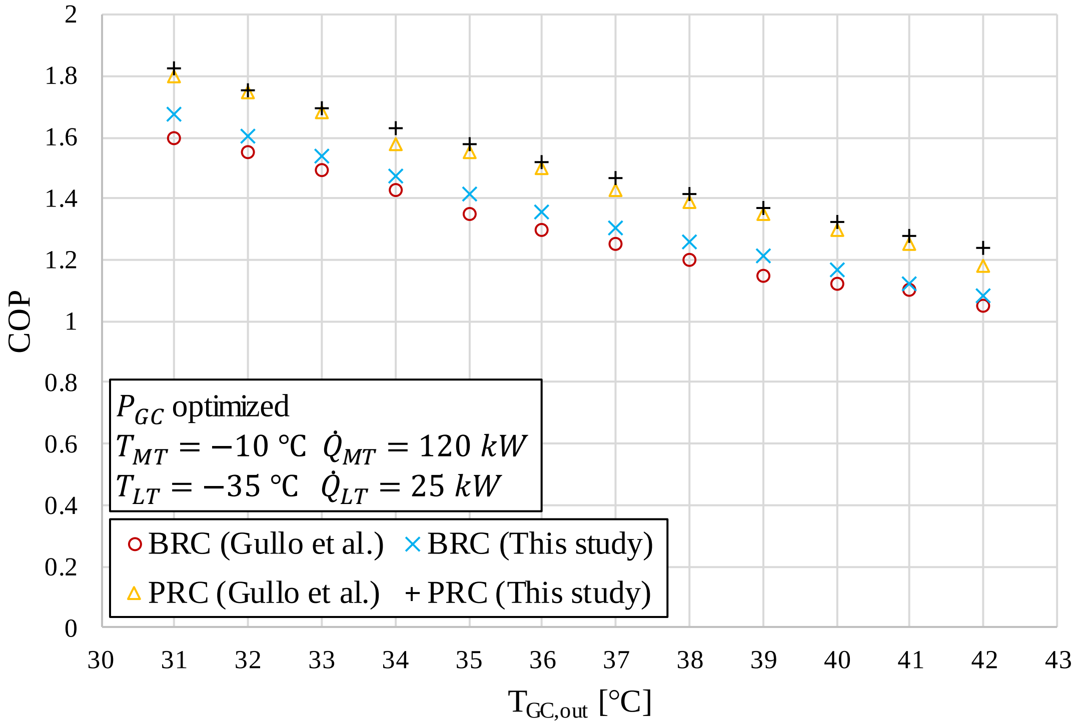

Figure 3. COP comparison of the investigated cycles with the literature

Figure 4 presents COP and total power consumption values of the cycles under different ambient temperatures. PRC was modeled as the same as BRC at ambient temperatures below 14 °C due to the insufficient mass flow rate in the PC. The cycles perform similarly under subcritical conditions. The performance difference increases under transcritical conditions with the increase in the ambient temperature. COP improvement and total power consumption reduction of PRC are up to 14.8% and up to 12.9%, respectively within the investigated ambient temperatures.

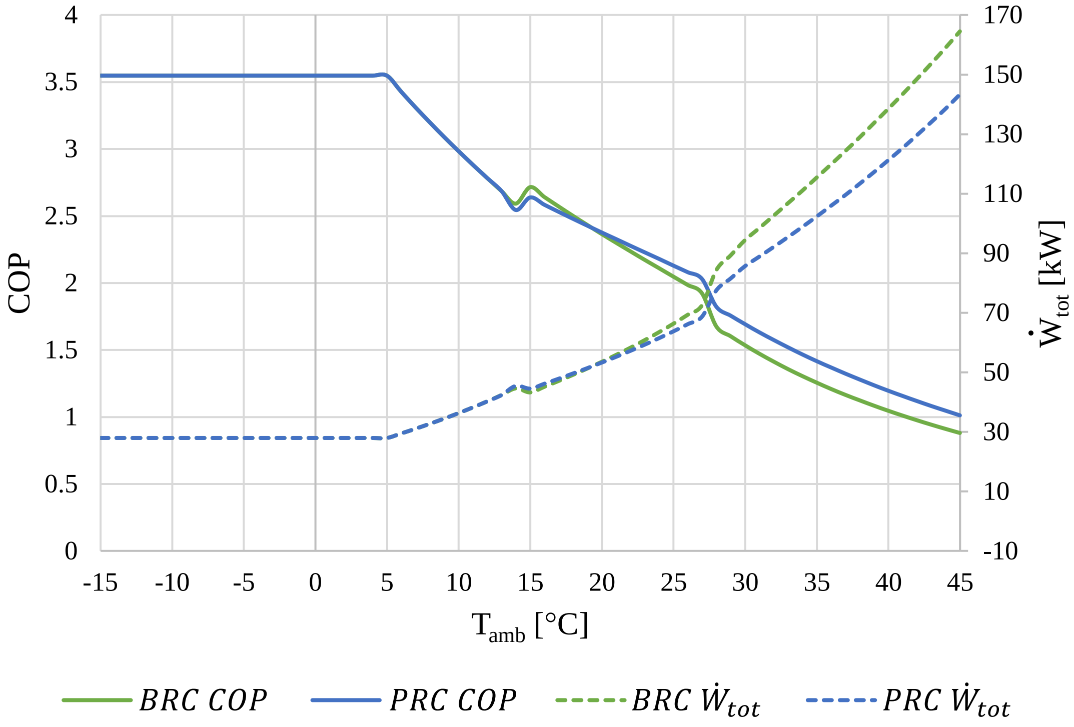

Figure 4. COP and total power consumption comparison of the cycles

Figure 5 presents bin-hour data for 11 provinces of Türkiye, which are in different climate zones. Bin-hours were derived using hourly dry-bulb temperatures of the provinces for the years 2016-2020 gathered from the Meteorological Data Information, Presentation and Sales System of Türkiye [44]. Temperature values were grouped into temperature bins with 3 °C increments and then the number of occurrence hours for each temperature bin was calculated. Power consumptions at the mid-point of each temperature bin were multiplied by the occurrence hours, and then these values were summed to calculate annual energy consumptions of the cycles.

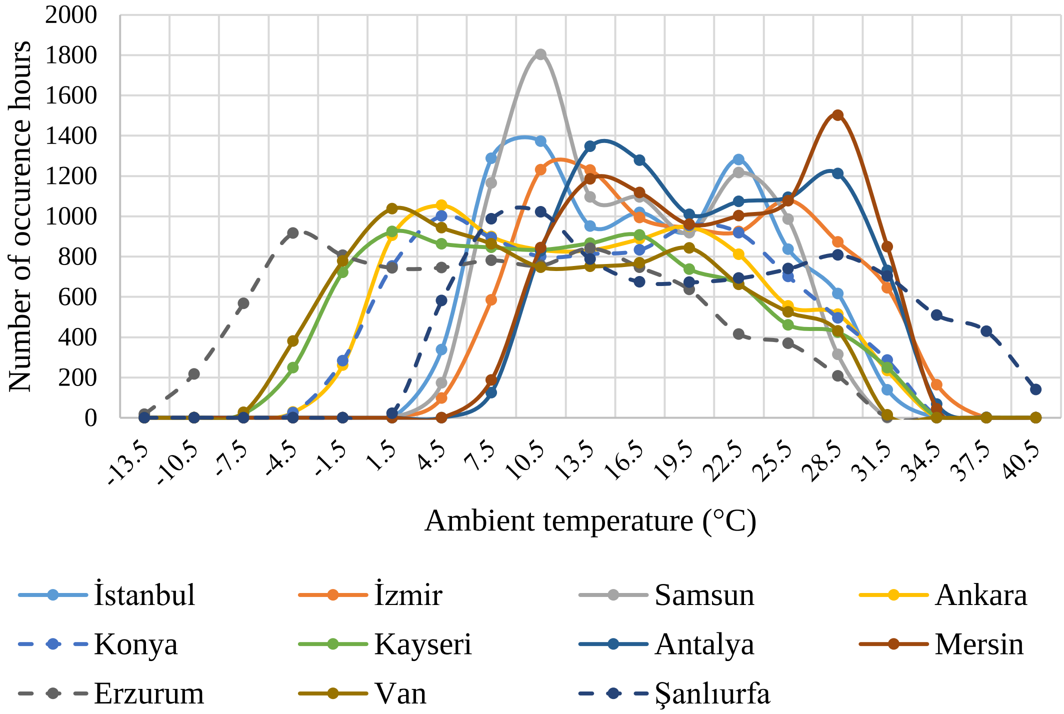

Figure 5. Bin-hour data for 11 provinces of Türkiye

Figure 6 shows the number of occurrence hours of the temperatures below and above 14 °C, which is the operating threshold of PC, for 11 provinces as well as annual mean temperatures. Erzurum and Van have the lowest annual mean temperatures while Antalya, Mersin, and Şanlıurfa have the highest. PC has the highest operation hours in Mersin and lowest in Erzurum.

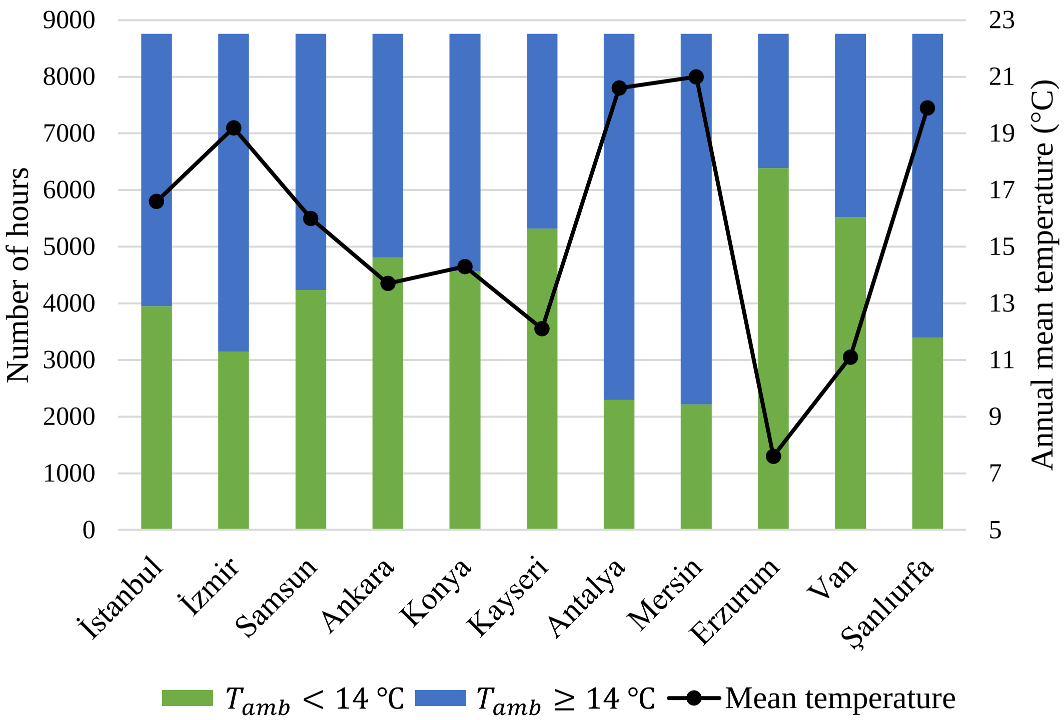

Figure 6. Number of hours at ambient temperatures below and above 14 °C, and annual mean temperatures of 11 provinces

Annual energy consumption values of the cycles for 11 provinces are presented in Figure 7. PRC outperforms BRC in all provinces. The difference is more identical in İzmir, Antalya, Mersin, and Şanlıurfa, which have warm climates. An increase in the ambient temperature increases the mass flow rate of PC which has a lower pressure difference compared to HPC. This results in higher performance compared to BRC in warmer provinces. In Erzurum and Van, which are cooler provinces, there is a small amount of difference as PC operation hours are low. Consumption for Erzurum and Van is below 400 MWh while it exceeds 500 MWh for Antalya, Mersin, and Şanlıurfa. PRC consumed 5.6% less energy than BRC in Şanlıurfa. Since GWP value of CO_2 is 1, direct TEWI of CO_2 cycles is negligible and total TEWI highly depends on the energy consumption. Figure 8 shows 10-year TEWI values of the cycles for 11 provinces. TEWI for Erzurum is below 2000 tons while it is above 2500 tons for İzmir, Antalya, Mersin, and Şanlıurfa.

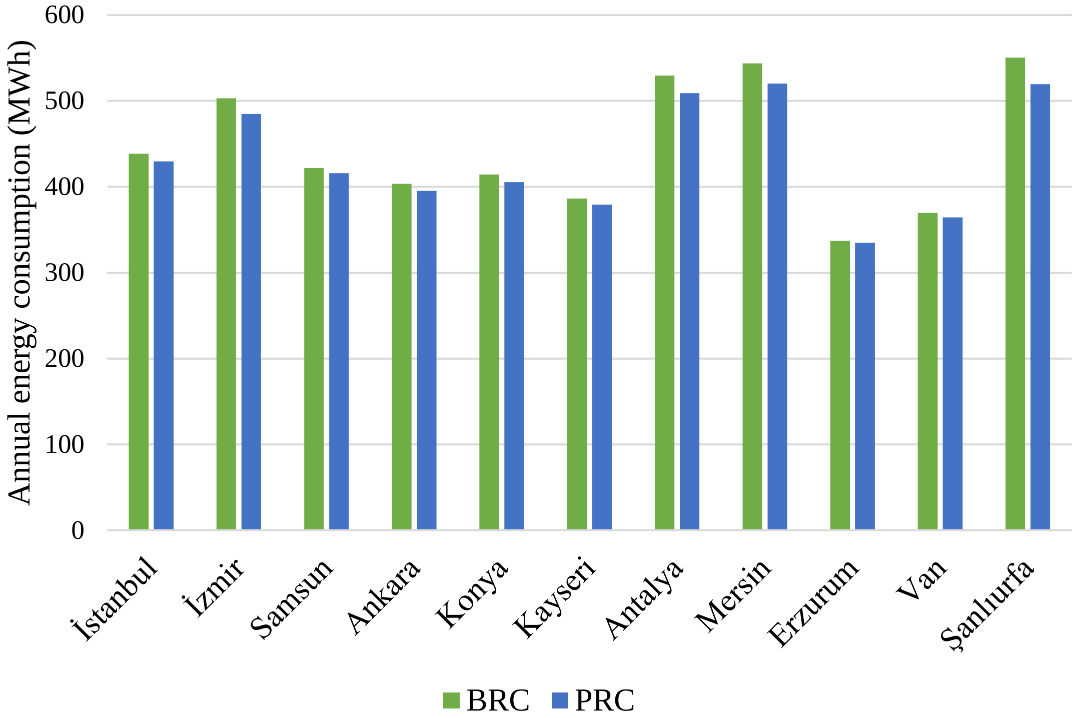

Figure 7. Annual energy consumption of the cycles for 11 provinces

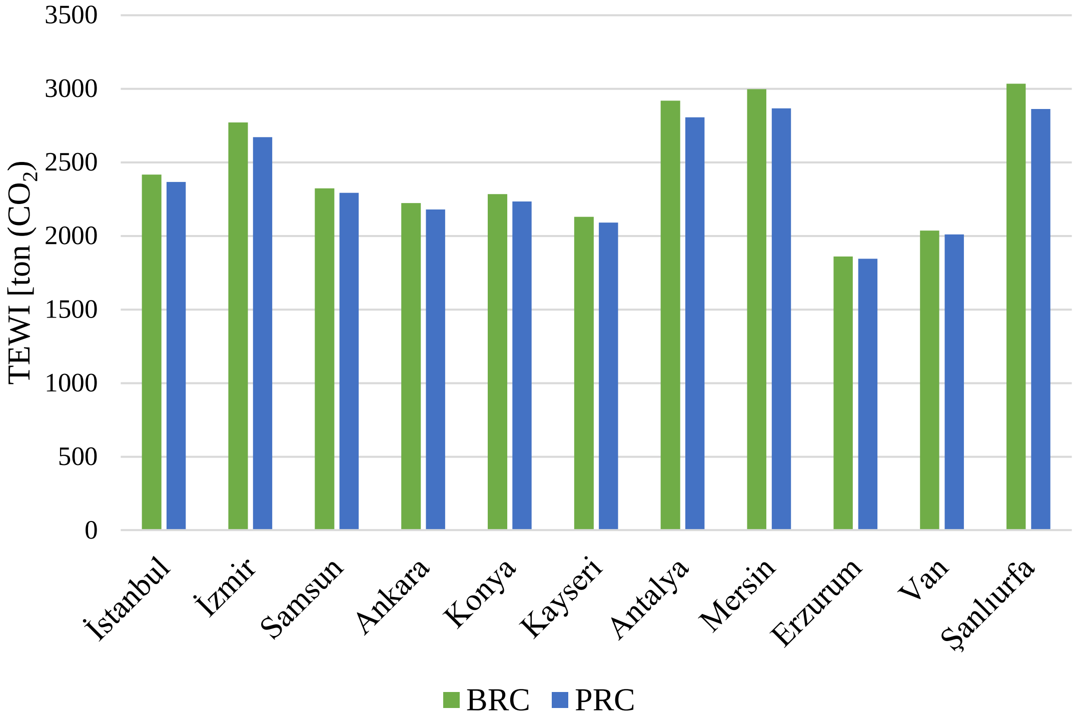

Figure 8. 10-year TEWI values of the cycles for 11 provinces

Figure 9 presents PEC values of the cycles for 11 provinces. Maximum ambient temperatures for the provinces were considered to ensure the proper operation of the cycles. The cost of PRC is higher than BRC as compressors are expensive components.

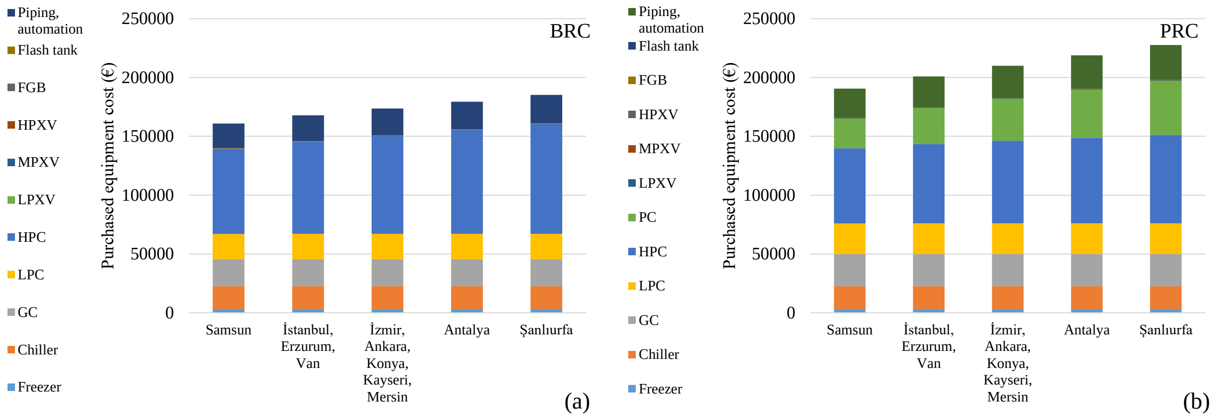

Figure 9. Purchased equipment costs of (a) BRC, (b) PRC

Unit product exergy costs of the cycles for 11 provinces at maximum ambient temperatures are presented in Table 4. Levelized electricity price was calculated as 0.3474 €/kWh. Although the equipment cost of PRC is higher than BRC, reduced energy consumption leads to lower product cost. Cost reduction in PRC varies between 8.2% and 18%. Provinces that have higher ambient temperatures have higher product costs due to higher energy consumption.

Table 4. Unit product exergy costs of the cycles for 11 provinces

		Unit product exergy cost (€/kWh)		
Province	$T_{amb,\max}$ (°C)	**BRC**	**PRC**	**Reduction**
İstanbul	31.5	1.957	1.748	10.7%
İzmir	34.5	2.021	1.75	13.4%
Samsun	28.5	1.894	1.739	8.2%
Ankara	34.5	2.021	1.77	12.4%
Konya	34.5	2.021	1.766	12.6%
Kayseri	34.5	2.021	1.779	12%
Antalya	37.5	2.088	1.752	16.1%
Mersin	34.5	2.021	1.743	13.8%
Erzurum	31.5	1.957	1.796	8.2%
Van	31.5	1.957	1.77	9.6%
Şanlıurfa	40.5	2.159	1.77	18%

4. CONCLUSIONS

In this paper, transcritical CO_2 booster (BRC) and parallel compression (PRC) supermarket refrigeration cycles were modeled in EES software and annual energy, TEWI, and exergoeconomic analysis were made deriving bin-hour data of 11 provinces in Türkiye. Up to 14.8% higher COP was obtained with PRC compared to BRC between the investigated ambient temperatures. Annual energy consumption in Erzurum and Van is below 400 MWh while It is above 500 MWh in Antalya, Mersin, and Şanlıurfa. Similarly, TEWI value in Erzurum is below 2000 tons while it is above 2500 tons in İzmir, Antalya, Mersin, and Şanlıurfa. PRC has lower energy consumption and TEWI values between 0.7% and 5.6% compared to BRC in the investigated provinces. The equipment cost of PRC is higher than BRC between 18% and 23% due to the additional compressor. Unit product exergy cost difference at the maximum ambient temperatures is between 8.2% and 18%. It was seen that since parallel compressor operates above the ambient temperatures of 14 °C, there is no significant annual consumption reduction in cooler provinces such as Erzurum and Van. The advantage of the use of parallel compressor is more significant in warmer provinces such as Antalya, Mersin, and Şanlıurfa. The performance of the cycles can be improved more by applying more advanced solutions such as ejector expansion, which is used for expansion work recovery, especially in warm climates [30]. Performance enhancement of the cycles using environmentally friendly refrigerants will contribute to a sustainable future.

Declaration of Ethical Standards

The authors of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

Credit Authorship Contribution Statement

Oğuz ÇALIŞKAN: Conceptualization, Methodology, Formal analysis, Software, Resources, Writing - original draft, Visualization.

H. Kürşad ERSOY: Conceptualization, Methodology, Formal analysis, Resources, Writing - review & editing, Supervision.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Data Availability

The data used to support the findings of this study are included within the article.

REFERENCES

[1] EPA, "Putting energy into profits : Energy Star guide for small business," *U.S. Environmental Protection Agency*, 2007. [Online]. Available: https://nepis.epa.gov/Exe/ZyPURL.cgi?Dockey=P100FCSW.txt. [Accessed: Dec. 02, 2022].

[2] IPCC/TEAP, "Safeguarding the ozone layer and the global climate system: Issues related to hydrofluorocarbons and perfluorocarbons," Intergovernmental Panel on Climate Change Technology and Economic Assessment Panel, 2005.

[3] M. Schulz and D. Kourkoulas, "Regulation (EU) No 517/2014 of the European Parliament and of the Council of 16 April 2014 on fluorinated greenhouse gases and repealing Regulation (EC) No 842/2006," 517/2014, 2014. Available: http://eur-lex.europa.eu/eli/reg/2014/517/oj.

[4] ASHRAE, "ANSI/ASHRAE Standard 34-2019: Designation and Safety Classification of Refrigerants," *American Society of Heating, Refrigerating and Air-Conditioning Engineers*, 2019.

[5] B. Yu, J. Yang, D. Wang, J. Shi, and J. Chen, "An updated review of recent advances on modified technologies in transcritical CO2 refrigeration cycle," *Energy*, vol. 189, p. 116147, Dec. 2019, doi: 10.1016/j.energy.2019.116147.

[6] F. Yilmaz and R. Selbaş, "Energy and Exergy Analyses of CO2/HFE7000 Cascade Cooling System," *Süleyman Demirel Univ. J. Nat. Appl. Sci.*, vol. 21, no. 3, pp. 854–860, Oct. 2017, doi: 10.19113/sdufbed.58140.

[7] S. Zong, X. Yin, T. Miao, S. Zhang, X. Yang, and F. Cao, "Experimental investigation on the cooling performance of direct and secondary loop CO2 air conditioning systems for electric vehicles," *Int. J. Refrig.*, vol. 152, pp. 376–386, Aug. 2023, doi: 10.1016/j.ijrefrig.2023.05.019.

[8] V. Sharma, B. Fricke, and P. Bansal, "Comparative analysis of various CO 2 configurations in supermarket refrigeration systems," *Int. J. Refrig.*, vol. 46, pp. 86–99, 2014, doi: 10.1016/j.ijrefrig.2014.07.001.

[9] H. Fritschi, F. Tillenkamp, R. Löhrer, and M. Brügger, "Efficiency increase in carbon dioxide refrigeration technology with parallel compression," *Int. J. Low-Carbon Technol.*, vol. 12, no. 2, pp. 171–180, Jun. 2017, doi: 10.1093/ijlct/ctw002.

[10] A. Chesi, F. Esposito, G. Ferrara, and L. Ferrari, "Experimental analysis of R744 parallel compression cycle," *Appl. Energy*, vol. 135, pp. 274–285, Dec. 2014, doi: 10.1016/j.apenergy.2014.08.087.

[11] C. Papazahariou, "Natural refrigerants faster to market, commercial refrigeration with low gwp alternatives," presented at the Joint Meeting of the Regional Ozone Networks for Europe & Central Asia and South Asia, Istanbul: Shecco, 2010.

[12] W. Chakroun, *Lower-GWP Alternatives in Commercial and Transport Refrigeration: An expanded compilation of propane, CO2, ammonia and HFO case studies*. Paris: United Nations Environment Programme, 2016.

[13] M. Garry, "Carrefour Poland Retrofits 46 Stores with R744," *r744.com*, Oct. 22, 2019. Available: https://r744.com/carrefour-poland-retrofits-46-stores-with-r744/. [Online], [Accessed: Nov. 18, 2022].

[14] R744.com, "Carrefour Timisoara: New CO2 multi-ejector refrigeration system is major success," *R744.com*, Dec. 16, 2015. [Online], Available: https://r744.com/carrefour-timisoara-new-co2-multi-ejector-refrigeration-system-is-major-success/. [Accessed: Nov. 18, 2022].
[15] A. Rubatto, "Carrefour using ejectors in CO2 transcritical system in Italy," *R744.com*, Mar. 12, 2018. [Online], Available: https://r744.com/carrefour-using-ejectors-in-co2-transcritical-system-in-italy/. [Accessed: Nov. 18, 2022].
[16] M. Garry, "Tesco Has Transcritical CO2 Refrigeration at 1,000 Stores.," *r744.com*, Oct. 19, 2022. [Online], Available: https://r744.com/tesco-has-transcritical-co2-refrigeration-at-1000-stores-one-third-of-its-estate/. [Accessed: Nov. 11, 2022].
[17] J. O. Haroldsen, "Canadian Longo's Commits to CO2 in New Stores and Renovations," *R744.com*, May 22, 2023. [Online], Available: https://r744.com/with-44-of-stores-using-co2-refrigeration-canadian-retailer-longos-commits-to-co2-in-new-stores-and-renovations/. [Accessed: Aug. 22, 2023].
[18] J. O. Haroldsen, "Japanese C-Store Operator Lawson Operates 5,028 Stores – Over 34% of Its Chain – With CO2 Refrigeration," *R744.com*, Jun. 21, 2023. [Online], Available: https://r744.com/japanese-c-store-operator-lawson-operates-5028-stores-over-34-of-its-chain-with-co2-refrigeration/. [Accessed: Aug. 22, 2023].
[19] S. A. Klein, "Engineering Equation Solver." F-Chart Software, 2020.
[20] S. Girotto, S. Minetto, and P. Neksa, "Commercial refrigeration system using CO2 as the refrigerant," *Int. J. Refrig.*, vol. 27, no. 7, pp. 717–723, 2004, doi: 10.1016/j.ijrefrig.2004.07.004.
[21] P. Gullo, A. Hafner, and G. Cortella, "Multi-ejector R744 booster refrigerating plant and air conditioning system integration – A theoretical evaluation of energy benefits for supermarket applications," *Int. J. Refrig.*, vol. 75, pp. 164–176, 2017, doi: 10.1016/j.ijrefrig.2016.12.009.
[22] M. Karampour and S. Sawalha, "State-of-the-art integrated CO2 refrigeration system for supermarkets: A comparative analysis," *Int. J. Refrig.*, vol. 86, pp. 239–257, Feb. 2018, doi: 10.1016/j.ijrefrig.2017.11.006.
[23] J. S. Brown, S. F. Yana-Motta, and P. A. Domanski, "Comparitive analysis of an automotive air conditioning systems operating with CO2 and R134a," *Int. J. Refrig.*, vol. 25, no. 1, pp. 19–32, Jan. 2002, doi: 10.1016/S0140-7007(01)00011-1.
[24] D. Tsimpoukis *et al.*, "Energy and environmental investigation of R744 all-in-one configurations for refrigeration and heating/air conditioning needs of a supermarket," *J. Clean. Prod.*, vol. 279, Jan. 2021, doi: 10.1016/j.jclepro.2020.123234.
[25] M. Zhang, "Energy Analysis of Various Supermarket Refrigeration Systems," in *International Refrigeration and Air Conditioning Conference*, Purdue, 2006.
[26] A. Zottl, M. Lindahl, R. Nordman, P. Rivière, and M. Miara, "Evaluation method for comparison of heat pump systems with conventional heating systems, D4.3. Concept for evaluation of CO2-reduction potential," *European Commision*, 2011.
[27] J. A. Shilliday, "Investigation and optimisation of commercial refrigeration cycles using the natural refrigerant CO2," Ph.D. dissertation, Brunel University, London, 2012. Available: http://bura.brunel.ac.uk/handle/2438/7454.
[28] P. Gullo, "Thermodynamic and environmental comparison of R744 booster supermarket refrigeration systems operating in Southern Europe," Ph.D. dissertation, University of Udine, Udine, Italy, 2016. Available: https://air.uniud.it/retrieve/handle/11390/1132962/251100/10990_827_PhD_thesis_GULLO.pdf.
[29] H. Dulkadiroğlu, "Türkiye'de elektrik üretiminin sera gazı emisyonları açısından incelenmesi," *Niğde Ömer Halisdemir Univ. J. Eng. Sci.*, vol. 7, no. 1, pp. 67–74, 2018, doi: 10.28948/ngumuh.369948.
[30] O. Caliskan and H. K. Ersoy, "Energy analysis and performance comparison of transcritical CO2 supermarket refrigeration cycles," *J. Supercrit. Fluids*, vol. 189, p. 105698, Oct. 2022, doi: 10.1016/j.supflu.2022.105698.

[31] T. J. Kotas, *The exergy method of thermal plant analysis*. London; Boston: Butterworths, 1985.

[32] P. Gullo, A. Hafner, and K. Banasiak, "Thermodynamic Performance Investigation of Commercial R744 Booster Refrigeration Plants Based on Advanced Exergy Analysis," *Energies*, vol. 12, no. 3, p. 354, Jan. 2019, doi: 10.3390/en12030354.

[33] A. Bejan, G. Tsatsaronis, and M. J. Moran, *Thermal design and optimization*. in A Wiley-Interscience publication. New York: Wiley, 1996.

[34] A. Lazzaretto and G. Tsatsaronis, "SPECO: A systematic and general methodology for calculating efficiencies and costs in thermal systems," *Energy*, vol. 31, no. 8–9, pp. 1257–1289, Jul. 2006, doi: 10.1016/j.energy.2005.03.011.

[35] F. Fazelpour and T. Morosuk, "Exergoeconomic analysis of carbon dioxide transcritical refrigeration machines," *Int. J. Refrig.*, vol. 38, pp. 128–139, Feb. 2014, doi: 10.1016/j.ijrefrig.2013.09.016.

[36] ECB, "Official interest rates," *European Central Bank*, May 04, 2023. [Online]. Available: https://www.ecb.europa.eu/stats/policy_and_exchange_rates/key_ecb_interest_rates/html/index.en.html. [Accessed: May 10, 2023].

[37] O. Rezayan and A. Behbahaninia, "Thermoeconomic optimization and exergy analysis of CO2/NH3 cascade refrigeration systems," *Energy*, vol. 36, no. 2, pp. 888–895, 2011, doi: 10.1016/j.energy.2010.12.022.

[38] Id. M. C. Santosa, K. M. Tsamos, B. L. Gowreesunker, and S. A. Tassou, "Experimental and CFD investigation of overall heat transfer coefficient of finned tube CO2 gas coolers," *Energy Procedia*, vol. 161, pp. 300–308, Mar. 2019, doi: 10.1016/j.egypro.2019.02.096.

[39] Friterm, "Friterm Product Selection Software." 2022.

[40] A. H. Mosaffa, L. G. Farshi, C. A. Infante Ferreira, and M. A. Rosen, "Exergoeconomic and environmental analyses of CO2/NH3 cascade refrigeration systems equipped with different types of flash tank intercoolers," *Energy Convers. Manag.*, vol. 117, pp. 442–453, Jun. 2016, doi: 10.1016/j.enconman.2016.03.053.

[41] Eurostat, "Inflation in the euro area," 2023. [Online]. Available: https://ec.europa.eu/eurostat/statistics-explained/index.php?title=Inflation_in_the_euro_area. [Accessed: Apr. 13, 2023].

[42] Eurostat, "Electricity price statistics," 2023. [Online]. Available: https://ec.europa.eu/eurostat/statistics-explained/index.php?title=Electricity_price_statistics. [Accessed: May 10, 2023].

[43] P. Gullo, B. Elmegaard, and G. Cortella, "Energetic, Exergetic and Exergoeconomic Analysis of CO2 Refrigeration Systems Operating in Hot Climates," in *ECOS 2015 28th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems*, Pau, France, Jul. 2015.

[44] O. Çalışkan, "CO2 akışkanlı ejektör genleştiricili süpermarket soğutma sistemlerinin Türkiye iklim şartlarında termodinamik, çevresel ve termoekonomik analizi," Ph.D. dissertation, Konya Technical University, Konya, Türkiye, 2022. Available: https://tez.yok.gov.tr/UlusalTezMerkezi/TezGoster?key=sELqxhTlFGAjsbjOuuiyCEpA0yENEpzFQC8tfkzv2-m2Bn7L1xl9fWYroF97GvpJ.