

Turkish Journal of Engineering



Turkish Journal of Engineering (TUJE)
Vol. 5, Issue 2, pp. 62-68, April 2021
ISSN 2587-1366, Turkey
DOI 10.31127/tuje.672888
Research Article

INVESTIGATION OF EFFICIENCY OF R717 REFRIGERANT SINGLE STAGE COOLING SYSTEM AND R717/R744 REFRIGERANT CASCADE COOLING SYSTEM

Ahmet Erhan, Akan ^{*1}, Fatih, Ünal ² and Derya Burcu, Özkan ³

¹Namık Kemal University, Çorlu Vocational School, Machinery and Metal Technologies, Tekirdağ, Turkey
ORCID ID 0000-0003-1806-7943
aeakan@nku.edu.tr

²Mardin Artuklu University, Mardin Vocational School, Machinery and Metal Technologies, Mardin, Turkey
ORCID ID 0000-0001-6660-9984
funal81@hotmail.com

³Yıldız Technical University, Mechanical Engineering, İstanbul, Turkey
ORCID ID 0000-0002-8964-3015
tumer@yildiz.edu.tr

* Corresponding Author

Received: 09/01/2020 Accepted: 02/03/2020

ABSTRACT

This study is an adaptation of ammonia cascade cooling systems using carbon dioxide on ice cream production machines and includes thermodynamic analysis of R717/R744 cascade refrigeration system with R717 refrigerant single-stage refrigeration system and investigation of its efficiency. As a result of the analyses, the COP value of the single-stage system was 3.67, the Carnot efficiency was 0.57, the second law efficiency was 0.19 and the power required to operate the compressor was 27.55 kW. In the cascade cooling system, the COP value was 4.46, the Carnot efficiency 0.59 and the compressor power 22.7 kW in the high-temperature part, while the COP value was measured as 14.65, the Carnot efficiency 0.58 and the compressor power 6.4 kW in the low-temperature part. For the whole cascade system, the COP 3.24 and the second law efficiency were found to be 0.43. When the results were compared, it was concluded that although the COP value of the cascade cooling system was 0.43 points lower than the single-stage ammonia system, when our country's climate conditions and the thermophysical properties carbon dioxide gas were considered together, one of the most suitable designs was the cascade cooling system.

Keywords: Cooling, Cascade systems, Refrigerant, Efficiency, Thermodynamic analysis.

1. INTRODUCTION

Ice cream is known as a popular nutrient for people because of its superior nutritional value, easy digestion, popular taste, aroma and refreshing effect. Today, the ice cream industry is developing rapidly. In this study, ice cream production processes to be examined are shown in Figure 1.

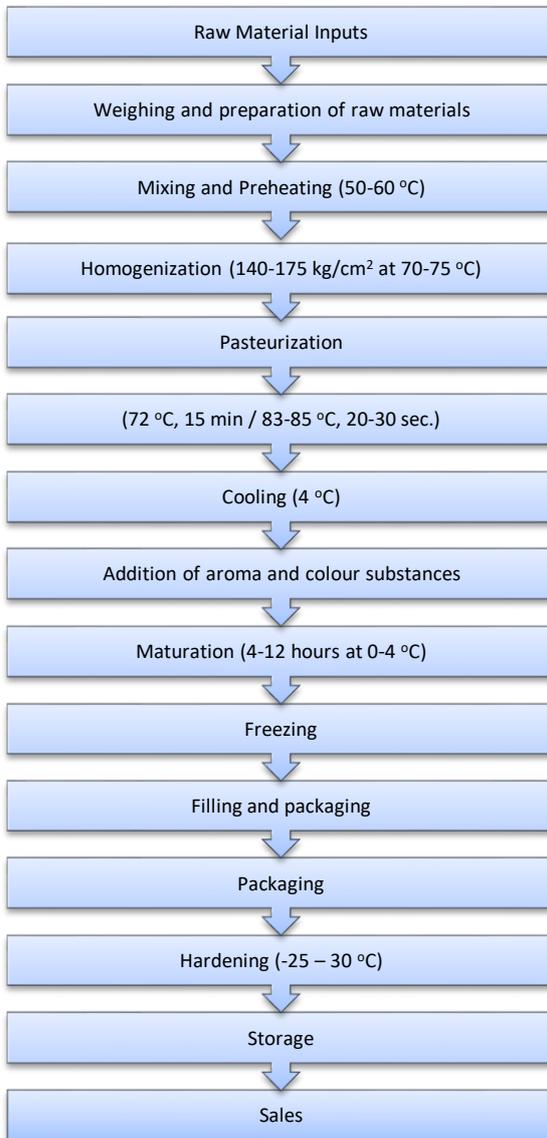


Fig. 1. Schematic representation of industrial ice cream production process

The mixture, which has matured, must be frozen in special machines in order to become freeze. The function of freezing is to freeze some of the water in the mixture. The freezing temperature varies depending on the type of freezer and the composition of the mixture but is generally between -3.5°C and -5.5°C . The freezing process is the most important stage of industrial ice cream production. However, this process needs to be studied in terms of energy consumption and efficiency. As an alternative to the single-stage mechanical compression cooling system used in the freezing process, cascade cooling systems can be preferred. The cascade cooling

systems are suitable for use in industrial applications, especially in the supermarket refrigeration industry, where the evaporation temperature of frozen food cabinets varies from -30°C to -50°C (Getu et al., 2007). Prepare your manuscript and illustrations in appropriate format, according to the instructions given below. Please also be sure that your paper conforms to the scientific and style instructions of the Journal, given below here.

When the studies in the literature on cascade cooling systems examined;

Dopazo et al., (2009), analyzed a CO_2/NH_3 cascade cooling system and determined the optimum CO_2 condensation temperature from the energy and exergy perspectives. The effects of operating parameters on the performance of a CO_2/NH_3 cascade cooling system were experimentally investigated by Bingming et al. (Bingming et al., 2009). The researchers observed that the COP value of the system was only slightly sensitive to the degree of overheating, but also was greatly affected by the temperature, evaporation, and condensation of the temperature differences in the graded heat exchanger. Dopazo et al., (2011), experimentally evaluated the CO_2/NH_3 cascade cooling system in an industrial freezer with an evaporation temperature of -50°C . They also investigated the effect of operating parameters on system performance and compared the results of common NH_3 two-stage cooling systems under the same operating conditions. They found that the COP of the cascade system had about 20% higher COP than a two-stage ammonia system with economizer and intercooler. Rezayan et al, (2011), developed a thermo economic optimization for a simple CO_2/NH_3 cascade cooling system without considering environmental analysis. They also investigated the effect of design parameters on the total annual cost of the system where ambient temperature, cooling capacity, and cold space temperature were limited. Ozyurt et al., (2016), conducted a thermodynamic analysis of the cooling system operating under two-stage and subcritical conditions using R717 (ammonia) in the high-temperature cycle and R744 (carbon dioxide) in the low-temperature cycle in their study. They carried out energy and exergy analyses in order to improve the optimum working conditions, performance coefficient and the efficiency of the second law in the cascade condenser of the cooling system and each equipment. By calculating the mass, energy, and exergy equations of the cascade cooling system, they calculated the optimum cascade condenser temperature. Moreover, researchers showed that increasing the evaporation temperature, superheating value, and super cooling value had a positive effect on the performance coefficient value. In a master's study, which was conducted by Alan (2011), the energy-saving potential of the two-stage overcritical CO_2 cycle was theoretically examined. Evaporator and gas cooler pressures were taken with variable parameters and how the performance coefficient is affected in the case of expansion turbines instead of the expansion valve was observed.

As can be seen from the aforementioned researches, studies are being carried out to find environmentally compatible refrigerant solutions instead of chlorofluorocarbons (CFC) and hydro chlorofluorocarbons (HCFC) which are used in cooling systems that accelerate global warming, have harmful effect to the ozone layer, and have a negative impact on the environment. Hydro fluorocarbons (HFCs) are

obtained by synthesis from natural gases such as chlorine-free ethane and methane, which substitute hydrogen for chlorine and are therefore defined as relatively more environmentally friendly. Natural refrigerants that do not affect global warming and do not harm the ozone layer are undoubtedly the most important alternatives in this process. CO₂ gas is one of the natural gases used for cooling. Supermarkets and cold stores require cooling at very low temperatures. Cooling at very low temperatures is either impossible or very difficult with single-stage cooling systems. However, in the low-temperature range of cascade systems, halogenated hydrocarbons are not the first choice that comes to mind because of their relatively insufficient thermo physical properties. Thus, there is a tendency to use carbon dioxide at the low-temperature stage. The superior heat transfer capability and large volumetric capacity of carbon dioxide make it possible to use in the cascade cooling systems with the low-temperature range. Furthermore, the environmental harm of carbon dioxide and its impact on global warming are less than that of artificial refrigerants. This is another factor that makes carbon dioxide the preferred one instead of others (Çerkezoğlu, 2010).

In this study, R717/R744 refrigerant cascade refrigeration system with R717 refrigerant is used as an alternative to the existing refrigeration system with single-stage mechanical compression, to freeze the ice cream in a factory producing ice cream in addition to the studies in the literature. Thermodynamic analysis of the used cascade cooling and the single-stage cooling systems were made, with the help of the Coolpack program and the systems were compared with each other. The study is thought to be useful in the selection of cooling systems used in the ice cream production.

2. MATERIAL AND METHODS

2.1. Introduction of Cooling Systems

The working principle of the vapor cycle cooling system is as follows. The refrigerant vaporized in the evaporator enters the compressor in the vapor phase and is compressed to a certain pressure in the compressor. The temperature of the gas, whose pressure increases, also increases spontaneously. The compressed gas in the compressor then enters the condenser. Condensation occurs at constant pressure in the condenser. The liquid exiting the condenser is subjected to constant enthalpy expansion as it passes through the throttling valve. The refrigerant from the expansion valve is in the form of a liquid-vapor mixture and this mixture enters the evaporator again. The same operations are repeated throughout the cycle (Çerkezoğlu, 2010). A schematic representation of the cycle is shown in Figure 2.

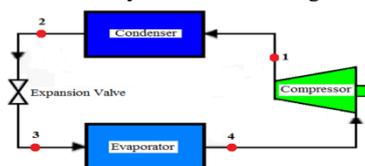


Fig. 2. The single-stage mixture refrigeration cycle

In the cycle shown in the Figure 2, the processes occur respectively as follows.
4-1 Adiabatic compression (in compressor)

- 1-2 Condensation at constant pressure and temperature (in condenser)
- 2-3 Pressure decrease in constant enthalpy (in expansion valve)
- 3-4 Evaporation at constant temperature and pressure (in evaporator).

The cascade refrigeration cycle is a cooling cycle consisting of two cycles with two different fluids that do not mix. One of the two cycles is usually the compression cycle, in which heat is transferred to the outside. In these systems, mainly R717 (NH₃, ammonia), R404A and other suitable HFC fluids are being used (Vestergaard, 2007). In this type of systems, the working pressure is around 40-45 bar (Onbaşıoğlu, 2010). The schematic representation of the system is shown in Figure 3 and the ln P-h diagram of the cycle is shown in Figure 4.

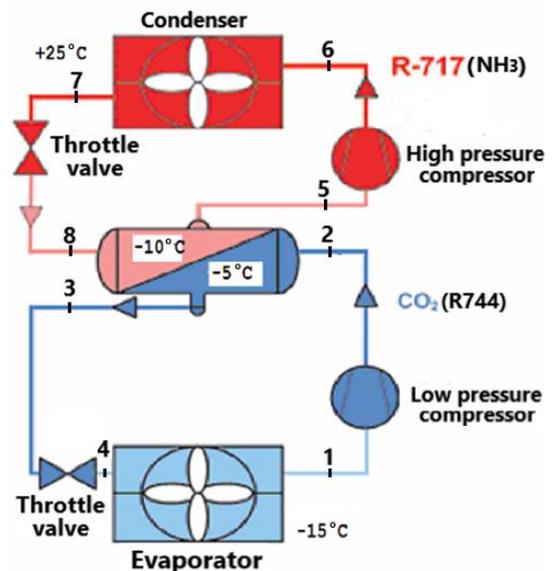


Fig. 3. The Cascade cycle diagram(Vestergaard, 2007)

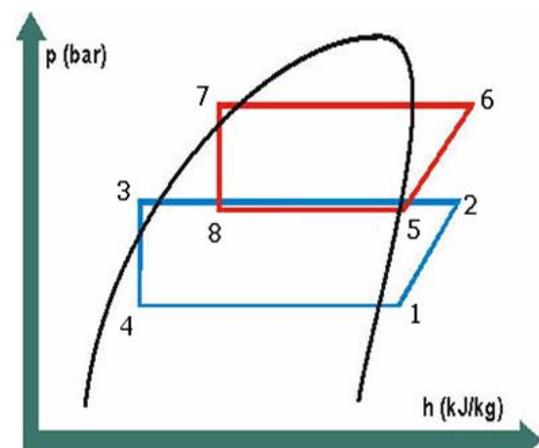


Fig. 4. ln P - h diagram of the Cascade Cycle (Kasap et al.,2011)

The state changes occurring in the cascade cycle diagram shown in Figure 3 and Figure 4 are as follows, respectively.

- 1-2 Isentropic compression in the CO₂ compressor
- 2-3 Condensation of CO₂ at constant pressure in the cascade condenser (barrel type)

- 3-4 Expansion of liquid CO₂ in the expansion valve
- 4-1 Evaporation at constant pressure in the evaporator
- 5-6 Isentropic compression in the R717 compressor
- 6-7 Condensation at constant pressure in the R717 condenser
- 7-8 Expansion on constant enthalpy in the R717 expansion valve
- 8-5 Evaporation of NH₃ at constant pressure in the cascade condenser (barrel type)

2.2. Determination of Cooling Load

Assumptions made in calculations;

*The compressors in the systems perform adiabatic compression.

*The expansion of the throttling valves takes place at constant enthalpy.

*Kinetic and potential energy changes were neglected.

*Evaporation temperature is -15°C, condensation temperature is 25°C, isentropic efficiency of compressors is 70% and heat loss rates of compressors are taken as 10%.

*The annual production capacity of the factory is 50x10⁶ liters.

*Density of ice cream is 490 g/Lt

*Mass flow of ice cream is 0.777 kg/s.

*The specific heat of the milk (c_{milk}) is 0.94 kcal/kg°C at 20°C, 0.90 kcal/kg°C at 4°C.

*10 people work 20 hours a day around the ice cream machine. The heat emitted by a person; room temperature was accepted as 215 kcal/h for 20°C.

*In order to illuminate the cooled area, 15 piece 100W fluorescent luminaires have operating 24 hours a day.

*Evaporator has 6 piece fan motors.

*The diameters of the two surfaces of the rotary type hopper are 0.2286 m and 0.240 m, respectively.

The following formulas were used to determine the heat load.

Depending on the room temperature, the heat load from the cooling of the ice cream was determined with the aid of Equation 1.

$$Q = m \cdot c \cdot \Delta T \quad (1)$$

In Eq. 1, Q is the cooling load (kW), c is the specific heat of the milk (kcal/kg°C), ΔT is the temperature difference (°C).

The heat load from working people and lighting fixtures is obtained with the help of Eq. 2.

$$Q = n \cdot W \cdot H \quad (2)$$

In Eq. 2, n is the number of elements, W is element strength (Watt) and H is the working time of the element.

The heat load caused by electric motors is calculated using Eq. 3.

$$Q = n \cdot W \cdot q \cdot H \quad (3)$$

In this Equation, q is the heat load from the electric motor. If the engine power is between 1/8 - 1/2 HP and the engine and driven elements are in the cold room, this value is taken as 1070 (kcal/hHP).

The heat load resulting from the temperature of the rotary freezer was determined with the help of Eq. 4.

$$Q = U \cdot A \cdot \Delta T \quad (4)$$

In this Equation, U is the total heat transfer coefficient (W/m²C) and A is the cross-sectional area of the rotor (m²).

The formulas used in the selection of system elements for the single-stage cooling system are given below, respectively.

Evaporator Selection,

$$Q_e = \dot{m}_{717}(h_4 - h_3) \quad (5)$$

Condenser Selection,

$$Q_c = \dot{m}_{717}(h_1 - h_2) \quad (6)$$

Compressor Selection,

$$Q_{comp} = \dot{m}_{717}(h_1 - h_4) \quad (7)$$

The formulas used in the selection of system elements for the two-stage cooling system (Cascade) are listed below.

For the high temperature part (HT)

Evaporator Selection,

$$Q_{e,HT} = \dot{m}_{717}(h_7 - h_6) \quad (8)$$

Condenser Selection,

$$Q_{c,HT} = \dot{m}_{717}(h_3 - h_5) \quad (9)$$

Compressor Selection,

$$Q_{comp,HT} = \dot{m}_{717}(h_2 - h_8) \quad (10)$$

For the low temperature part (LT)

Evaporator Selection,

$$Q_{e,LT} = \dot{m}_{744}(h_8 - h_6) \quad (11)$$

Condenser Selection,

$$Q_{c,LT} = \dot{m}_{744}(h_3 - h_5) \quad (12)$$

Compressor Selection,

$$Q_{comp,LT} = \dot{m}_{744}(h_8 - h_2) \quad (13)$$

Moreover, the cooling performance of the system was calculated with equation 14 and the second law efficiency was calculated with equation 15.

Cooling performance of systems,

$$COP = \frac{Q_e}{W_{net}} = \frac{Q_e}{W_{HT} + W_{LT}} \quad (14)$$

Second law efficiency of systems,

$$\eta_{II} = \frac{COP}{COP_{tr}} \quad (15)$$

3. FINDINGS and DISCUSSION

The findings obtained from the researches performed on the systems with the help of the equations given above are presented in Table 1 and 2.

Table 1. Cooling Load of System

The heat load from the cooling of the ice cream (Eq.1)	$Q_{1@20^{\circ}C} = 48.84 \text{ kW}$
	$Q_{2@4^{\circ}C} = 26.30 \text{ kW}$
The heat from lighting fixtures (Eq.2)	$Q_{\text{lighting}} = 1.54 \text{ kW}$
The heat from humans (Eq.2)	$Q_{\text{human}} = 2.27 \text{ kW}$
The heat from electric motors (Eq.3)	$Q_{\text{fan motors}} = 2.2 \text{ kW}$
The heat from the rotary freezer (Eq.4)	$Q_{\text{freezer}} = 10.67 \text{ kW}$
TOTAL	$Q_{\text{total}} = 91.82 \text{ kW}$

In the area where the ice cream machine is located, the total heat load was 91.82 kW and it was assumed that the total heat load was 101.3 kW with a 10% safety margin. In this case, the capacities of the selected system elements for both systems are presented in Table 2.

Table 2. Selection of system elements

For the single-stage cooling system	
$h_1 = 1723.6 \text{ kJ/kg} = 412.35 \text{ kcal/kg}$	
$h_2 = h_3 = 291 \text{ kJ/kg} = 69.62 \text{ kcal/kg}$	
$h_4 = 1439.6 \text{ kJ/kg} = 344.40 \text{ kcal/kg}$	
$\dot{m}_{717} = 0.08819 \text{ kg/s}$	
Evaporator Selection	$Q_e = Q_{\text{total}}$ $Q_e = \dot{m}_{717} \times (1439.6 - 291)$ = 101.3 kW
Condenser Selection	$Q_c = 0.08819 \times (1723.6 - 291)$ = 126.4 kW
Compressor Selection	$Q_{\text{comp}} = 0.08819 \times (1723.6 - 1439.6)$ = 25.1 kW
For the two-stage cooling system (Cascade)	
High Temperature Part (HT)	
$h_5 = h_6 = 291 \text{ kJ/kg} = 69.62 \text{ kcal/kg}$	
$h_7 = 1448.6 \text{ kJ/kg} = 345.992 \text{ kcal/kg}$	
$h_8 = 1451.2 \text{ kJ/kg} = 346.613 \text{ kcal/kg}$	
$\dot{m}_{717} = 0.0875 \text{ kg/s}$	
Evaporator Selection	$Q_e = Q_{\text{total}} = 101.3 \text{ kW}$ $Q_e = \dot{m}_{717}(h_7 - h_6)$ $\dot{m}_{717} = 0.0875 \text{ kg/s}$
Condenser Selection	$Q_c = 0.0875 \times (1682 - 291)$ = 121.7 kW
Compressor Selection	$Q_{\text{comp}} = 0.0875 \times (1681.8 - 1451.2)$ = 23.06 kW
Low Temperature Part (LT)	
Evaporator Selection	$Q_e = Q_c$
	$\dot{m}_{744} = \dot{m}_{717} \frac{(h_7 - h_6)}{(h_3 - h_5)}$
	$\dot{m}_{744} = 0.36 \text{ kg/s}$
	$Q_{e,LT} = 0.36 \times (-58.5 - (-323.5))$ = 95.4 kW
Condenser Selection	$Q_c = 101.3 \text{ kW}$

Compressor Selection	$Q_{\text{comp}} = 0.36 \times (42.4 - (-58.5))$ = 16.1 kW
-----------------------------	--

The accuracy of the values obtained after the theoretical calculations were compared with the CoolPack cooling analysis program. The analyzes of the systems were given in Figure 5-6 and the found values in Table 3-6 respectively.

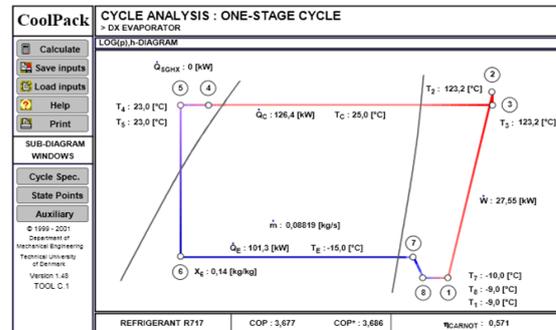


Fig. 5. Input values of the single-stage cycle examined in the CoolPack

Table 3. Values at each point of the single-stage cycle

State Point	Temperature (°C)	Pressure (kPa)	Enthalpy (kJ/kg)	Density (kg/m³)
1	-9.0	231.2	1442.4	1.9
2	123.2	1021.6	1723.6	5.5
3	123.2	1006.0	1724.0	5.4
4	23.0	1006.0	291.0	605.8
5	23.0	1006	291.0	605.8
6	-15.0	236.2	291	-
7	-10.0	236.2	1439.6	1.9
8	-9.0	231.2	1442.4	1.9

Furthermore, it has been seen that the COP value of the single stage refrigeration cycle was 3.67, the pressure ratio (P_2 / P_1) was 4.418, the $T_{2,1s}$ value was 98°C and the $T_{2,w}$ value was 135°C. In this expression, $T_{2,1s}$ value is the temperature of the discharge gas assuming reversible and adiabatic compression. In the same way, $T_{2,w}$ value is the temperature of the discharge gas assuming real and adiabatic compression.

Table 4. Supporting information on the single-stage cycle

Cooling Capacity		
$Q_e = 101.3 \text{ kW}$	$Q_c = 126.4 \text{ kW}$	$\dot{m} = 0.08819 \text{ kg/s}$
Compressor Performance		
Isentropic efficiency, $\eta_{1s} = 0.7$		$W_{\text{comp}} = 27.55 \text{ kW}$
Compressor Heat Loss		
Heatloss factor $f_0 = 10\%$	$T_2 = 123.2^{\circ}C$	$Q_{\text{loss}} = 2.755 \text{ kW}$
Suction Line		
Unuseful superheat	$Q_{sL} = 0.249 \text{ kW}$	$T_s = -9^{\circ}C$

For the two-stage cascade cooling system

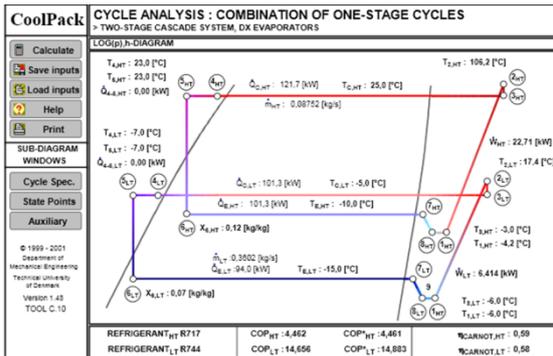


Fig. 6. Input values of the cascade cycle examined in CoolPack

Table 5. Values at each point of the cascade cycle

High Temperature (HT)				
State Point	Temperature (°C)	Pressure (kPa)	Enthalpy (kJ/kg)	Density (kg/m ³)
1	-4.2	288.6	-4.2	2.3
2	106.2	1012.2	1681.8	5.7
3	106.2	1006.0	1682.0	5.7
4	23.0	1006.0	291.0	605.8
5	23.0	1006.0	291.0	605.8
6	-10.0	290.9	291.0	-
7	-4.0	290.9	1448.6	2.3
8	-3.0	288.6	1451.2	2.3
9	-4.2	288.6	1448.3	2.3
Low Temperature (LT)				
State Point	Temperature (°C)	Pressure (kPa)	Enthalpy (kJ/kg)	Density (kg/m ³)
1	-6.0	2277.2	-58.5	55.9
2	17.4	3062.5	-42.4	69.1
3	17.4	3045.8	-42.2	68.7
4	-7.0	3045.8	-323.5	968.3
5	-7.0	3045.8	-323.5	968.3
6	-15.0	2290.7	-323.5	-
7	-9.0	2290.7	-62.5	57.7
8	-6.0	2277.2	-58.5	55.9
9	-	-	-	-

Table 6. Supporting information on the cascade cycle

Cycle Capacity		
High Temperature (HT)		
Q _{e,HT} =101.3 kW	ṁ=0.08752 kg/s	
Compressor Performance		
Isantropic efficiency, η _{is} =0.7	W _{HT} = 22.71 kW	
Compressor Heat Loss		
Heat loss factor f _Q = 10%	T ₂ =106.2 °C	Q _{loss} =2.3 kW
Suction Line		
Unuseful superheat	Q _{sL} =0.228 kW	T _s =-3 °C
Low Temperature (LT)		
Q _{e,LT} =94 kW	ṁ=0.3602 kg/s	
Compressor Performance		
Isantropic efficiency, η _{is} =0.7	W _{LT} = 6.414 kW	
Compressor Heat Loss		
Heat loss factor f _Q = 10%	T ₂ =17.4 °C	Q _{loss} =0.6 kW
Suction Line		

Unuseful superheat	Q _{sL} =1.456 kW	T _s =-6 °C
For the whole system; Total compressor power	W _{tot} = 29.12 kW	

If Figure 4-6 and Table 3-6 are examined, lower and upper stage evaporation and condensation temperatures of each cycle, heating and cooling capacities, compressor powers, heat loss rates of compressors, overcooling, overheating, unuseful superheating temperature differences and pressure losses can be seen.

4. CONCLUSIONS

In this study, a single-stage cooling system with R717 refrigerant and a cascade cooling system with R717/R744 system were compared with each other. As a result of the calculations, an evaporator load of 101.3 kW was found and a comparison was made for a system operating in the temperature range (25°C, -15°C) following the freezing production steps.

The calculations were made both manually and with the Coolpack program and the results were confirmed. As a result of the calculations, the COP value of the single-stage system was calculated as 3.67, the Carnot efficiency 0.57, the second law efficiency 0.19 and the power required to operate the compressor was 27.55 kW. In the cascade cooling system, the COP value was 4.46, the Carnot efficiency 0.59 and the compressor power 22.7 kW at the high-temperature part, the COP value was 14.656, the Carnot efficiency 0.58 and the compressor power 6.4 kW at the low-temperature part. For the whole cascade system, the COP 3.24 and the second law efficiency were found to be 0.43. When the results compared, it was seen that the COP values of the high and low-temperature parts of the cascade system were higher than the single-stage ammonia system, but when evaluated for the whole system, the COP value was less than 0.43 points. However, the usage of the cascade system is important when considering the fact that the cooling system will be used in the food sector and reducing environmental damage, as well as reducing the heat and losses emitted by decreasing compressor pressure rates.

Besides, considering both the climatic conditions of our country and the thermo physical properties of carbon dioxide gas, it was seen that the most suitable design that can use carbon dioxide gas is the cascade cooling system. Usage of the carbon dioxide in the low-temperature cycle of the cascade system eliminates the disadvantage of high pressure caused by the low triple point (31°C) and increases its usability in this system since carbon dioxide has a positive effective pressure even at temperatures as low as -55°C. The minimum temperature that can be achieved when ammonia used in a low-temperature cycle is -35°C. The pressure of ammonia gas below this temperature falls below the atmospheric pressure and it circulates with the air-fluid, causes damage to the equipment due to moisture in it, leads leakage, and poisoning. However, carbon dioxide does not have such disadvantages and provides optimum operation at -10°C. In conclusion, the overall superiorities of carbon dioxide make it a highly suitable secondary refrigerant for the cascade system.

REFERENCES

- Alan, A. (2011). The effect of expansion turbine usage on energy potential in dual stage CO₂ cooling systems, MSc Thesis, University of Süleyman Demirel, Isparta, Turkey.
- Bingming, W., Huagen, W., Jianfeng, L., Ziwen, X. (2009). "Experimental investigation on the performance of NH₃/CO₂ cascade refrigeration system with twin-screw compressor." *International Journal of Refrigeration*, Vol. 32, No. 6, pp. 1358–1365.
- Çerkezoğlu, B. (2010). Analysis of Cascade Cooling System Using Co₂, MSc Thesis, Istanbul Technical University, Istanbul, Turkey.
- Dopazo, J. A., Fernández-Seara, J., Sieres, J., Uhía, F. J. (2009). "Theoretical analysis of a CO– NH cascade refrigeration system for cooling applications at low temperatures." *Applied Thermal Engineering*, Vol. 29, No. 8-9, pp. 1577–1583.
- Dopazo, J. A., Fernández-Seara, J. (2011). "Experimental evaluation of a cascade refrigeration system prototype with CO₂ and NH₃ for freezing process applications." *International Journal of Refrigeration*, Vol. 34, No. 1, pp. 257–267.
- Getu, H. M., Bansal, P. K. (2007). "Modeling and performance analysis of evaporators in frozen food supermarket display cabinets at low temperatures." *International Journal of Refrigeration*, Vol. 30, No. 7, pp. 1227-1243.
- Kasap, F., Acül, H., Canbaz, H., Erbil, S. (2011). "R744 (CO₂) Refrigerant cooling systems, finned tube R744 (CO₂) evaporator and gas cooler design principles." X. *National Installation Engineering Congress*, İzmir, Turkey, pp. 391-410.
- Onbaşıoğlu, H. (2010). "An Overview of Vapour Compression System Using Natural Refrigerant CO₂". *İSKİD ACV&R Journal of Turkey*, Vol.2, pp.48-64.
- Özyurt, A., Erdönmez, N., Yılmaz, B., Yılmaz, D., Sevindir, M. K., Mançuhan, E. (2014). "Thermodynamic Analysis and Performance Evaluation of CO₂ / NH₃ Cascade Cooling System." 15th *International Refrigeration And Air Conditioning Conference*, Plant Engineering, Purdue, Indiana pp. 2598-2608.
- Rezayan, O., Behbahaninia, A. (2011). "Thermoeconomic optimization and exergy analysis of CO₂/NH₃ cascade refrigeration systems." *Energy*, Vol. 36, No. 2, pp. 888–895.
- Vestergaard, N. P. (2007). "Danfoss Refrigeration and Air Conditioning Division, CO₂ Refrigerant for Industrial Refrigeration". Danfoss A/S (RA Marketing/MWA), DKRCLPZ.000.C1.02 / 520H2242.