Thermodynamic analysis on the performance of an R717/R744 cascade refrigeration system for food retail applications

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Abstract: Natural refrigerants, which naturally occur in the environment and are non-synthetics substances, have been used as cooling fluids in refrigerator systems for more than hundred years. However, due to safety and performance issues the chemical refrigerant substances replaced them. Over the last two decades, due to the component availability on the market natural refrigerants are getting back all the attention to re-establish their use into applications where previously HFCs were the preferred and only options. Natural refrigerant substances include CO_2 (R744,) ammonia (R717) and hydrocarbons. Due to the limitation on refrigerant charge capacity for hydrocarbon systems, ammonia and CO_2 are promising alternatives for food retail centralised refrigeration systems. Both ammonia and carbon dioxide have superior thermodynamic properties. On the other hand, the high toxicity of ammonia and the very high operating pressure of carbon dioxide create barriers on the system applications. Large amount of ammonia is prohibited from the sales area of food retail shops. Moreover, carbon dioxide is less efficiency comparing to HFCs installations for warm climate applications. The combination of the two refrigerants in a cascade configuration, where the ammonia side is located far away from the sales area and the subcritical operation of the carbon dioxide without effect from external ambient conditions, may lead to high system efficiencies. In this paper, a thermodynamic analysis on the performance of a R717/R744 cascade refrigeration system is discussed. The cascade configuration consists of the R717 side, the cascade heat exchanger, a R744 liquid receiver, medium and low direct expansion load, a low temperature compressor and high temperature compressor. Modelling results show that the coefficient of performance of the analysed cascade refrigeration system decreased from 2.19 to 1.56 when the ambient temperature varied from 7 ºC to 40 ºC. In addition, the second law efficiency of the analysed cascade refrigeration system varied from 0.494 to a maximum of 0.544 between ambient temperatures of 7 and 31 ºC, respectively. Finally, the amount of exergy destruction in the condenser, MT evaporator, cascade exchanger and the R717 compressor of the analysed system accounts for large percentage in total exergy destruction.

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Keywords: Thermodynamic analysis, cascade system, ammonia, CO_2, food retail.

1. INTRODUCTION

Montreal-Kyoto Protocols in the late of 1990s and F-Gas regulations earlier this century describe the necessity to phase-out and eventually eliminate the usage of harmful refrigerants with high Global Warming Potential (GWP). The impact of those has renewed the interest in natural refrigerants such as ammonia (R717), carbon dioxide (R744) and hydrocarbons (R290/R600a/R1270). R717 carries a B2 (BS EN 378-1:2016) safety classification which indicates that it is a high toxic refrigerant and carries a medium flammability risk. The high toxicity of R717 places some restrictions on where it can be applied in terms of safety. R717 is a very common refrigerant for industrialised applications where high cooling load demands are needed. Regarding the hydrocarbon refrigerants, they have a very high flammability risks. The main barrier for refrigeration applications is the restriction of the amount of charge per system which drives the hydrocarbon solution to be more appropriate for small capacity systems such as remote refrigerated cabinets and vending machines. In the case of the R744, it has grown in popularity over the other natural refrigerants, especially in supermarket refrigeration systems given the fact that it has negligible Global Warming Potential (GWP), zero Ozone Depletion Potential (ODP). Alongside its excellent environmental characteristics, R744 has favourable thermo-physical properties in terms of higher density, specific heat, volumetric cooling capacity, latent heat, thermal conductivity, and it is non-toxic and non-flammable in a wide range of concentration. Some physical properties of the R744 and R717 refrigerants are presented in Table 1. On the other hand, when the R744 refrigeration systems operate in warm climate conditions as the only refrigerant (booster solutions) the system moves from subcritical to transcritical operation. In transcritical operation the heat rejection heat exchanger operates as a gas cooler without any phase change. Transcritical operation is less efficient due to the high pressures on the gas cooler side and pressure ratio across the high-pressure compressors (HP). Several studies available in the open literature present different system arrangements analysis where the R744 is used as only refrigerant and potential solution to increase the system performance even for high ambient temperature applications such as those in Athens, Greece (Tsamos et al., 2017). The main barrier for R744 applications is the high pressure of the system and reduction of the performance when it is operating in transcritical mode. For supermarket applications with big refrigerated plants a steel pipe or K65 copper pipe may be used to prevent any damage of the system. In addition, various pressure relief valves need to be carefully designed and installed on the system to prevent over pressure events. A good alternative solution for big supermarket refrigeration systems is the cascade system solution where two different refrigerants exchange heat on a cascade heat exchanger. This system is divided in two sections; an upper stage and low stage circuit. The operation of the high stage circuit of the system is controlled by the temperature difference across the cascade heat exchanger. Both systems exchange heat on the cascade condenser which acts as an evaporator for the high stage and as a condenser for the low stage circuit. This arrangement can feed both medium temperature (MT) and low temperature (LT) cooling loads. In this case the high stage of the system uses R717; the circuit can be located in the secured distance machine room away from the sales area and R744, which operates in lower stage circuit of the system, flows inside the store and feed the refrigerated cabinets. The great advantage of the cascade configuration is that the R744 system operates in a subcritical mode during the whole year around without being affected from the ambient conditions. Also, the R717 system operates in a refrigeration plant very far from the customer area.

Ma et al., (2014) presented the use of a falling film evaporator-condenser as the cascade heat exchanger for a R717/R744 refrigeration system. The results show lower temperature difference across the proposed cascade heat exchanger which drives the system to higher COP values compared with the system where the plate, shell-and-plate or shell-and-tube heat exchangers are used. Rezayan and Behbahaninia (2011) investigated the thermos-economic optimization for the R717/R744 cascade refrigeration cycle. The authors proposed method included the thermal and economical aspects of the system design and operation. Based on the proposed design and operation of the system and for constant cooling capacity of 40 kW the annual cost reduction of the system is equal to 9.34% compared with the base case design. Messineo (2012) compared the R744/R717 cascade system with a R404A two-stage system based on similar operating conditions. The author reported the overall COP for different condensing temperatures, degrees of sub-cooling and evaporating temperatures. The experimental analysis of a R717/R744 cascade system for supermarket refrigeration applications has been presented from Sawalha (2008). The R744 refrigerant used to cool-down the LT and MT where a DX and flooded coil were used respectively. Sawalha concluded that the performance for the R717/R744 cascade solution was 50 to 60 % higher than that of a direct R404A system installed in the same laboratory environment. The effect of variation in evaporator and condenser temperature on cascade condenser temperature and the system performance of R717/R744 and R1270/R744 refrigeration systems were presented by Arora (2016). The author used energy and exergy analysis to analyse the effects of various parameters on the system performance. The author identified the optimum cascade condenser temperature where the system achieved the higher COP and exergetic efficiency. The relationship between cascade temperature with evaporator temperature and isentropic efficiency of the compressors was also described. The values of maximum COP and exergetic efficiency of R717/R744 were 7 - 9% higher than those of the R1270/R744 system.

In the most of published works the analysis involved only cascade refrigeration systems where the low stage circuit feeds only low temperature evaporators (Mosaffa et al.,2016, Sun et al. 2016, Nasruddin et al.,2016, Kilicarslan and Hosoz, 2010). In this paper, a thermodynamic analysis based on the first (energy analysis) and second law (exergy analysis) is carried out to assess the performance of a R717/R744 multi-temperature/multi-compression refrigeration system including a low temperature heat recuperator. The multi-temperature describes the system ability to feed both medium temperature (MT) and low temperature (LT) sides. Multi-compression is also used to feed the needs of different pressure levels across the proposed refrigeration system. The energy analysis allows determining the coefficient of performance of the analysed cascade configuration while the exergy analysis allows estimating the irreversibility rates in the system in order to identify the most critical component.

Table 1: Refrigerants Data and Safety Classification (BS EN 378-1:2016)

Inorganic Compounds	Molecular mass ($g.mol^{-1}$)	T_b (°C)	T_c (°C)	P_c (Mpa)	ASHRAE Safety Group	ODP	GWP	OEL (ppm)
CO_2 (R477)	44,01	-78,5	31	7,38	A1	0	1	5000
NH_3 (R717)	17,03	-33,4	132	11,3	B2	0	0	25

2. DESCRIPTION OF THE SYSTEM

Figure 1 shows a simplified schematic diagram of the proposed R717/R744 multi-temperature cascade refrigeration system for food retail applications. The proposed system consists of two loops: the upper-side loop fed with R717 and low-side with R744 as working fluid. Both sides are exchanging heat through the cascade heat exchanger which operates as evaporator for the R717 side and condenser for the R744 side. The cascade system solution guarantees a subcritical operation all the year around for the R744 system and higher performance when this is used in warm climate applications. Additionally, the high-pressure R744 expansion valve is no longer required due to the low temperature of R744 condensation. The upper-side loop consists of a R717 screw compressor, an air-cooled condenser, an expansion valve, and the upper side of the cascade heat exchanger (evaporator). The low-side R744 consist of a liquid receiver, and direct expansion (DX) medium temperature (MT) and low temperature (LT) evaporators. In order to control the pressure difference between the LT and MT side a double stage compression is employed. A solenoid valve and by-pass valve is installed (in series) at the upper side of the liquid receiver. This is an additional safety factor and will be used in order to by-pass the R744 gas phase from the liquid receiver to the suction of the compressor to prevent any over-pressure on the system when is needed. In additional, an internal heat recuperator (IHX) is used to ensure the system have superheated R744 vapour at the suction line of the low-pressure compressor. For both the MT and LT compressors, semi-hermetic compressors were used.

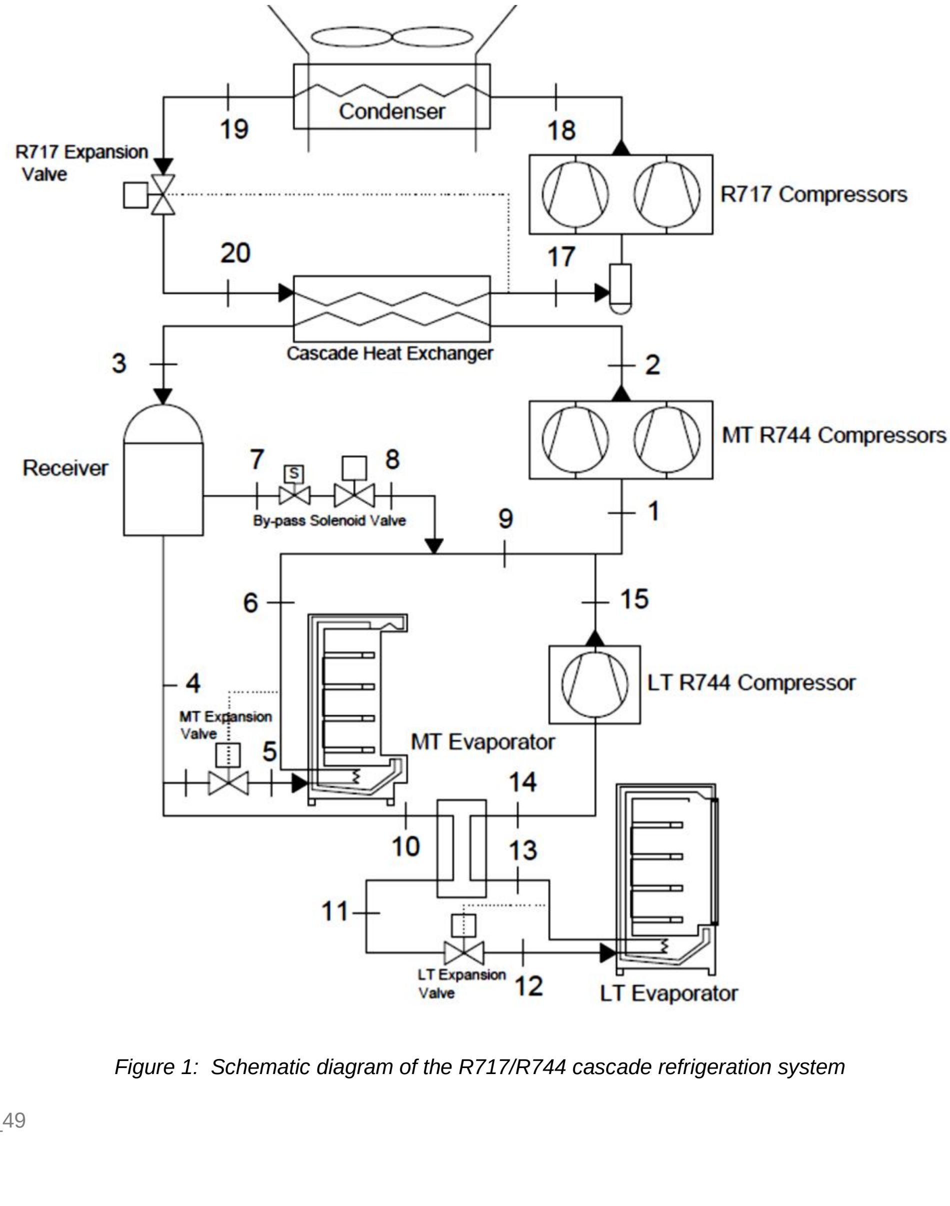

Figure 1: Schematic diagram of the R717/R744 cascade refrigeration system

The superheated R717 after the upper cascade side is compressed from the screw compressor and pass from the air-cooled condenser. After the isenthalpic expansion, the saturated R717 flows through the upper side of the cascade heat exchanger. At the same time, the superheated R744 vapour enters the cascade condenser. In this point, the R717 evaporates to a superheated vapour and close the cycle for the high-side loop while the R744 condenses to a saturated liquid. The saturated liquid flows to the liquid receiver. After the isenthalpic expansion on the LT side, the saturated R744 pass through IHX and then the LT evaporators where is saturated vapour. Before the R717 vapour enters the suction of the LT compressor, it is passed though the IHX to make sure that the LT compressor is fed with superheated R744 vapour. The saturated R744 vapour leaving the MT evaporator is mixed with R744 vapour coming from the LT compressor before entering to suction of MT compressor and complete the cycle. Figure 2 depicts the P-h diagrams of the R744 and R717 circuits.

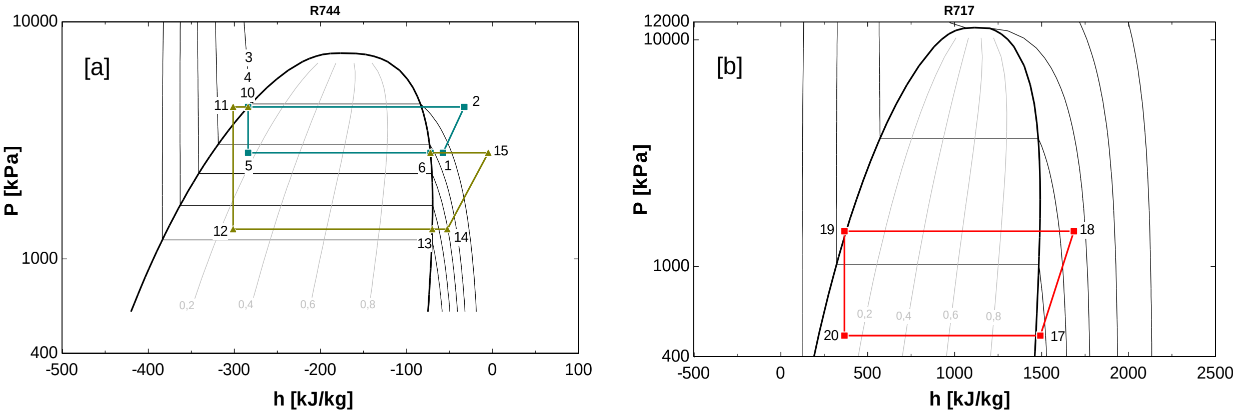

Figure 2: P-h diagrams of [a] the R744 refrigeration system and [b] R717 system.

3. THERMODYNAMIC MODEL DETAILS

The R717/R744 cascade refrigeration system was simulated performing mass, energy, and exergy balances on each one of the components of the whole system. The thermodynamic analysis of the system was conducted assuming that the cascade refrigeration system operates under steady state conditions and that pressure drops and kinetic and potential energy through the system components are negligible, as well as heat and friction losses. It was also assumed the ambient air as heat sink and an approach difference between the refrigerant condensing temperature and temperature of the inlet air of 5°C. The sub-cooling of the refrigerant leaving the condenser was assumed as 5 °C. In the case of the refrigerant leaving the MT and LT evaporators, it was assumed saturated vapour. In addition, the heat released by R744 condensation process is rejected to the R717 evaporation process. In the case of the R717 refrigerant leaving the cascade heat exchanger, the superheating was set to 10 °C. The throttling process within the expansion valves was assumed as isenthalpic and the effectiveness of the low temperature heat exchanger as 0.4. Finally, the power consumption of the MT/LT evaporators fans, lights, and other electrical devices as the defrost system was set to 10.5 kW, while the condenser fan power consumption was set to 7.5 kW (Tsamos et al., 2017). The Engineering Equation Solver software was used for the modelling (EES, 2014).

3.1. Energy balances

Applying the first law of thermodynamics, the coefficient of performance of the R717/R744 cascade refrigeration system shown in Figure XX is expressed as:

$$COP = \frac{\dot{Q}_{MT}+\dot{Q}_{LT}}{\dot{W}_{Total}} \tag{1}$$

$$\dot{W}_{Total} = \dot{W}_{comp,HT} + \dot{W}_{comp,MT} + \dot{W}_{comp,LT} + \dot{W}_{cond,fans} + \dot{W}_{MT,fans} + \dot{W}_{LT,fans} \tag{2}$$

Where $\dot{Q}_{MT}$ and $\dot{Q}_{LT}$ are the medium and low temperature evaporators cooling loads, respectively. $\dot{W}_{Total}$ is the total power input, $\dot{W}_{comp,HT}$ is the R717 compressor power consumption, $\dot{W}_{comp,MT}$ and $\dot{W}_{comp,LT}$ are the R744 medium and low temperature compressor power consumptions, respectively. $\dot{W}_{cond,fans}$, $\dot{W}_{MT,fans}$, and $\dot{W}_{LT,fans}$ are the R717 condenser, medium temperature evaporator and low temperature evaporator fans consumptions, respectively.

The MT and LT evaporators cooling loads of the cascade system are defined as follows:

$$\dot{Q}_{MT} = \dot{m}_6 h_6 \text{-} \dot{m}_5 h_5 \tag{3}$$

$$\dot{Q}_{LT} = \dot{m}_{13}h_{13}\text{-}\dot{m}_{12}h_{12} \tag{4}$$

Where $\dot{m}$ and h stand for the mass flow rate and enthalpy of the refrigerant. The power input of the compressors can be approached as:

$$\dot{W}_{Comp} = \frac{\dot{m}(h_{out,s} - h_{in})}{\eta_{comp}} \tag{5}$$

Where η_{comp} is the total isentropic efficiency of the compressor while $(h_{out,s}$ - $h_{in})$ is the enthalpy difference representing an isentropic work. The efficiencies of the R717 and R744 compressors were estimated employing the correlations (5) and (6) which were reported by Mosaffa et al. (2016) and Lee et al. (2006), respectively.

$$\eta_{comp,R717} = 0.0071{P_R}^5 - 0.1264{P_R}^4 + 0.9023{P_R}^3 - 3{,}2277{P_R}^2 + 5.7871P_R - 3.3429, \text{ for } P_R<4.3 \tag{6}$$

$$\eta_{comp,R744} = 0{,}00476{P_R}^2 - 0{,}09238P_R + 0{,}89810 \tag{7}$$

Where P_R is the pressure ratio of the compressor. Table 2 shows the energy balances for each one of the components of the cascade system.

Table 2: Energy balances for the R717/R744 cascade refrigeration system.

Components	Energy Balances
	R717 system
Condenser	$\dot{Q}_{Cond} = \dot{m}_{18}h_{18} - \dot{m}_{19}h_{19}$
Expansion Valve	$\dot{m}_{20}h_{20} = \dot{m}_{19}h_{19}$
Compressor	$\dot{W}_{MT} = (\dot{m}_{18}h_{18,S} - \dot{m}_{17}h_{17})/\eta_{comp,R717}$
Cascade heat exchanger	$\dot{m}_2h_2 + \dot{m}_{20}h_{20} = \dot{m}_3h_3 + \dot{m}_{17}h_{17}$
	R744 system
MT Expansion Valve	$\dot{m}_5h_5 = (\dot{m}_4 - \dot{m}_{10})h_4$
MT Evaporator	$\dot{Q}_{MT} = \dot{m}_6h_6 - \dot{m}_5h_5$
MT Compressor	$\dot{W}_{MT} = (\dot{m}_2h_{2,S} - \dot{m}_1h_1)/\eta_{comp,R744}$
LT Heat Exchanger	$\dot{m}_{10}h_{10} + \dot{m}_{13}h_{13} = \dot{m}_{11}h_{11} + \dot{m}_{14}h_{14}$
LT Expansion Valve	$\dot{m}_{12}h_{12} = \dot{m}_{11}h_{11}$
LT Evaporator	$\dot{Q}_{LT} = \dot{m}_{13}h_{13} - \dot{m}_{12}h_{12}$
LT Compressor	$\dot{W}_{LT} = (\dot{m}_{15}h_{15,S} - \dot{m}_{14}h_{14})/\eta_{comp,R744}$

3.2. Exergy balances

The exergy analysis, which is based on the second law of the thermodynamics, is applied to thermal systems in order to quantify reversibility rates and identify the critical components of the system that need to be improved.

The energetic efficiency of the proposed R717/R744 cascade refrigeration system applying the second law of the thermodynamics can be expressed as follows:

$$\eta_{ex} = \frac{\dot{E}_{out}}{\dot{E}_{in}} = 1 - \frac{\dot{I}_D}{\dot{E}_{in}} \tag{8}$$

Where $\dot{E}_{out}$ is the total inlet exergy, $\dot{E}_{in}$ is the total outlet exergy. $\dot{I}_D$ stands for the total exergy destruction or irreversibility. $\dot{E}_{in}$ and $\dot{I}_D$ can be approached as follows:

$$\dot{E}_{in} = \dot{W}_{Total} \tag{9}$$

$$\dot{I}_D = \dot{E}_{in} - \dot{\mathrm{E}}_{\mathrm{out}} = \dot{I}_{Cond} + \dot{I}ev_{R717} + \dot{I}comp_{R717} + \dot{I}casc_{HE} + \dot{I}ev_{Rec} + \dot{I}ev_{MT_{R744}} + \dot{I}e_{MT} + \dot{I}_{LTHE} + \dot{I}ev_{LTR744} + \dot{I}e_{LT} + \dot{I}comp_{LTR744} + \dot{I}comp_{MTR744} \tag{10}$$

Where $\dot{I}_{Cond}$ is the exergy destruction in the condenser, $\dot{I}ev_{R717}$ is the exergy destruction in the R717 expansion valve, $\dot{I}comp_{R717}$ is the exergy destruction in the R717 compressor, $\dot{I}casc_{HE}$ is the exergy destruction in the cascade heat exchanger, $\dot{I}ev_{Rec}$ is the exergy destruction in the expansion just after de receiver, $\dot{I}ev_{MT_{R744}}$ is the exergy destruction in the MT expansion valve, $\dot{I}e_{MT}$ is the exergy destruction in the MT evaporator, $\dot{I}_{LTHE}$ is the exergy destruction in the LT heat exchanger, $\dot{I}ev_{LTR744}$ is the exergy destruction in the LT expansion valve, $\dot{I}e_{LT}$ is the exergy destruction in the LT evaporator, and $\dot{I}comp_{LTR744}$ is the exergy destruction in the LT compressor, and finally $\dot{I}comp_{MTR744}$,is the exergy destruction in the MT compressor. The irreversibility in each one of the components was calculated as summarized in Table 3.

Table 3: Irreversibility of each component of the R717/R744 cascade refrigeration system.

Components	Exergy Balances
	R717 system
Condenser	$\dot{I}_{Cond} = \dot{m}_{18}(h_{18} - h_{19} - T_{amb,K}(s_{18} - s_{19}))$
Expansion Valve	$\dot{I}ev_{R717} = \dot{m}_{19}T_{amb,K}(s_{20} - s_{19})$
Compressor	$\dot{I}comp_{R717} = \dot{m}_{17}T_{amb,K}(s_{18} - s_{17})$
Cascade heat exchanger	$\dot{I}casc_{HE} = T_{amb,K}(\dot{m}_{17}(s_{17} - s_{20}) + (\dot{m}_1(s_3 - s_2))$
	R744 system
MT Expansion Valve	$\dot{I}ev_{MT} = \dot{m}_5T_{amb,K}(s_5 - s_4)$
MT Evaporator	$\dot{I}e_{MT} = \dot{m}_5T_{amb,K}(s_6 - s_5 + (h_5 - h_6)/T_{a,MT,K}))$
MT Compressor	$\dot{I}comp_{MT} = \dot{m}_0T_{amb,K}(s_2 - s_1)$
LT Heat Exchanger	$\dot{I}_{LTHE} = T_{amb,K}\dot{m}_{11}(s_{11} - s_{10} + s_{14} - s_{13})$
LT Expansion Valve	$\dot{I}ev_{LT} = \dot{m}_{12}T_{amb,K}(s_{12} - s_{11})$
LT Evaporator	$\dot{I}e_{LT} = \dot{m}_{12}T_{amb,K}(s_{13} - s_{12} + (h_{13} - h_{12})/T_{a,LT,K}))$
LT Compressor	$\dot{I}comp_{LT} = \dot{m}_{15}T_{amb,K}(s_{15} - s_{14})$

4. RESULTS AND DISCUSSION

In this section, the main results extracted from the first and second law thermodynamic analysis applied to an R717/R744 cascade refrigeration system are presented and discussed. The R717/R744 cascade refrigeration system include two different direct expansion loads which is representative for food retail applications. Firstly, the coefficient of performance and total power consumption supplied to the R717/R744 cascade refrigeration system at different ambient temperatures are estimated. Then, the second law efficiency of the present configuration is approached together with the irreversibility rates per component. The operating conditions of interest for modelling of the R717/R744 cascade refrigeration system are summarises in Table 4. The cooling loads of the MT and LT evaporators were set to 100 and 30 kW, respectively, as assumed by Tsamos et al. (2017), considering a supermarket total sale area of 1400 m^2.

Table 4: Thermodynamic conditions assumed in the modelling.

Parameters	Values
Ambient Temperature, °C	7 - 40
Atmospheric Pressure, kPa	101,3
Condenser approach temperature, °C	5
MT Evaporator Cooling Load, kW	100
LT Evaporator Cooling Load, kW	30
MT Evaporating Temperature, °C	-8
LT Evaporating Temperature, °C	-32
Cascade HX Temperature Difference, $T_{20} – T_3$, °C	5
Air Temperature at the MT Evaporator, $T_{a,MT}$, °C	5
Air Temperature at the LT Evaporator, $T_{a,LT}$, °C	-18
Superheating at the outlet of the MT and LT Evaporators, °C	0
LT Heat Exchanger Effectiveness	0,4

4.1. Energy analysis

Table 5 shows the thermodynamic state conditions obtained from the modelling of the described R717/R744 cascade refrigeration system. Table 5 also includes the coefficient of performance of the cascade system (COP cascade), heat released through the condenser (Q cond), the heat rates exchanged at the cascade (Q cascade) and LT heat exchangers (Q LTHX), the power input of the R717 compressor (W comp R717) as well as the power input of the MT (W MTcomp) and LT (W LTcomp) R744 compressors. Results presented in Table 5 correspond to those obtained at an ambient temperature of 25 ºC.

Table 5 indicates that at an ambient temperature of 25 ºC, the coefficient of performance of the cascade cycle is 1.86 while the total power consumption was 69.38 kW. It can be noted that the R717 compressor consumes around 33.51 % of the total power energy supplied to the system while it is 21.54.% for the MT compressor. Table 5 also shows that for the given MT cooling load of 100 kW, que required mass flow rate is around 0.4722 kg.s^{-1}. In the case of the LT evaporator and for the given cooling load of 30 kW, que required mass flow rate is around 0.1297 kg.s^{-1}. Regarding the R717 cycle, a mass flow rate of 0.1313 kg.s^{-1} is required to dissipate 151.1 kW from the R744 system thought the cascade heat exchanger.

Table 5: Thermodynamic state conditions of the R717/R744 cascade refrigeration system at an ambient temperature of 25°C.

State	P (kPa)	T (°C)	m (kg.s^{-1})	h (kJ.kg^{-1})	s (kJ.kg^{-1}.K^{-1})
1	2803	2.07	0.6019	-57.8	-0.797
2	4380	38.82	0.6019	-33.0	-0.778
3	4380	8.89	0.6019	-284.1	-1.661
4	4380	8.89	0.6019	-284.1	-1.661
5	2803	-8.00	0.4722	-284.1	-1.650
6	2803	-8.00	0.4722	-72.3	-0.851
9	2803	-8.00	0.4722	-72.3	-0.851
10	4380	8.89	0.1297	-284.1	-1.661
11	4380	2.49	0.1297	-301.4	-1.723
12	1334	-32.00	0.1297	-301.4	-1.690
13	1334	-31.99	0.1297	-70.1	-0.731
14	1334	-15.64	0.1297	-52.8	-0.661
15	2803	47.54	0.1297	-5.1	-0.620
17	496	13.89	0.1313	1493.0	5.665
18	1351	100.70	0.1313	1674.0	5.757
19	1351	30.00	0.1313	341.8	1.488
20	496	3.89	0.1313	341.8	1.512

Q cond (kW)	174,9	W_comp R717 (kW)	23,76
Q_cascade (kW)	151,1	W_MTcomp (kW)	14,95
Q_LTHX (kW)	2,25	W_LTcomp (kW)	6,17
COP_cascade	1,86	W_Total (kW)	69,88

Figure 3 shows the coefficient of performance (COP) of the R717/R744 cascade refrigeration system and the total power input of the system at ambient temperatures ranging from 7 to 40 ºC. In obtaining these results, other operating parameters such as the MT and LT evaporators cooling loads and evaporating temperature are kept constant. It is observed that at the given operating conditions, the coefficient of performance of the cascade refrigeration system decreases from 2.19 to 1.56 (decrease of 29%) when the ambient temperature varies from 7 ºC to 40 ºC. Since the cooling loads at the MT and LT evaporators are set constant, the decrease in the COP is basically due to the increase in the total power input. In addition, the R717 compressor appears as the most affected component when the ambient temperature is higher, thus causing the mayor effect on increment in the total power input to run the cascade system. This is due to an increase in the R717 flow rate and vapour pressure at the outlet of the R717 compressor as a result of the ambient temperature variation. In the case of the R744 side, the pressure of the fluid through the cascade heat exchanger is not affected by the high ambient temperatures which represent an advantage for the cascade configuration since the R744 keeps operating at subcritical conditions without significant variations.

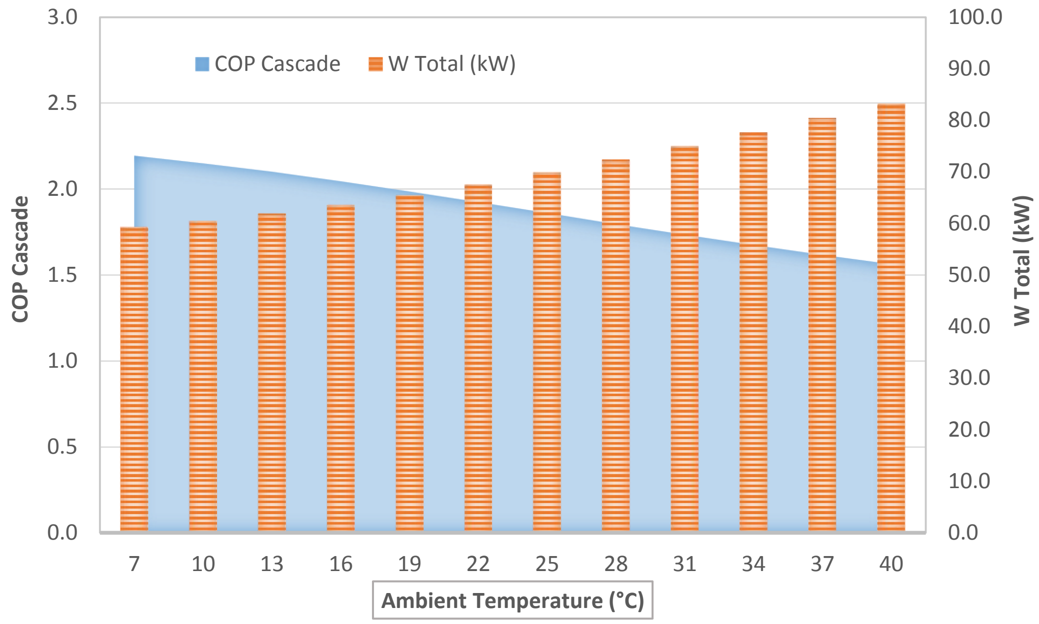

Figure 3: Effect of the ambient temperature on the COP of the cascade system and total power to the system.

4.2. Exergy analysis

Table 6 depicts the exergy destruction rates per component obtained from the modelling of the R717/R744 cascade refrigeration system. Table 6 also includes second law efficiency of the cascade system. Modelling results in Table 6 correspond to those obtained at an ambient temperature of 25 ºC and operating conditions described in Table 4.

Table 6: Exergy destructions at the R717/R744 cascade refrigeration system at an ambient temperature of 25°C.

Components	[kJ.s^{-1}]	Percentage %
R717 system		
Condenser	7.804	24.42
Expansion Valve	0.9277	2.903
Compressor	3.581	11.21
Cascade heat exchanger	4.238	13.26
R744 system		
MT Expansion Valve	1.574	4.923
MT Evaporator	5.256	16.44
MT Compressor	3.375	10.56
LT Heat Exchanger	0.2882	0.9016
LT Expansion Valve	1.272	3.98
LT Evaporator	2.035	6.367
LT Compressor	1.61	5.037
2nd Law Efficiency of the Cycle		0.543

Table 6 indicates that at an ambient temperature of 25 ºC, the highest exergy destruction rate in the R717/R744 cascade refrigeration system is at the condenser with 7.804 kJ.s^{-1} (24.42 % of the total exergy destruction rate), followed by the MT evaporator with 5.256 kJ.s^{-1} (16.44 %), the cascade heat exchanger with 4.238 kJ.s^{-1} (13.26 %), the R717 compressor with 3.581 kJ.s^{-1} (11.21 %), and the MT R717 compressor with 3.375 kJ.s^{-1} (10.56 %). The high exergy destruction rate at the condenser if compared to the other components is mainly associated with the heat transfer between the condenser and its high temperature condensing medium. In the case of the evaporator, it is mainly associated with the superheating level, and the heat transfer between the evaporation process and the refrigerated medium. Regarding the compressor, the higher the compressor outlet temperature the higher the exergy destruction rate is. The modelling results evidence that the condenser, the MT evaporator, the cascade heat exchanger, the R717 compressor, and the MT R717 compressor are the critical components of the system that can be improved in order to increase the whole cascade refrigeration system efficiency.

Furthermore, Figure 4 shows the second law efficiency and the exergy destruction percentage per component at ambient temperatures ranging from 7 to 40 ºC. The modelling results indicate that the second law efficiency increases from 0.494 to 0.544 when the ambient temperature goes from 7 to 31 ºC. Then, the efficiency slightly decreases to 0.542 for an ambient temperature of 40 ºC. This effect is obtained since the total exergy destruction rate increment was slightly lower than that of the required energy input when the ambient temperature increases from 7 to 31 ºC. For an ambient temperature from 31 to 40 ºC, the total exergy destruction rate increment was slightly higher than that of the required energy input (see equation 8). It can also be observed that the condenser is the most critical component of the present configuration. The exergy destruction rate at the condenser increases from 6.68 to 9.89 $kJ.s^{-1}$ when the ambient temperature is varied from 7 to 40 ºC and reaching the maximum values from temperatures above 34 ºC. Figure 4 also evidences that the exergy destruction at the R717 compressor decreases from 4.74 $kJ.s^{-1}$ to 3.36 $kJ.s^{-1}$ when the ambient temperature goes from 7 to 19 ºC, and then it increases from 3.36 to 5.1 $kJ.s^{-1}$ from an ambient temperature of 22 ºC to 40 ºC. The high exergy destruction found at the compressor for the lowest ambient temperatures is due to very low isentropic efficiencies found at these ambient conditions and the high entropy values obtained at the compressor outlet at these low temperatures. It can also be observed that the R717 expansion valve presents a significant increase in exergy destruction from 0.09 to 2.4 $kJ.s^{-1}$ when the ambient temperature increases. In the case of the other components below the cascade heat exchanger, it is noted that the exergy destruction keeps almost constant as it is also observed in the operating conditions of the R744 cycle.

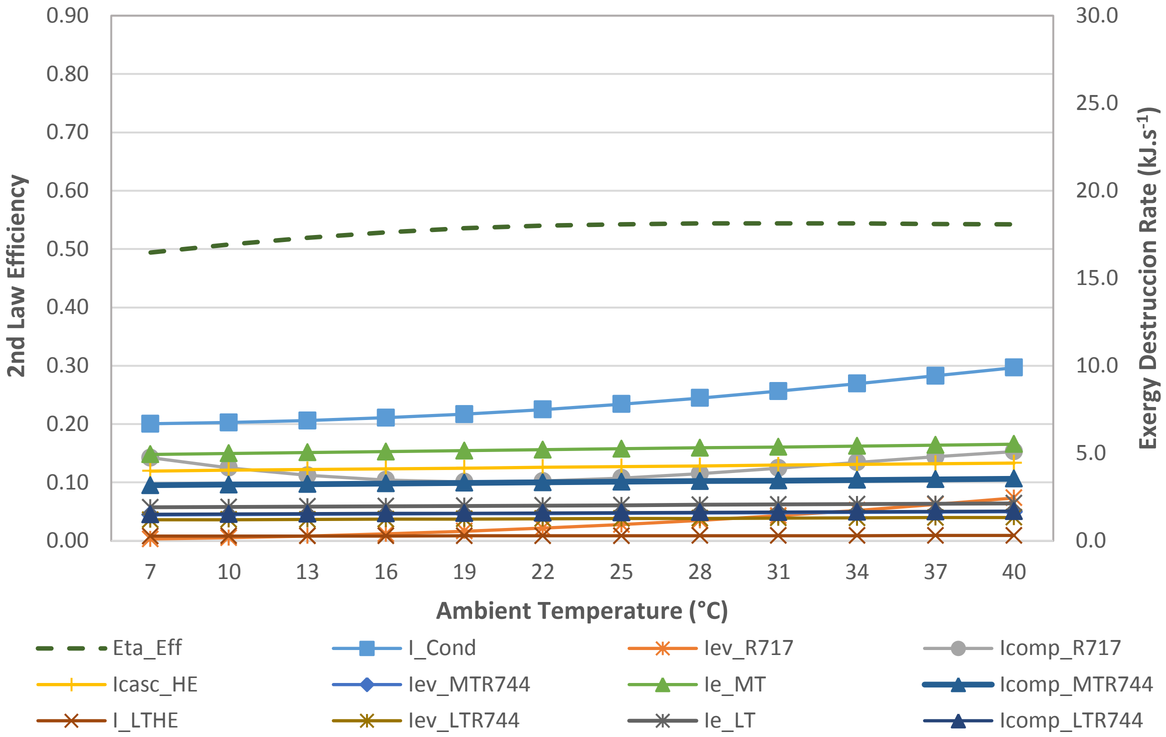

Figure 4: Effect of the ambient temperature on the 2nd law efficiency of the system and exergy destruction percentage per component.

5. CONCLUSIONS

In this paper, a thermodynamic analysis based on the first and second law was performed to assess an R717/R744 cascade refrigeration system involving two different direct expansion loads. It included the estimation of the coefficient of performance and irreversibility of the R717/R744 cascade refrigeration system at ambient temperatures ranging from 8 to 40 ºC representing moderate and warm weather conditions.

The main conclusions from the present study can be drawn as follows.

• It was found that at an ambient temperature of 25 ºC the coefficient of performance of the analysed R717/R744 cascade refrigeration system was 1.86 and that the most energy consuming component was the R717 compressor.

• It was also found that the coefficient of performance of the analysed R717/R744 cascade refrigeration system decreased from 2.19 to 1.56 when the ambient temperature varied from 7 ºC to 40 ºC.

• The second law efficiency of the analysed R717/R744 cascade refrigeration system varied from 0.494 to a maximum of 0.544 between ambient temperatures of 7 and 31 ºC, respectively.

• The amount of exergy destruction in the condenser, MT evaporator, cascade heat exchanger and the R717 compressor accounts for large percentage in the total exergy destruction.

• The present study also showed that the condenser operating conditions have significant effects on system COP and second law efficiency.

• Further studies will include the validation of the present R717/R744 cascade refrigeration system and the comparison with others proposed cycles in the literature in order to assess the practical usefulness of this system for food retail applications especially at warm climates.

6. REFERENCES

ARORA A., 2016. Effects of Variation in Evaporator and Condenser Temperature on Cascade Condenser Temperature, COP and Second Law Efficiency of a Cascade Refrigeration System. *International Journal of Advance Research and Innovation*, 4, 261-273

BS EN 378-1:2016, Refrigerating systems and heat pumps – Safety and environmental requirements. Part 1: Basic requirements, definitions, classification and selection criteria. ISBN 978 0 580 84660 1

EES, 2014. Engineering Equation Solver, version 9.810, www.fChart.com

KILICARSLAN A., HOSOZ M., 2010. Energy and irreversibility analysis of a cascade refrigeration system for various refrigerant couples. *Energy Conversion and Management*, 51, 2947–2954.

LEE T.S., LIU C.H., CHEN T.W., 2006. Thermodynamic analysis of optimal condensing temperature of cascade refrigeration systems*. International Journal of Refrigeration,* 29, 1100-1108

MA M., YU J., WANG X., 2014. Performance evaluation and optimal configuration analysis of a CO_2/NH_3 cascade refrigeration system with falling film evaporator–condenser. *Energy Conversion and Management,* 79, 224–231.

MESSINEO A., 2012. R744-R717 Cascade Refrigeration System: Performance Evaluation compared with a HFC Two-Stage System. *Energy Procedia*, 14, 56-65.

MOSAFFA A.H., GAROUSI FARSHI L., INFANTE FERREIRA C.A., ROSEN M.A., 2016. Exergoeconomic and environmental analyses of CO_2/NH_3 cascade refrigeration systems equipped with different types of flash tank intercoolers. *Energy Conversion and Management,* 117, 442-453.

NASRUDDIN, SHOLAHUDIN S., GIANNETTI N., ARNAS, 2016. Optimization of a cascade refrigeration system using refrigerant C3H8 in high temperature circuits (HTC) and a mixture of C2H6/CO2 in low temperature circuits (LTC). *Applied Thermal Engineering* 104 (2016) 96–103.

REZAYAN O., BEHBAHANINIA A., 2011. Thermoeconomic optimization and exergy analysis of CO_2/NH_3 cascade refrigeration systems. *Energy,* 36, 885-895.

SAWALHA S., 2008. Theoretical evaluation of trans-critical CO_2 systems in supermarket refrigeration. Part I: Modelling, simulation and optimization of two system solutions. *International Journal of Refrigeration, 31*, 516-524.

SUN Z., LIANG Y., LIU S., JI W., ZANG R., LIANG R., GUO Z., 2016. Comparative analysis of thermodynamic performance of a cascade refrigeration system for refrigerant couples R41/R404A and R23/R404A. *Applied Energy*, 184, 19–25.

TSAMOS K.M., GE Y.T., IDEWA SANTOSA, TASSOU S.A., BIANCHI G., MYLONA Z., 2017, Energy analysis of alternative refrigeration system configurations for retail food applications in moderate and warm climates. *Energy Conversion and Management*, DOI: https://doi.org/10.1016/j.enconman.2017.03.020.