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A COMPARATIVE INVESTIGATION OF NATURAL REFRIGERANTS: A CASE STUDY FOR COLD STORAGE APPLICATION

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Abstract

The aim of this study is to investigate the thermodynamic performance of a refrigeration system for a cold storage using natural refrigerants which can be considered as substitutes of CFC, HCFC and HFC type working fluids. As these kinds of refrigerants menace the environment because of their higher ozone depletion and global warming potentials, the investigations are focused on new alternative refrigerants. For this purpose, seven natural refrigerants and three chlorine based ones are analyzed in terms of first and second laws of thermodynamic. The analyses are made for a refrigeration capacity of 98.46 kW and cold room temperature of -2 °C. The energy performance indicator COP, exergy performance indicator exergy efficiency and exergy destruction rates of the system are determined. The results are showed that the best COP value is obtained using R600a with a value of 2.5. According to the exergy analyses, the refrigerant R600a has the highest exergetic efficiency rate with 24.9 %. Also the highest exergy destruction rates occurred using R170 and R744 with 57.47 and 56.63 kW, respectively.

Keywords: Natural Refrigerant, Cold Storage, Energy, Exergy.

DOĞAL SOĞUTUCU AKIŞKANLARIN KARŞILAŞTIRMALI OLARAK İNCELENMESİ: SOĞUK DEPOLAMA UYGULAMASI İÇİN ÖRNEK BİR ÇALIŞMA

Özet

Bu çalışmanın amacı, doğal akışkanlar kullanılan bir soğutma sisteminin soğuk depolama uygulamaları için termodinamik performansının incelenmesidir. CFC, HCFC ve HFC tipi akışkanlar, yüksek ozon tahrip ve küresel ısınma potansiyeline sahip oldukları için araştırmalar yeni alternatif akışkanlara odaklanmıştır. Bu amaçla, yedi adet doğal akışkan ve üç adet klasik akışkan, termodinamiğin birinci ve ikinci kanunu yönünden analiz edilmiştir. Analizle, 98.46 kW kapasitesinde ve -2 °C muhafaza sıcaklığındaki bir soğuk oda için yapılmıştır. Enerji performans göstergesi olan COP, ekserji performansı göstergesi olan ekserji verimi ve ekserji kayıpları tespit ediliştir. Sonuçlara göre en yüksek COP değer R600a akışkanı kullanıldığında 2.5 olarak elde edilmiştir. Ekserji analizi sonuçlarına göre ise, en yüksek ekserji verimi R600a akışkanı için % 24.9 olarak elde edilmiş ve en yüksek ekserji yıkımı değerleri R170 ve R744 akışkanları için 57.47 ve 56.63 kW olarak hesaplanmıştır.

Anahtar kelimeler: Doğal Akışkan, Soğuk Depolama, Enerji, Ekserji.

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Nomenclature

e	specific exergy (kJ/kg)
$\dot{E}x$	exergy rate (kW)
h	specific enthalpy (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
$\dot{Q}$	heat energy rate (kW)
P	Pressure (kPa)
s	specific entropy (kJ/kgK)
$\dot{S}_{gen}$	entropy generation rate (kW/K)
T	temperature (°C)
$\dot{W}$	work rate (kW)

Greek letters

η_{ex}	exergy efficiency
η_{is}	isentropic efficiency of compressor
η_{mec}	mechanical efficiency of compressor
η_{elec}	Electrical efficiency of compressor

Subscripts

0	reference state
Comp	compressor
CR	cold room
CT	cooling tower
el	electricity
in	inlet
out	outlet
P	pump
SH	superheating
SC	subcooling
W	water

Acronyms

ODP	ozone depleting potential
GWP	global warming potential
HCFC	hydrochlorofluorocarbon
CFC	chlorofluorocarbon
HFC	hydrofluorocarbon
COP	coefficient of performance
EG	ethylene-glycol

1. Introduction

As the refrigeration technology plays a significant role in today's life, ozone depletion and global warming are two major environmental concerns with serious implications for the future development of the refrigeration-based industries (Bolaji and Huan, 2013). The ozone depleting potential (ODP) and global warming potential (GWP) have become one of the most important criteria in analyzing new alternatives to CFC and HCFC refrigerants in vapor-compression refrigeration systems. The general consensus regarding ODP is that free chlorine radicals remove ozone from the atmosphere, and later, chlorine atoms continue to convert more ozone to oxygen. The presence of chlorine in the stratosphere is the result of the migration from chlorine containing chemicals (da Silva et al., 2011; Akash and Said, 2003). As the global environment is progressively concerned, greenhouse warming has also become very important these days besides the ozone depletion issue. One of the ways to alleviate greenhouse warming is to adopt some new refrigerants with low GWP and develop high efficiency refrigeration and heat pump devices (Yu et al., 2010).

CFC and HCFC type refrigerants have been used predominantly in refrigeration systems for the past few decades. Since the effects of these refrigerants to the environment form undesirable situations, it was decided to phase out CFC and HCFC type refrigerants eventually and the regulation for production begun from 1996 in the developed countries by Montreal protocol (Park et al., 2007; Kabul et al., 2008) As a result of these regulations, many non-ozone depleting mixtures and new refrigerants have been developed and applied in actual systems (Park and Jung, 2007). From this point of view, refrigerants which have zero or lower ODP and GWP values - such as natural ones-can be possible alternatives for CFCs and HCFCs for refrigeration applications. Natural refrigerants are non-synthetic substances that occur in nature's material cycles and which can and should be used as the cooling agents in refrigeration systems. These substances have been used as refrigerants for years but are only now beginning to replace F-gases in certain applications (Shecco, 2015).

A literature survey about the utilization of natural refrigerants shows that there are significant developments on refrigeration system working with alternative fluids. He et al. (2014), analyzed the use of natural refrigerant propane (R290) and its mixtures substituting for R134a in a large capacity chest freezer. They examined theoretically and experimentally the application possibilities of the refrigerants. Bolaji and Huan (2013), made a review on ozone depletion and global warming situation of natural refrigerants. They also analyzed the potentials of various natural refrigerants and their areas of application in refrigeration and air-conditioning systems. In their results, they concluded that the natural refrigerants found to be the most suitable long time alternatives in refrigeration and air-conditioning systems. Cecchinato et al. (2012) investigated the energy performance of a supermarket refrigeration and air conditioning integrated system working with natural refrigerants. In their study, they analyzed the systems working with ammonia, propane and carbon dioxide. Dubey et al. (2014), analyzed the use of natural refrigerant propylene (R1270) for transcritical cascade refrigeration system. They evaluated the thermal performance of the cycle for different combinations of design and operating variables. Lorentzen (1995), investigated the use of natural refrigerants for solution to the CFC/HCFC type refrigerants. He discussed the advantages of some of natural refrigerants such as ammonia, propane and carbon dioxide used in refrigeration and heat pump applications, using conventional compressor systems. Li et al. (2014), investigated the energy and exergy performance of secondary loop systems using HFC-152a and HC-290 in automotive air-conditioning systems. They found that the COP was increased by 8% to 15% with the use of HC-290 instead of HFC-134a. Rasti, et al. (2013), investigated a feasibility study of substitution of two hydrocarbon refrigerants namely R436A and R600a, instead of R134a in a domestic refrigerator. They declared that with the use of R436A and R600a, the energy consumption was reduced about 14% and 7%, respectively in comparison to R134a. Joybari et al. (2013), carried out exergy analysis to investigate the performance of a domestic refrigerator for R134a and R600a. They also applied Taguchi method to design experiments to minimize exergy destruction while using R600a. Yan et al. (2015a), analyzed a modified vapor-compression refrigeration cycle operating with the zeotropic mixture R290/R600a for domestic refrigerator-freezers. They carried out a theoretical energy and exergy analysis on the performance of the system by using the developed mathematical model, and then compared with that of the traditional vapor-compression refrigeration cycle operating with the refrigerant R600a and the zeotropic mixture R290/R600a, respectively. In their other study (2015b) they investigated an internal auto-cascade refrigeration cycle operating with the zeotropic mixture of R290/R600a or R290/R600 for domestic refrigerator-freezers. Mafi et al. (2009) examined an exergy analysis for multistage cascade low temperature refrigeration systems used in olefin plants. They utilized propylene refrigeration at several temperature levels to cool and heat the feed in the initial fractionation sections of the plant. Sivakumar and Somasundaram (2014) studied three stage auto refrigerating cascade system for the existence using two combinations of (R290/R23/R14, R1270/R170/R14) three component zeotropic mixture of five different refrigerants. They utilized exergy analysis method for three stage system. Padilla et al. (2010), dealt an exergy analysis of the impact of direct replacement (retrofit) of R12 with the zeotropic mixture R413A on the performance of a domestic vapour-compression refrigeration system. They evaluated the parameters and factors affecting the performance of both refrigerants using an exergy analysis.

In this study, the performance of a cold storage facility is investigated for seven natural refrigerants as working fluids. These refrigerants are selected to be substitutes of R12 (CFC), R22 (HCFC) and R134a (HFC), namely, ethane (R170), propylene (R1270), propane (R290), butane (R600), isobutene (R600a), ammonia (R717) and carbon dioxide (R744). Also analyses are carried out for R12, R22 and R134a in order to figure out the performance of the natural refrigerants comparatively. In order to determine the energetic and exergetic performance of the cold storage facility, first and second law of thermodynamics are applied to the system.

2. Natural Refrigerants and Description of Case Study

Natural refrigerants are naturally occurring substances, such as hydrocarbons (propane, iso-butane), CO_2, ammonia, water and air. These substances can be used as cooling agents (heat transfer medium) in refrigerators and air conditioners, don't harm the ozone layer and have no or negligible climate impact (Refrigerants Naturally, 2015). These kinds of fluids have been used as refrigerants for many years, however, they are now finding their way into applications where previously fluorocarbons were the preferred option (AGDE, 2015).They are now being used more extensively due to their low impact on the environment (Linde, 2015). From an economic perspective, these refrigerants are inexpensive, in some cases even cheaper than HFCs. Also, natural refrigerants are extremely energy-efficient, sometimes up to 40% more than HFCs (Shecco, 2015).

As declared previously, the analyses are made for seven natural refrigerants and three chlorine based refrigerants. The general properties of these refrigerants are given in Table 1 (ASHRAE 2004; Restrepo et al., 2008; Calm and Hourahan, 2011). It must be noted that, these properties are obtained from the references given above while the properties of critical temperature and pressure values are obtained from EES software (F-Chart, 2015).

Table 1. Physical, safety and environmental properties of investigated the refrigerants

ASHRAE number	Molecular formula	Critical Pressure (kPa)	Critical Temperature (°C)	Safety group	ODP (relative to R11)	GWP (relative to CO_2)	Atmospheric life time (year)
R170	CH_3CH_3	4872.2	32.172	A3	0	20	0.21
R1270	$CH_3CH{=}CH_2$	4664.6	92.420	A3	0	3	0.001
R290	$CH_3CH_2CH_3$	4247.1	96.675	A3	0	20	0.041
R600	$CH_3CH_2CH_2CH_3$	3796.0	151.975	A3	0	20	0.018
R600a	$CH(CH_3)_3$	3640.0	134.667	A3	0	20	0.019
R717	NH_3	11333	132.25	B2	0	0	0.25
R744	CO_2	7377.3	30.98	A1	0	1	120
R12	CCl_2F_2	4114.1	112.017	A1	0.82	10720	100
R22	$CHClF_2$	4988.5	96.130	A1	0.05	1780	12
R134a	CH_2FCF_3	4059.0	101.030	A1	0.000015	1410	14

For the case study, the cold storage facility consist of a vapor compression refrigeration system, a cold room to be maintained at -2 °C and a cooling tower for heat rejection process from the condenser (Figure 1). In the vapor compression refrigeration cycle, the refrigerant vapor is assumed to reach the inlet (suction) of the compressor as a superheated vapor, and the refrigerant liquid is assumed to be a subcooled liquid before the inlet of the expansion valve excluding transcritical refrigerants R744 and R170. Since the critical temperatures of these two refrigerants are lower than operating temperatures, both refrigerants are assumed to be cooled down by condenser (gas cooler) to a certain temperature. T-s diagram of the investigated refrigeration system is given in Figure 2 for classic cycle and transcritic cycle.

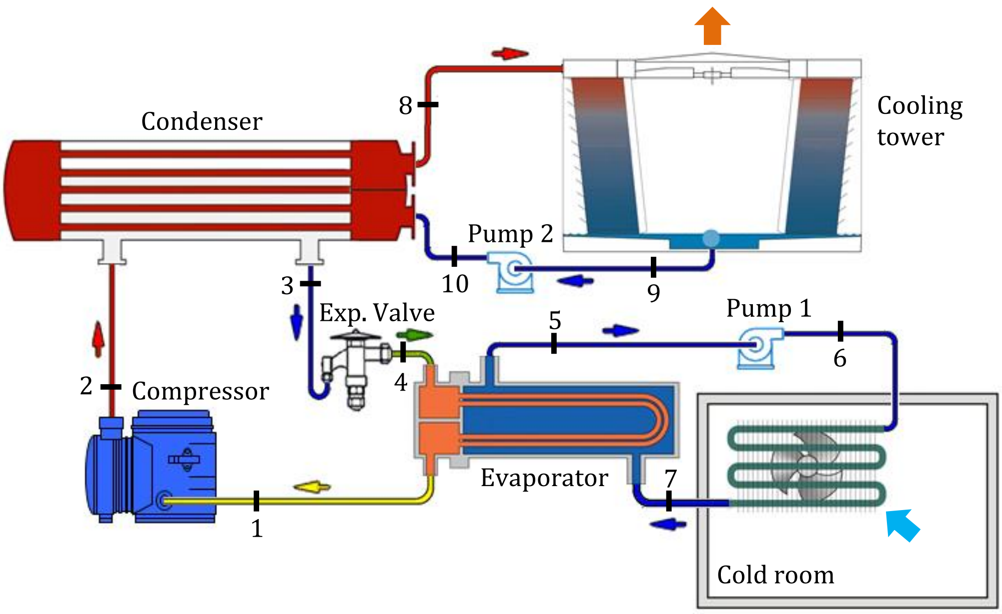

Figure 1. Schematic representation of the cold storage facility

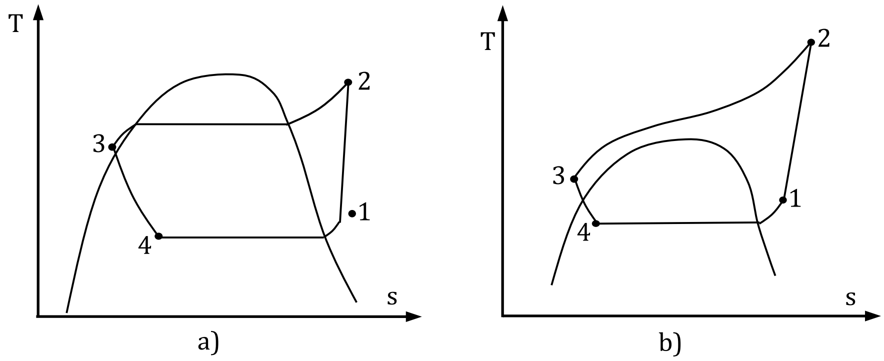

Figure 2. T-s diagram of the vapor compression refrigeration cycle a) classic b) transcritic

The refrigeration load from the cold room is absorbed by a secondary fluid, which is ethylene-glycol (EG) and selected in order to prevent freezing problem. The freezing point of EG is -18.84 °C for the specified concentration. Also the condenser heat load is absorbed by cooling water and the water is cooled down by means of a cooling tower. The operating parameters of the system for the baseline conditions are given in Table 2. It must be noted that for the transcritical refrigerants, namely R170 and R744, temperature of the fluid at the exit of the condenser is taken to be the sum of condenser temperature, and subcooling temperature.

Table 2. Operating parameters of the system for the baseline conditions

Reference state temperature, T_0	25 °C
Reference state pressure, P_0	101.325 kPa
Refrigeration capacity, $\dot{Q}_{CR}$	98.46 kW
Cold room temperature, T_{CR}	-2 °C
Evaporator temperature, T_E	-14 °C
Condenser temperature, T_C	40 °C
Superheating temperature, ΔT_{SH}	4 °C
Subcooling temperature, ΔT_{Sc}	4 °C
Isentropic efficiency of compressor, η_{is}	0.88
Mechanical efficiency of compressor, η_{mec}	0.92
Electrical efficiency of compressor, η_{elec}	0.86
Ethylene -glycol concentration	0.35

3. Governing Balance Equations

The performance characteristics of the vapor compression refrigeration cycle for the cold storage facility are assessed by applying first and second law analysis of thermodynamics. The balance equations are used to determine the work and heat interactions, energy and exergy efficiencies and exergy destruction rates for each system component. The general mass balance equation for a steady-state and steady-flow processes can be written as (Cengel and Boles, 2006)

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

The energy balance equation is defined as:

$$\sum \dot{E}_{in} = \sum \dot{E}_{out} \tag{2}$$

Equation (2) can be written in the form given below:

$$\dot{Q} + \sum \dot{m}_{in} h_{in} = \dot{W} + \sum \dot{m}_{out} h_{out} \tag{3}$$

In above equations, $\dot{m}$ is the mass flow rate, $\dot{E}$ is the rate of energy, $\dot{Q}$ is the rate of net heat, $\dot{W}$ is the rate of net work, and h is the specific the subscripts in and out stand for inlet and outlet respectively.

The Second Law of Thermodynamics overcomes with concepts of entropy and exergy. Exergy analysis of systems allows determining irreversibility and available energy (exergy) in the system. These analyses reveal the efficiency of the systems in terms of First and Second Law of Thermodynamics (Göktürk et al., 2013). For a steady state operation, the general exergy balance equation can be defined as (Dincer and Rosen, 2007)

$$\sum \dot{E}x_{in} = \sum \dot{E}x_{out} + \sum \dot{E}x_{dest} \tag{4}$$

The exergy balance equation can also be written more explicitly as:

$$\dot{E}x_{Q} - \dot{E}x_{W} = \sum \dot{m}_{in} e_{in} - \sum \dot{m}_{out} e_{out} + T_0 \dot{S}_{gen} \tag{5}$$

where, $\dot{E}x_{Q}$ and $\dot{E}x_{W}$ are the exergies of heat and work, respectively, e is the specific exergy, T_0 is the reference state temperature and $\dot{S}_{gen}$ is the entropy generation rate. In above equation, exergies of heat and work and entropy generation rate are given below (Kotas, 1985)

$$\dot{E}x_{dest} = T_0 \dot{S}_{gen} \tag{6}$$

$$\dot{E}x_{Q} = \dot{Q}\left(\frac{T - T_0}{T}\right) \tag{7}$$

$$\dot{E}x_{W} = \dot{W} \tag{8}$$

The specific exergy is expressed relative to the environment conditions as:

$$e = (h - h_0) - T_0(s - s_0) \tag{9}$$

where s is entropy, P is the pressure and the subscript 0 indicates properties at the reference state.

The performance of the vapor compression refrigeration system can be determined using energy and exergy efficiency definitions:

$$COP = \frac{\dot{Q}_{CR}}{\dot{W}_{Comp,el}} \tag{10}$$

$$\eta_{ex} = \frac{\dot{Ex}_{\dot{Q}_{CR}}}{\dot{Ex}_{\dot{W}_{Comp,el}}} \tag{11}$$

where $\dot{Q}_{CR}$ represents cold room refrigeration capacity and $\dot{W}_{Comp,el}$ represents compressor electric consumption.

The governing balance equations are given for all system components in Table 3 according to the reference points shown in Figure 1.

Table 3. Energy, exergy and enropy balance equations for system components.

Component	Energy balance	Exergy balnce	Entropy balance
Compressor	$\dot{W}_{Comp,el} = \frac{\dot{m}_R(h_2-h_1)}{\eta_{el}\,\eta_{mec}}$	$\dot{Ex}_1 + \dot{Ex}_{\dot{W}_{Comp,el}} = \dot{Ex}_2 + \dot{Ex}_{Dest,\dot{W}_C}$	$\dot{S}_1 + \dot{S}_{gen,Comp} = \dot{S}_2$
Condenser	$\dot{Q}_C = \dot{m}_R(h_2 - h_3)$	$\dot{Ex}_2 + \dot{Ex}_{10} = \dot{Ex}_3 + \dot{Ex}_8 + \dot{Ex}_{Dest,C}$	$\dot{S}_2 + \dot{S}_{10} + \dot{S}_{gen,C} = \dot{S}_3 + \dot{S}_8$
Expansion valve	$h_3 = h_4$	$\dot{Ex}_3 = \dot{Ex}_4 + \dot{Ex}_{Dest,ExpV}$	$\dot{S}_3 + \dot{S}_{gen,ExpV} = \dot{S}_4$
Evaporator	$\dot{Q}_E = \dot{m}_R(h_4 - h_1)$	$\dot{Ex}_4 + \dot{Ex}_7 = \dot{Ex}_1 + \dot{Ex}_5 + \dot{Ex}_{Dest,E}$	$\dot{S}_4 + \dot{S}_7 + \dot{S}_{gen,E} = \dot{S}_1 + \dot{S}_5$
Pump 1	$\dot{W}_{P1} = \dot{m}_{EG}(h_6 - h_5)$	$\dot{Ex}_5 + \dot{Ex}_{\dot{W}_{P1}} = \dot{Ex}_6 + \dot{Ex}_{Dest,\dot{W}_{P1}}$	$\dot{S}_5 + \dot{S}_{gen,\dot{W}_{P1}} = \dot{S}_6$
Cold room	$\dot{Q}_{CR} = \dot{m}_{EG}(h_7 - h_6)$	$\dot{Ex}_6 = \dot{Ex}_{\dot{Q}_{CR}} + \dot{Ex}_7 + \dot{Ex}_{Dest,CR}$	$\dot{S}_6 + \dot{S}_{gen,CR} = \frac{\dot{Q}_{CR}}{T_{CR}} + \dot{S}_7$
Cooling tower	$\dot{Q}_{CT} = \dot{m}_W(h_8 - h_9)$	$\dot{Ex}_8 = \dot{Ex}_{\dot{Q}_{CT}} + \dot{Ex}_9 + \dot{Ex}_{Dest,CT}$	$\dot{S}_8 + \dot{S}_{gen,CT} = \frac{\dot{Q}_{CT}}{T_{CR}} + \dot{S}_9$
Pump 2	$\dot{W}_{P2} = \dot{m}_W(h_{10} - h_9)$	$\dot{Ex}_{10} + \dot{Ex}_{\dot{W}_{P2}} = \dot{Ex}_9 + \dot{Ex}_{Dest,\dot{W}_{P2}}$	$\dot{S}_{10} + \dot{S}_{gen,\dot{W}_{P2}} = \dot{S}_9$

4. Results and Discussion

In order to simulate the vapor compression refrigeration cycle for natural refrigerants the following assumptions are made:

- All operations are steady state and steady flow.
- Pressure losses through pipelines are neglected.
- Heat losses and heat gains from or to the system are neglected.
- The changes in potential and kinetic energies are neglected.
- The pump operations are adiabatic and isentropic.
- The directions of heat transfer to the system and work transfer from the system are taken positive.

Using the balance equations, and under the assumptions given above, the analyses are performed for different natural refrigerants using EES software (F-Chart, 2015). The results of thermodynamic analyses of the refrigeration system for the given cooling load are given in Table 4 for all refrigerants. The table is divided into two parts, natural refrigerants and chlorine based refrigerants. It can be seen from the table that the best COP value is obtained using R600 followed by R717, R600a and R290 in natural refrigerants. The performances of these refrigerants are very similar to that of R12, R22 and R134a. Also the trend of exergy efficiency is the same as COP. For the exergy destruction rates, the highest destruction is occurred using R170 and R744 for the given refrigeration duty. Furthermore, the electricity consumption and pressure ratio of the compressor for all refrigerants are given in Table 4.

Table 4. The results of thermodynamic analyses of the refrigeration system

	Refrigerant	COP	η_{ex}	$\dot{E}x_{Dest}$, kW	Pressure ratio	$\dot{W}_{Comp,el}$, kW
Natural	R600	2.500	0.249	29.58	6.443	29.04
	R717	2.462	0.2452	30.19	6.312	29.52
	R600a	2.448	0.2437	30.42	5.746	29.71
	R290	2.369	0.2359	31.75	4.537	30.76
	R1270	2.366	0.2356	31.82	4.385	30.81
	R170	1.463	0.1457	57.47	3.585	51.11
	R744	1.482	0.1476	56.63	3.963	50.45
Chlorine based	R12	2.455	0.2445	30.3	5.071	29.61
	R22	2.419	0.2409	30.89	4.991	30.08
	R134a	2.403	0.2393	31.17	5.951	30.3

The other important result is given in Figure 3, displaying the variation of the refrigerant charge in refrigeration system. Regarding the refrigerant charge of R12 to be 1 kg/kg, natural refrigerants show a significant reduction in refrigerant charge. As can be seen from Figure 3, natural refrigerants have about half the density of CFC, HCFC and HFC refrigerants excluding R744.

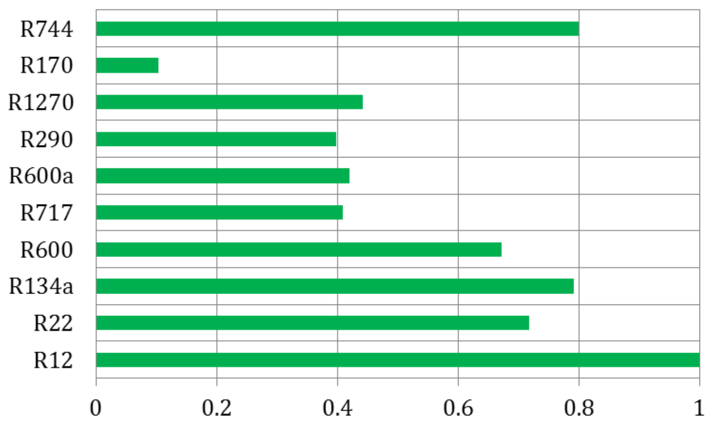

Figure 3. Comparison of refrigerant charges relative to R12.

To compare the variations of some system parameters with performance indicators, a parametric study is also carried out. For the analysis, the operating conditions are kept constant, whereas the condenser temperature is varied between 35 to 45 °C and the evaporator temperature is varied between -20 to -10 °C. Also for the exergy analysis, the reference temperature is varied between 18 to 28 °C to identify the variations of exergy efficiency and exergy destruction rates. In the following figures, the left hand side of the y-axis represents the values of all refrigerants excluding R170 and R744. The values for these two working fluids are given in right hand side of y-axis for better observation of the figures since their parametric values are differed too much from the others. Also the ranges and intervals of right and left hand sides of y-axis are different.

In Figures 4 to 6, the variations of COP, exergy efficiency and exergy destruction rates are given with the variation condenser temperature. As can be seen from the figures, with the increase of T_C, COP and exergy efficiency values decrease for all refrigerant. The exergy destruction rates increase with T_C on contrary to COP. The increase of exergy destruction with condenser temperature is because of while the condenser temperature increases due to compressor discharge temperature, the refrigerant vapor becomes superheated at a higher temperature and higher heat capacity.

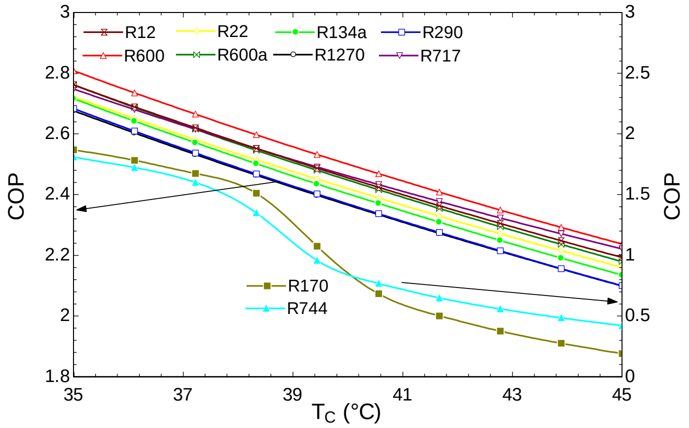

Figure 4. Variation of condenser temperature with COP

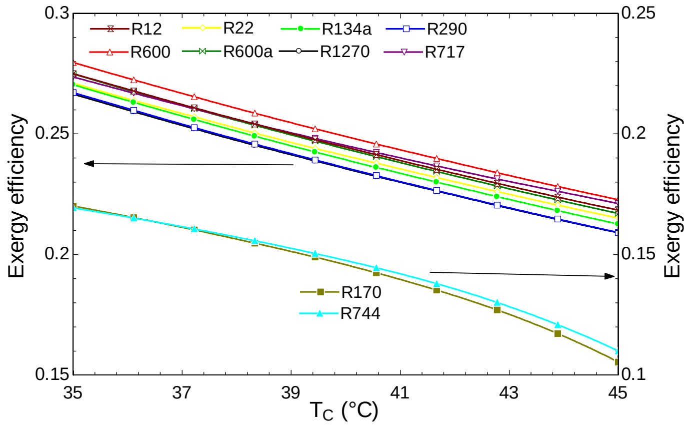

Figure 5. Variation of condenser temperature with exergy efficiency

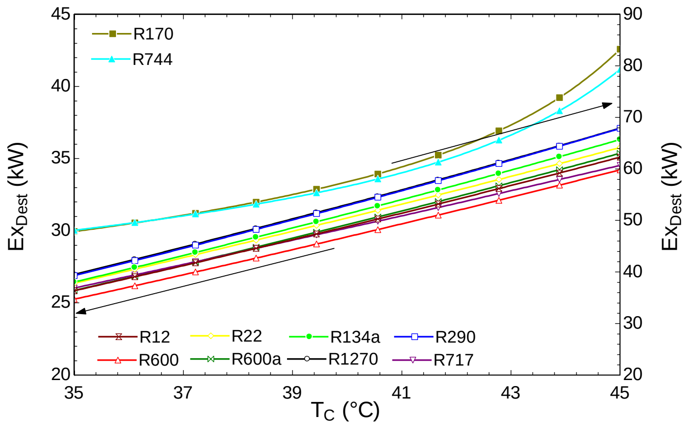

Figure 6. Variation of condenser temperature with exergy destruction

The analyses of the evaporator temperature variations show that the increase of T_E results an increase in COP and exergy efficiency (Figure 7 and 8). This is because the temperature difference between the cold chamber and evaporator is reduced. As expected, while the COP and exergy efficiency increase with T_E, the exergy destruction rates decrease for all refrigerants (Figure 9).

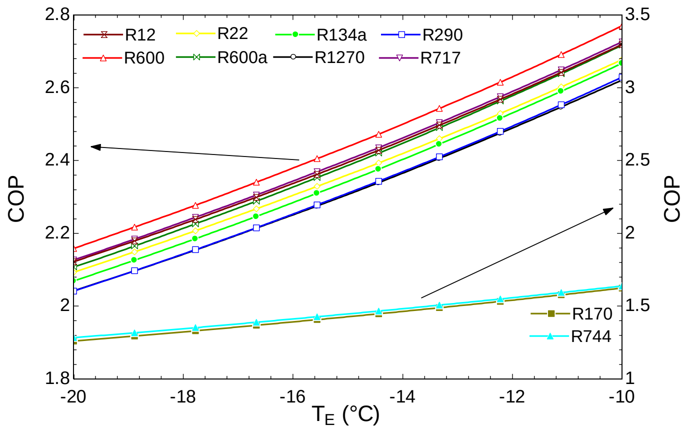

Figure 7. Variation of evaporator temperature with COP

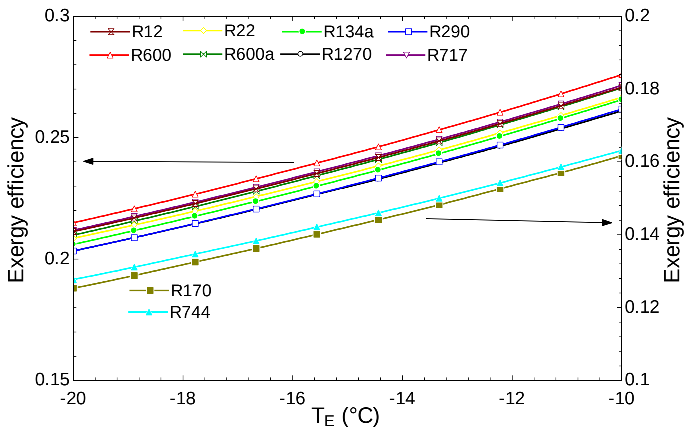

Figure 8. Variation of evaporator temperature with exergy efficiency

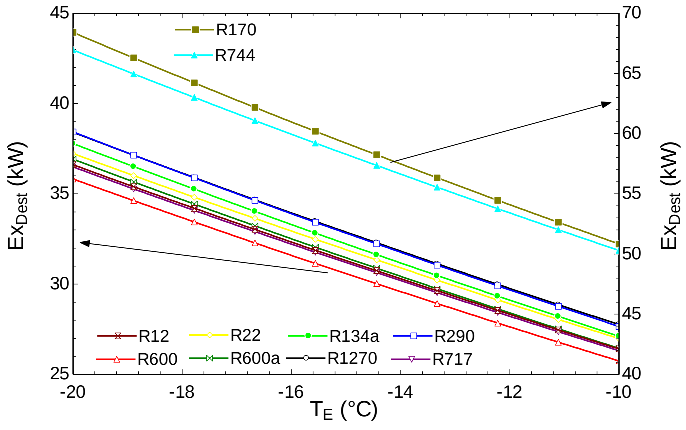

Figure 9. Variation of evaporator temperature with exergy destruction

In section 3, the exergy balance equations are given relative to reference state conditions. Also the definition of exergy is strictly related to reference temperature T_0 and reference pressure P_0. Accordingly, the results of exergy analyses generally are sensitive to variations of these properties. Before exergy analysis, it is very significant to assess reasonable values for reference state properties (Rosen and Dincer, 2004). Figure 10 shows the exergy efficiency of the refrigeration system with the variation of reference state temperature. While T_0 increases from 18 to 28 °C, the exergy efficiency increases considerably. Unlike this, the exergy destruction rate decreases for all working fluids during the increase of reference temperature (Figure 11).

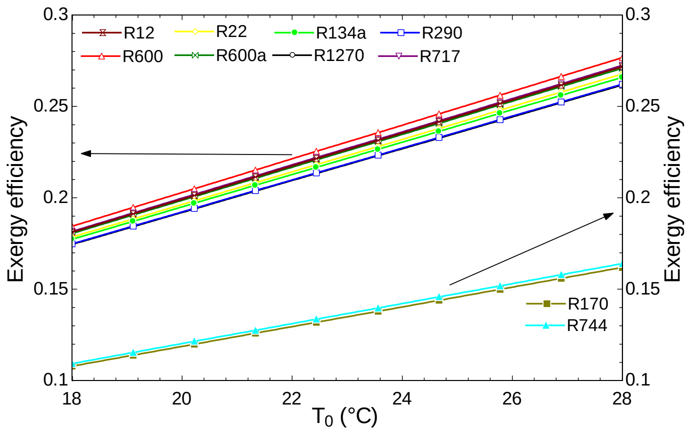

Figure 10. Variation of reference temperature with exergy efficiency

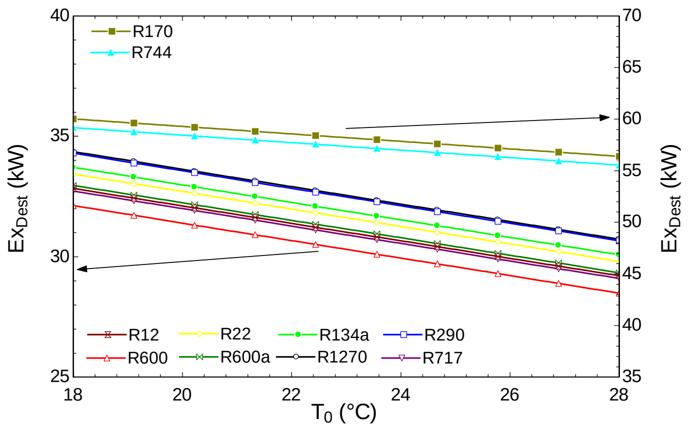

Figure 11. Variation of reference temperature with exergy destruction

5. Conclusions

In this study, a comparative analysis of refrigeration system for a cold storage facility was carried out using natural refrigerants. Also some chlorine based refrigerants such as R12, R22 and R134a were analyzed in order to identify the replacement possibilities. The results of the analyses were showed that the best COP was obtained by using R600 followed by R717, R600a and R290. One of the other important achieved results was that using the natural refrigerants, the refrigerant charge could be reduced considerably. According to the exergy analyses results, the highest exergy efficiency was found to be 24.9 % when using R600a and the highest exergy destruction rate was calculated as 57.47 kW for R170. The results of parametric studies showed that the increase of evaporator temperature had a positive effect on coefficient of performance, exergy efficiency and exergy destruction of the system while the increase of condenser temperature had negative effect. Consequently, it can be concluded that natural refrigerants can be considered as the suitable substitutes of CFC, HCFC and HFC type refrigerants; however more detailed and experimental studies should be done about these alternatives.

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