Isı Bilimi ve Tekniği Dergisi, 35, 2, 89-97, 2015
J. of Thermal Science and Technology

ISSN 1300-3615

SECOND LAW ANALYSIS OF DIFFERENT REFRIGERANTS IN A TWO STAGE VAPOR COMPRESSION CYCLE

Sertaç Samed SEYİTOĞLU*[1] and Ali KILIÇARSLAN**
*Hitit University Faculty of Engineering Department of Mechanical Engineering 19030, Çorum, Turkey,
[1]Corresponding email: sertacseyitoglu@hitit.edu.tr
** Hitit University Faculty of Engineering Department of Mechanical Engineering 19030, Çorum, Turkey,
alikilicarslan@hitit.edu.tr

(Geliş Tarihi: 03.07.2014, Kabul Tarihi: 02.04.2015)

Abstract: In this study, the second law analysis has been made by means of the program developed using the software program EES (EES-V9.172-3D) in a two stage vapor-compression refrigeration cycle for the refrigerants such as R600 (butane), R290 (propane), R152a (HFC) and R141b (HCFC) as refrigerants. It has been assumed that superheating and subcooling occur in the system and the calculations have been made accordingly. Irreversibilities occurring in the components of the cycle including compressors, evaporator, condenser, expansion valve and flash tank have been examined for different evaporator and condenser temperatures. As the condenser temperatures were increased, the irreversibilities of the components increased while they decreased with the increase of evaporator temperatures. The minimum irreversibility values occurred in the flash tank for all the refrigerants under the study. Furthermore, at the increasing values of evaporator temperatures, the total irreversibility of the refrigeration system decreased, but it increased with the condenser temperature. The total lowest irreversibility values are obtained by using R141b as a refrigerant in the refrigeration system for the increasing values of the condenser and evaporator temperature
Key Words: Refrigeration, two-stage refrigeration, irreversibility, R600, R290, R152a, R141b

İKİ KADEMELİ BUHAR SIKIŞTIRMALI ÇEVRİMLERDE FARKLI SOĞUTUCU AKIŞKANLARIN İKİNCİ KANUN ANALİZİ

Özet: Bu çalışmada soğutucu akışkan olarak R600 (bütan), R290 (propan), R152a (HFC) ve R141b (HCFC) kullanıldığı buhar sıkıştırmalı soğutma çevriminde, ikinci kanun analizi EES (EES-V9.172-3D) yazılım programı kullanılarak geliştirilen program yardımıyla yapılmıştır. Sistemde aşırı kızdırma ve aşırı soğutma olduğu varsayılmış ve buna göre hesaplamalar yapılmıştır. Buhar sıkıştırmalı soğutma sisteminin elemanlarında meydana gelen tersinmezlikler, farklı buharlaştırıcı ve yoğuşturucu sıcaklıklarına göre incelenmiştir. Yoğuşturucu sıcaklığı arttıkça sistem elemanlarında meydana gelen tersinmezlikler artarken buharlaştırıcı sıcaklığının arttırılması ile sistem elemanlarında meydana gelen tersinmezlikler azalmıştır. En düşük tersinmezlik değerleri çalışmada kullanılan bütün soğutucu akışkanlar için flash tank da meydana gelmiştir. Buharlaştırıcı ve yoğuşturucu sıcaklıklarının arttırılması durumunda sistemde en düşük toplam tersinmezlik değerleri R141b soğutucu akışkanı kullanılması ile meydana gelmiştir.
Anahtar Kelimeler: Soğutma, iki kademeli soğutma, tersinmezlik, R600, R290, R152a, R141b

NOMENCLATURE

ODP = Ozone Depletion Potential
HFC= Hydrofluorocarbons
HCFC= Hydrochlorofluorocarbons
COP= Coefficient of performance
$\dot{W}$ = Power [W]
$\dot{m}$ = mass flow rate [kg s^{-1}]
h= specific enthalpy [kJ kg^{-1}]
$\dot{s}_{gen}$ = entropy generation [kW K^{-1}]
s = specific entropy [kJ kg^{-1} K^{-1}]
T = Temperature [^oC]

Subscripts

$Comp_1$ = Compressor I
$Comp_2$ = Compressor II
Eva = Evaporator
Con = Condenser
Ft= Flash Tank

INTRODUCTION

As a result of technological developments, there have been significant improvements in terms of refrigeration. Due to these developments, refrigeration applications and fields of use have become widespread. Refrigeration applications are used in wide range of applications from local to industrial applications. Refrigeration has become very important especially in the area of food and health. Vapor compression refrigeration system is the most commonly used one among the refrigeration systems. The removal of heat for lowering the temperature of a substance or an environment and maintaining at that temperature is called cooling. In order to cool an environment heat must be removed with a refrigerant having temperatures colder than that of the environment.

In a vapor-compression refrigeration cycle, the heat absorbed from a low-temperature environment and it is rejected to an environment at a higher temperature. The electrical energy is delivered to the compressor and compressor circulates the refrigerant throughout the system by increasing both temperature and pressure of the refrigerant. The heat is rejected to the environment in the condenser while the heat is absorbed by the refrigerated space in the evaporator. These processes are called a vapor compression refrigeration cycle. There are four main elements in a classic vapor-compression refrigeration system. These are compressor, evaporator, expansion valve and condenser.

Sometimes, obtaining lower temperatures is desirable in the current refrigeration applications. In this case, the temperature range of the application may be too large for simple vapor compression refrigeration cycle to work effectively. Large temperature range leads to loss of too much pressure which causes compressor to operate at lower efficiency. One of the methods appealing to avoid this situation is to make the cooling system with two-stage.

The objectives of the studies related to the vapor compression refrigeration systems in the literature are to increase the efficiency of the vapor compression refrigeration systems and to reduce operating costs of those systems. In order to increase the coefficient of performance of the refrigeration systems, some studies in the literature are based on increasing evaporator temperature and decreasing condenser temperature. Also in the literature the subcooling and superheating effects are taken into consideration for the purpose of increasing the performance of the system. In the vapor compression refrigeration cycle, subcooling of the refrigerant occurs at the outlet of the condenser. After the refrigerant leaves the condenser, it enters the expansion valve. The expansion valve reduces the heat removal capacity of the refrigerant before it enters the evaporator. Subcooling process eliminates this drawback (ASHRAE, 1993). Superheating occurs at the end of the evaporator. It provides enhanced system performance. Furthermore, superheating is prevented the liquid refrigerant entering at the compressor. For all these reasons, the amount of superheating and subcooling affects the system performance directly (Dossat, 1997).

According to different systems and different refrigerants, the first law of thermodynamics (energy conservation) analysis is the most commonly used method. The first law only deals with the energy conservation and does not provide any information about the amount of change of the system performance where and how it occurs (Yumrutas *et al*., 2002). In this respect, the analysis based on energy is insufficient in some cases. For example, in the some applications where the heat exchangers (condenser, evaporator) are used at different temperatures and compressors having different efficiencies. These differences can be detected only by the second law of thermodynamics (Saidur *et al*., 2007).

With the second law analysis, namely exergy analysis, irreversibility of the system is determined. In the second law analysis process, exergy losses of the element in the entire system must be examined individually and the total irreversibility occurring in the system must be calculated in this way. Here is the lowest amount of irreversibility gives the most favorable working conditions.

The second law analysis used for different refrigerants and for different system elements in the vapor compression refrigeration systems have been found in a limited number of studies in the literature. Leidenfrost *et al*. (1980) have examined exergy analysis of the cycle in their study Freon-12 as refrigerant. Arcaklioglu and Erisen (2002) have performed the second law analysis of vapor compression refrigeration system using R12, R22, and R502 as refrigerants. Wongwises *et al*. (2006) have studied on the use of R290, R600 and R600a refrigerant instead of R134a for automotive air conditioning in their experimental study. Also Wongwises and Chimres (2005) have studied on the use of propane, butane and a mixture containing isobutene instead of HFC134a refrigerant for home refrigerator in their experimental study. As a result, they have shown that the mixture of 60% propane and 40% butane may be an alternative. Han *et al*. (2007) have experimented a new hydrocarbon mixture instead of R407c refrigerant R134a. As a result of the experimental and theoretical studies, R32/R125/R161 mixture has shown better characteristics of cooling capacity and cooling coefficient according to R407c refrigerant. Dalkilic and Wongwises (2010) have tested non-binary isentropic, and mixed refrigerants theoretically intended to improve the performance of vapor compression refrigeration systems. Kilic (2012) has analyzed exergy for a two stage refrigeration cycle and intercooler using R507, R407c, and R404a as refrigerants. He has calculated the thermodynamic values by Solkane program. He has compared the values of irreversibility, COP and exergetic efficiency for different refrigerants. Bayrakci and Ozgur (2009) have experimented R290, R600,

R600a, R1270, R22, R134a fluids for the vapor-compression refrigeration cycles on their study. They have found R600 refrigerant as an alternative of R22 and R134a refrigerants. Halfaoui *et al*. (2014) have investigated a new two stage vapor compressor system by using R134a as the refrigerant. Menlik *et al.* (2014) conducted an experimental study on a heat pump unit using a hydrocarbon mixture as an alternative to R134a. Since the type and sweeping volume of compressor is designed specifically for R134a, the proposed hydrocarbon mixture has shown lower COP and rational efficiencies than that of R134a. However, this mixture has shown higher volumetric refrigeration capacity and cooling exergy content than that of R134a. Shilliday *et al*. (2009) have examined the energy and exergy analysis of R744, R404 and R290 refrigerants for vapor-compression refrigeration systems in different condenser and evaporator temperatures on their study. Cimsit and Ozturk (2014) have investigated thermodynamic analysis of a new vapor compression-absorption two stage refrigeration cycle for different evaporator and generator temperatures. They have used NH_3-H_2O and NH_3 for the absorption and the vapor compression part of the system, respectively. Kilicarslan and Hosoz (2010) have made analysis of energy and irreversibility for a cascade refrigeration cycle using R152a–R23, R290–R23, R507–R23, R234a–R23, R717–R23 and R404a–R23 as refrigerant. They have concluded in the study that refrigerant pair of R507-R23 has low COP and high irreversibility values while the refrigerant pair of R717-R23 has high COP and low irreversibility values. Cimsit *et al.* (2014) have examined energy and exergy analysis of compression-absorption cascade refrigeration cycles. As a result, they have shown that as generator and evaporator temperatures increased, the COP of the cascade system increased. Padmanabhan and Palanisamy (2013) have studied the use of R134a, R290 and R407c instead of R22 as a refrigerant in their experiments and they concluded that the R407c may be an alternative refrigerant. Ma and Zhao (2008) have compared the performance of a heat pump system having a flash tank with a system having a sub-cooler. They have shown that the flash tank is more efficient according to subcooler at -25 °C ambient temperature. Heo *et al*. (2008) have tested the effect of flash tank vapor injection to the heating performance of a two-stage heat pump with one inverter double rotary driven compressor that has a frequency range between 50 to 100 Hz at ambient temperature of -15, -5 and 0 ^{o}C in their study. They have observed the case where the ambient temperature is -15 ^{o}C, COP value of injection cycle and heating capacity increased 25% and 10%, respectively.
Nikolaidis and Probert (1998) have calculated the exergy analysis of R22 refrigerant in two stage cooling cycle in their study. They have changed the condenser temperature between 298K and 308K and the evaporator temperature 228K and 238K. They have shown the effect of change in condenser and evaporator temperature to system irreversibility.

It is seen from literature survey above that there is no study dealing of the irreversibilities of the elements of a two stage vapor compression refrigeration system. Furthermore, there is no study in which R141b is used as a refrigerant in a two stage vapor compression refrigeration system. In this study, R600, R152a, R290 and R141b are preferred as refrigerants. The properties of the selected refrigerants are shown Table 1.

Table 1. Properties of selected refrigerants

Ref	ODP	GWP (100yr)	Safety group	Type	Molar Mass	Boiling Point (^{0}C)	Cr. Temp (^{0}C)
R600	0^{+}	4^{+}	A3*	Butane^{+}	$58,12^{+}$	-1 -1	152,01
R290	$<0^{+}$	3.3^{+}	A3*	Propane^{+}	$44,10^{+}$	-42,25	96,6
R152a	0^{+}	124^{+}	A2*	HFC^{+}	$66,05^{+}$	-25	113,26
R141b	$0,12^{+}$	725^{+}	A2*	HCFC^{+}	$116,95^{+}$	32	204,2

* ANSI/ASHRAE Standard 34-2007
+ Refrigerant Reference Guide, 2011

To summarize properties of these refrigerants; R600 is named Butane which is an organic compound with formula C_4H_{10}. It is used as a petrol component, as a feedstock for the production of base petrochemicals in steam cracking. This refrigerant has zero ozone depletion potential and low global warming potential for instance in household refrigerators and freezers. R290 is normally a gas with the molecular formula C_3H_8 which name is propane. R290 and R600 have negligible ozone depletion potential and can serve as a functional replacement for R12, R22, R134a and other hydrofluorocarbon and chlorofluorocarbon refrigerants in conventional stationary refrigeration and air conditioning systems. R152a is an organofluorine compound with the chemical formula $C_2H_4F_2$ and its other name is difluoroethane. It has lately been approved for use in automobile applications as an alternative to R134a. R141b is a haloalkane with the formula $C_2H_3CI_2F$ which is one of the three isomers of dichlorofluoroethane. Its properties; not flammable, colorless, liquid in atmospheric conditions, very volatile and liquid belongs to the hydrochlorofluorocarbon family (ASHRAE, 2007).

In this study, the second law analysis has been made by means of the computer code developed using the software program EES (EES-V9.172-3D) in a two stage vapor-compression refrigeration cycle using R600 (butane), R290 (propane), R152a (HFC) and R141b (HCFC) as refrigerants. Also it has been assumed that superheating and subcooling occurs in the system and the calculations have been made accordingly. Irreversibilities occurring in the elements of the two stage vapor-compression refrigeration system have been examined as a function of evaporator and condenser temperatures.

THERMODYNAMIC ANALYSIS

The two stage vapor compression refrigeration cycle is shown in Figure 1. The two stage vapor compression system consists of two compressors, two expansion valve, condenser, evaporator and flash tank.

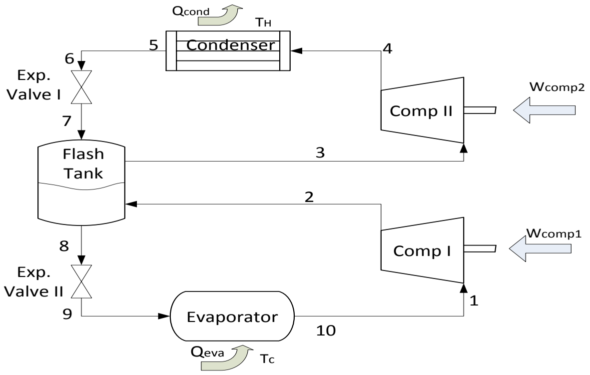
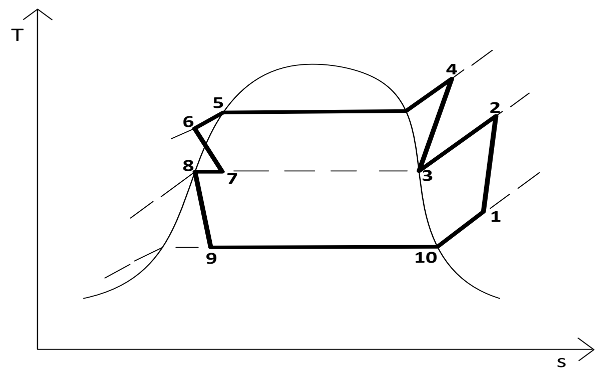
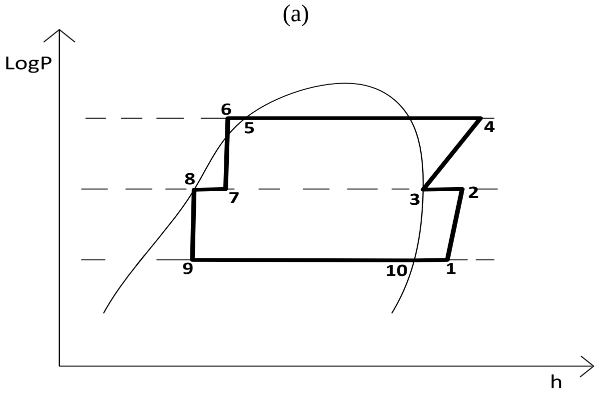


Figure 1. Schematic diagram of a two stage compression refrigeration cycle

The T-s (temperature-entropy) and LogP-h (logarithmic pressure-enthalpy) diagrams of the system are shown in Fig. 2 (a) and Fig. 2 (b), respectively.



(a)



(b)

Figure 2. (a) T-s diagram of the refrigeration cycle and (b) LogP-h diagram of the refrigeration cycle

In this cycle, the refrigerant at low temperature and low pressure enters the compressor I at state 1 as superheated steam. Compressor I increases both temperature and pressure of the refrigerant and then refrigerant enters the flash tank where it becomes saturated vapor. After leaving the flash tank, the refrigerant enters the compressor II at state 3. Then it flows to the condenser. After that the refrigerant leaves the condenser as a subcooled liquid. At this state, it enters to expansion valve I after leaving the expansion valve where its pressure and temperature are reduced. The refrigerant goes back to the flash tank and leaves it at state 8 as saturated liquid and then it enters the expansion valve II. The refrigerant leaving the expansion valve at state 9 enters the evaporator where it absorbs heat from the refrigerated space and become superheated vapor at state 10 and then returns the compressor I.

In the thermodynamic analysis of the two stage vapor compression refrigeration cycle, following assumptions are taken into consideration;

- The effects of variation of kinetic and potential energy are negligible during the flow of refrigerants throughout the system
- Flow through the pipes and the expansion valves are assumed to be adiabatic and compressors are also assumed to be adiabatic
- The flow of refrigerants into the refrigerating system components are analyzed in a steady and homogeneous regime.
- The refrigerant flows at constant pressure through the condenser and evaporator
- Dead state of the refrigerants at pressure P_0=1atm and temperature T_0=25°C are assumed.

In the light of the above assumption, according to the first law of thermodynamics, the amount of energy per unit time required by the compressors can be written as;

$$\dot{W}_{Comp1} = \dot{m}_2(h_2 - h_1) \tag{1}$$

$$\dot{W}_{Comp2} = \dot{m}_1(h_4 - h_3) \tag{2}$$

$$\dot{W}_{tot} = \dot{W}_{Comp1} + \dot{W}_{Comp2} \tag{3}$$

where $\dot{m}_1$, $\dot{m}_2$, h_1, h_2, h_3 and h_4 are the mass flow rate of refrigerant through the cycle and the specific enthalpies at the inlet and outlet of the compressors, respectively.

According to the conservation of mass, continuity equation for the flash tank can be written as;

$$\dot{m}_1 = \frac{\dot{m}_2(h_2 - h_8)}{(h_3 - h_7)} \tag{4}$$

where h_2, h_7, h_3 and h_8 are the specific enthalpies at the inlet and outlet of the flash tank, respectively. Exergy analysis gives information about the losses of all components of a system. Exergy destruction or irreversibility is the difference between reversible power and useful power. Irreversibility can be written as;

$$I = T_0\dot{s}_{gen} \tag{5}$$

where T_0 and $\dot{s}_{gen}$ are the absolute temperature and entropy generation, respectively. By using equation (5) the irreversibilities of the system components can be written as;

$$I_{Comp1} = \dot{m}_2T_0(s_2 - s_1) \tag{6}$$

$$I_{Comp2} = \dot{m}_1T_0(s_4 - s_3) \tag{7}$$

$$I_{Con} = \dot{m}_1 T_0 [(s_5 - s_4) + \frac{(h_4 - h_5)}{T_0}] \tag{8}$$

$$I_{Eva} = \dot{m}_2 T_0 [(s_{10} - s_9) + \frac{(h_9 - h_{10})}{T_{rs}}] \tag{9}$$

$$I_{TEV1} = \dot{m}_1 T_0 (s_7 - s_6) \tag{10}$$

$$I_{TEV2} = \dot{m}_2 T_0 (s_9 - s_8) \tag{11}$$

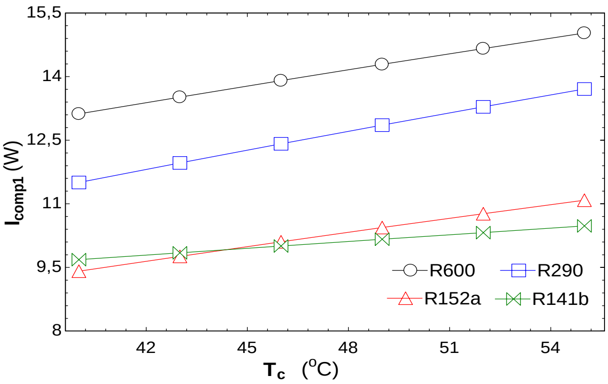
$$I_{ft} = T_0 [\dot{m}_1 (s_3 - s_7) + \dot{m}_2 (s_8 - s_2)] \tag{12}$$

RESULT AND DISCUSSION

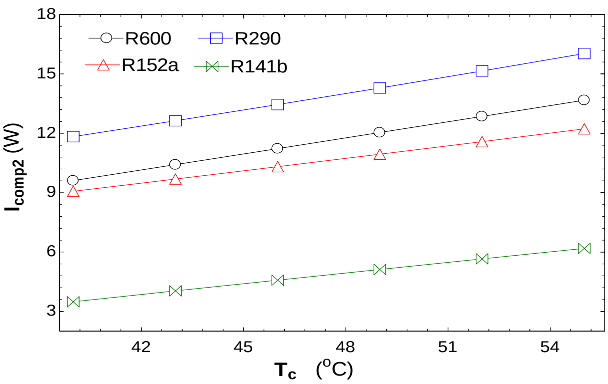
In this study, R600, R290, R152a and R141b refrigerants are used in two stage vapor compression refrigeration cycle. Second law analysis of the system is carried out by using EES (EES-V9.172-3D) software. During the operation of the programming code, some of the parameters were kept constant. These parameters and their values are given in Table 2.

Table 2. Data for a two stage vapor compression system

Environment Temperature	25 °C
Subcooling Temperature	5 °C
Superheating Temperature	7 °C
Isentropic Efficiency (Compressor I)	%75
Isentropic Efficiency (Compressor II)	%75



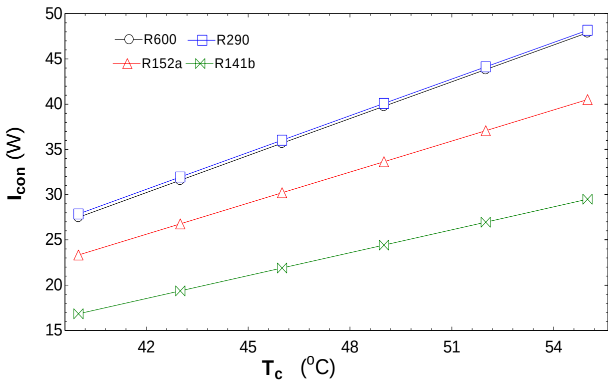
(a)



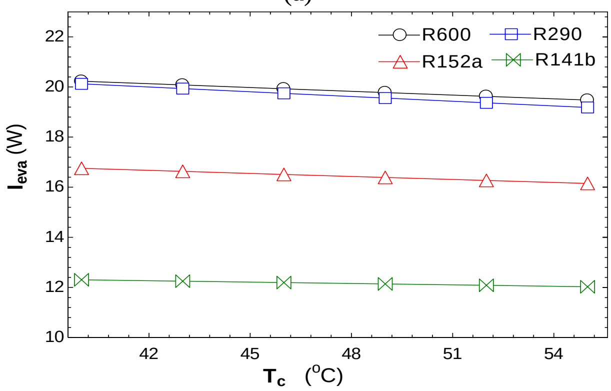
(b)

Figure 3. Variation of irreversibilities of Compressor-I and Compressor-II with condenser temperature

The results shown in Figure 3 are obtained by varying the condenser temperature from 40°C to 55°C. Evaporator temperature is kept constant at -10°C. As shown in Figure 3, as the condenser temperature increases, the irreversibilities of the compressor I and compressor II increase as well. The highest values of the irreversibility in compressor I is obtained by using R600 as a refrigerant in the refrigeration system but the highest irreversibility in compressor II is obtained by using R290 as a refrigerant. On the other hand, the lowest irreversibility in compressor I and compressor II are obtained by using R141b as a refrigerant in the refrigeration system. As the irreversibilities of compressor I and compressor II are compared, the irreversibility values of compressor I range from 9.685 to 15.03 while those of compressor II range from 3.488 to 16.03.



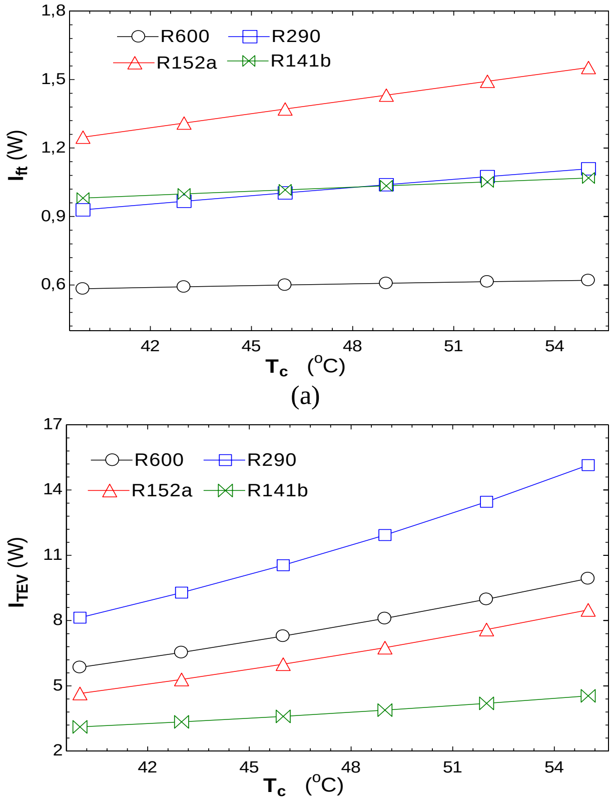
(a)



(b)

Figure 4. Variation of irreversibilities of condenser and evaporator with condenser temperature

The results shown in Figure 4 are obtained by varying the condenser temperature from 40°C to 55°C. As shown in Figure 4, as the condenser temperature increases, the irreversibilities of the condenser increase. The highest values of the irreversibility in condenser are obtained by using R600 as a refrigerant in the refrigeration system but the highest irreversibility in evaporator is obtained by using R290 as a refrigerant in the refrigeration system. On the other hand, the lowest irreversibility in condenser and evaporator are obtained by using R141b as a refrigerant in the refrigeration system. Moreover, in case of using R152a, the irreversibilities of the condenser and evaporator take place in the middle region the irreversibilities of the evaporator take place in the middle region in Figure 4. Also the irreversibilities of condenser and evaporator are very close to each other for refrigerants R290 and R600.



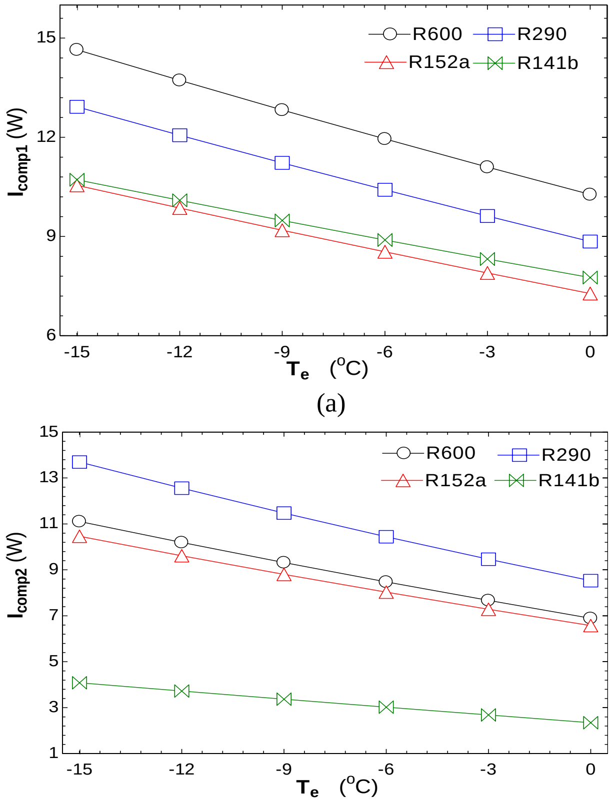
(a)

(b)

Figure 5. Variation of irreversibilities of flash tank and expansion valve with condenser temperature

Figure 5 depicts the change of irreversibilities of flash tank and expansion valve with the condenser temperature. Evaporator temperature is held constant at -10°C. As shown in Figure 5, as the condenser temperature increases, the irreversibilities of the flash tank and expansion valve increase. The highest values of the irreversibility in flash tank are obtained by using R152a as a refrigerant in the refrigeration system but the highest rates of irreversibility in the expansion valve are obtained by using R290. On the other hand, the lowest irreversibility in flash tank is obtained by using R600 as a refrigerant in the refrigeration system while those in the expansion valve is observed by using R141b is used as refrigerant. The irreversibility values of expansion valve take place in the middle region for refrigerants R600 and R152a increase parallel to each other as the condenser temperature increases. In the condenser temperature studied in this work, the irreversibilities of flash tank are very close to each other for refrigerants R290 and R141b.

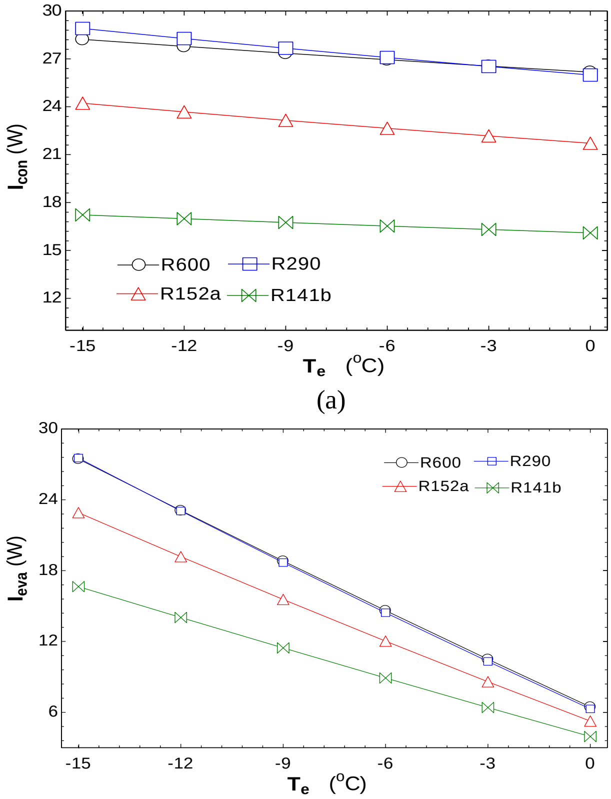
The variation of the irreversibilities of compressor I and compressor II are shown in figure 6 as a function of evaporator temperature. Evaporator temperature changes from -15°C to 0°C while holding the condenser temperature constant as 40°C. As shown in Figure 6, as the evaporator temperature increases, the irreversibilities of the compressor I and compressor II decrease. The highest values of the irreversibility in compressor I are obtained by using R600 as a refrigerant in the refrigeration system.



(a)

(b)

Figure 6. Variation of irreversibilities of Compressor-I and Compressor-II with evaporator temperature

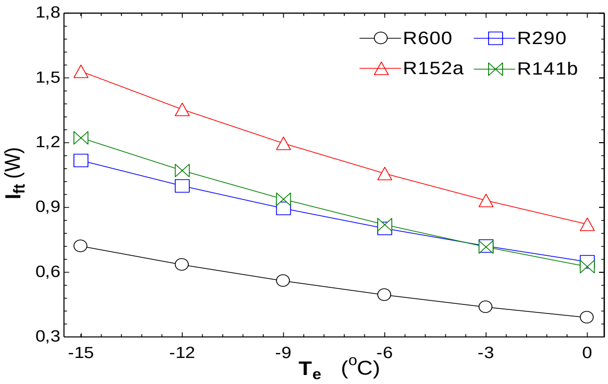


(a)

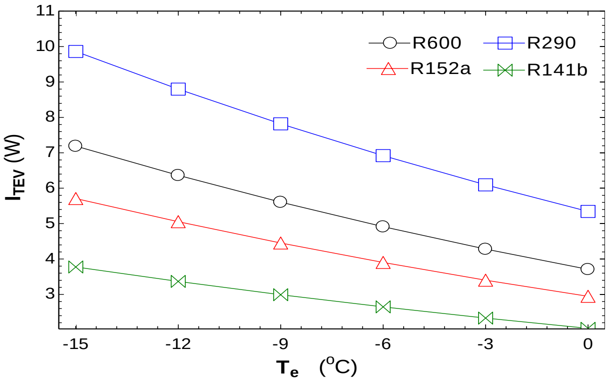
(b)

Figure 7. Variation of irreversibilities of condenser and evaporator with evaporator temperature.

The highest values of the irreversibility in compressor II are obtained by using R290 as a refrigerant. The lowest irreversibility in compressor II is obtained by using R141b as a refrigerant in the refrigeration system while those in the compressor I is obtained by using R152a as a refrigerant. The irreversibilities of compressor II obtained by using R600 and R152a in the refrigeration system decrease parallel to each other as the evaporator temperature increases.



(a)

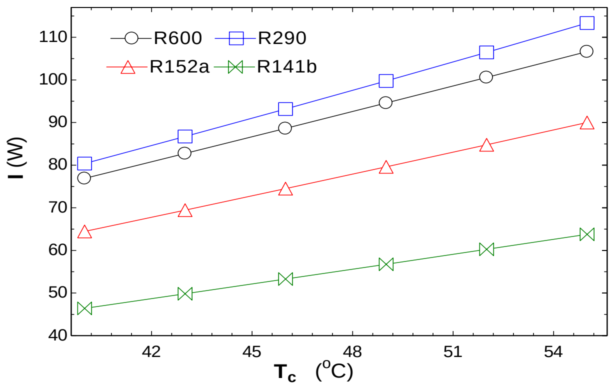


(b)

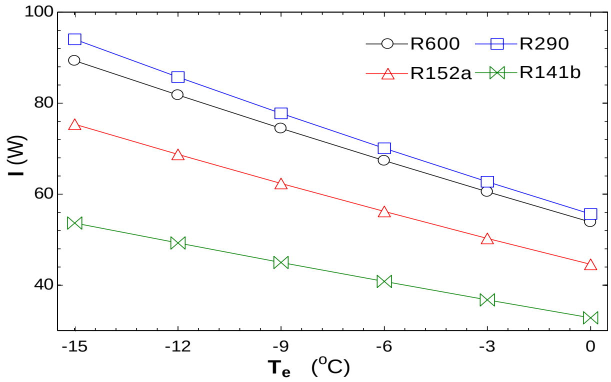
Figure 8. Variation of irreversibilities of flash tank and expansion valve with evaporator temperature

The results shown in Figure 7 are obtained by changing the evaporator temperature from -15°C to 0°C. Condenser temperature is kept constant at 40°C. As shown in Figure 7, as the evaporator temperature increases, the irreversibilities of the evaporator and condenser decrease. However, the irreversibilities of the evaporator for all the refrigerants in the study decrease and approach to each other at 0°C as the evaporator temperature increase. The highest values of the irreversibility in condenser are obtained by using R290 as a refrigerant in the refrigeration system while those in the evaporator are obtained by using R290. On the other hand the lowest irreversibility values in the evaporator and condenser are obtained by using R141b. Moreover, in case of using R152a, the irreversibilities of the condenser and evaporator take place in the middle region as shown in Figure 7.

The results depicted in Figure 8 are obtained by changing the evaporator temperature from -15°C to 0°C. Condenser temperature is kept constant at 40°C. As shown in Figure 8, as the evaporator temperatures increase, the irreversibilities of the flash tank and expansion valve decrease. The highest values of the irreversibility in expansion valve are obtained by using R290 as a refrigerant in the refrigeration system. While those in the flash tank are obtained by using R152a. The lowest irreversibilities are obtained in the expansion valve by using R141b but the lowest irreversibilities are obtained in the flash tank by using R600 as a refrigerant in the refrigeration system. In case of using R290 and R141b, the irreversibilities of the flash tank take place in the middle region and in case of using R152a and R600, the irreversibilities of the expansion valve take place in the middle region.



(a)



(b)

Figure 9. Variation of total irreversibilities of the systems with (a) Condenser temperature (b) evaporator temperature

The variation of the total irreversibility of the refrigeration system as a function of condenser and evaporator temperature for different refrigerants including R600, R290, R152a and R141b is depicted in Figure 9. The results shown in Figure 9a are obtained by varying the condenser temperature from 40°C to 55°C. Evaporator temperatures are kept constant at -10°C. As shown in Figure 9a, as the condenser temperature increases, the total irreversibility of the system increases. The highest values of the total irreversibility are obtained by using R290 while the lowest ones are obtained by using R141b in the refrigeration system. On the other hand, the variation of the total irreversibility as a function of evaporator temperature is shown in Figure 9b. Evaporator temperature changes from -15°C to 0°C while holding the condenser temperature constant as 40°C. As shown in Figure 9b, as the evaporator temperature increases, the total irreversibility decreases. The highest rates of total irreversibility are obtained by using R600 and R290 as a refrigerant in the refrigeration system. However, the lowest total irreversibility is obtained by using R141b as a refrigerant in the refrigeration system.

CONCLUSION

Vapor Compression Refrigeration systems are the most common type of refrigeration cycles. During the design of such systems, the determination of the parameters which have an impact on the overall system performance is important. Due to the increasing investment costs, more attention should be paid on the design. The main conclusions derived from this theoretical work can be summarized as follows;

- As the condenser and evaporator temperature increase, the lowest irreversibilities occurs in the flash tank. The highest irreversibilities occurs in the condenser for the increasing values of condenser temperatures, however the highest ones occur in the evaporator for the increasing values of the evaporator temperatures.

- For the increasing values of evaporator and condenser temperatures, the lowest irreversibilities are obtained by using R141b as a refrigerants in the elements (compressor II, evaporator, condenser, and expansion valve) however the lowest irreversibilities in the flash tank are obtained by using R600 as a refrigerant.

- The highest irreversibility values are obtained for the flash tank by using R152a as a refrigerant for increasing values of evaporator and condenser temperatures.

- The total lowest irreversibilities are obtained by using R141b as a refrigerant in the refrigeration system for the increasing values of the condenser and evaporator temperature while the highest total irreversibilities are obtained by using R290 as refrigerant in the cycle. Then the second lowest ones are obtained by using R152a as a refrigerant in the two stage vapor compression refrigeration system.

REFERENCES

Arcaklıoglu E., Erisen A., 2002, Exergy Analysis of Refrigerants R12, R22, R502, and Their Substitutes in Vapor Compression Refrigeration System, *Technology*, 3-4, 55-64.

ASHRAE, 1993, *Fundamentals Handbook*, American Society of Heating Refrigerating and Air Conditioning Engineers Inc, Atlanta.

Bayrakci H.C., Ozgur A.E., 2009, Energy and exergy analysis of vapor compression refrigeration system using pure hydrocarbon refrigerants, *International Journal of Energy Research*, 33, 1070-1075.

Cimsit C., Ozturk, I.T., 2014, The Vapour Compression-Absorption Two Stage Refrigeration Cycle And Its Comparison With Alternative Cycles, *Journal of Thermal Science and Technology*, 34, 1, 19-26.

Cimsit C.,Ozturk I.T., Hosoz M., 2014, Second Law Based Thermodynamic Analysis of Compression-Absorption Cascade Refrigeration Cycles, *Journal of Thermal Science and Technology*, 34, 2, 9-18.

Dalkilic A.S., Wongwises S., 2010, A performance comparison of vapour-compression refrigeration system using various alternative refrigerants, *International Communications in Heat and Mass Transfer*, 37, 1340-1349.

Dossat R.J., 1997, *Principles of Refrigeration*, Prentice Hall, New Jersey.

Halfaoui M.W., Tahar K., Ammar B.B., 2014, Performance analysis of a two stage vapor compression refrigeration cycle offering two cold temperatures, *5th International Renewable Energy Congress*, Tunisia.

Han X.H., Wang Q., Zhu Z.W., Chen G.M., 2007, Cycle performance study on R-32/R-125/R-161 as alternative refrigerant to R-407C, *Applied Thermal Engineering*, 27, 2559–2565.

Heo J., Jeona M.W., Kim Y., 2010, Effects of tank vapor injection on the heating performance of an inverter-driven heat pump for cold regions, *International Journal of Refrigeration.* 33, 848-855.

Kilic B., 2012, Exergy analysis of vapor compression refrigeration cycle with two-stage and intercooler, *Heat and Mass Transfer*, 48-7, 1207-1217

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.

Leidenfrost W., Lee K.H., Korenic K.H., 1980, Conservation of energy estimated by second law analysis of power-consuming process, *Energy*, 5, 47-61.

Ma G.Y., Zhao H.X., 2008, Experimental study of a heat pump system with flash-tank coupled with scroll compressor, *Energy Build*, 40, 697-701.

Menlik T., Ozcan H., Arcaklioglu E., 2014, Second Law Analysis of An Environmentally Friendly R290/R600/R600a Mixture in a Water-Cooled Heat Pump Unit, *Journal of Thermal Science and Technology*, 34, 2, 19-28.

Nikolaidis C., Probert D., 1998, Exergy-method Analysis of a Two-stage Vapour Compression Refrigeration Plants Performance, *Applied Energy,* 60, 241-256.

Padmanabhan V.M.V., Palanisamy S.K., 2013, Exergy efficiency and irreversibility comparison of R22, R134a, R290 and R407C to replace R22 in an air conditioning system, *Journal of Mechanical Science and Technology*, 27-3, 917-926

Saidur R., Masjuki H.H., Jamaluddin M.Y., 2007, An application of energy and exergy analysis in residential sector in Malaysia, *Energy Policy*, 35, 1050-1063.

Shilliday J.A, Tassou S.A, Shilliday N., 2009, Comparative energy and exergy analysis of R744, R404A and R290 refrigeration cycles, *International Journal of Low-Carbon Technologies*, 1–8.

Wongwises S. Chimres N., 2005, Experimental study of hydrocarbon mixtures to replace HFC-134a in a domestic refrigerator, *Energy Conversion and Management*, 46, 85–100.

Wongwises S., Kamboon A., Orachon B., 2006, Experimental investigation of hydrocarbon mixtures to replace HFC-134a in an automotive air conditioning system, *Energy Conversion and Management*, 47, 1644–1659.

Yumrutas R., Kunduz M., Kanoglu M., 2002, Exergy analysis of vapor compression refrigeration systems, *Exergy*, 2, 266-272.

Sertaç Samed SEYITOGLU is a research assistant at the Department of Mechanical Engineering, Hitit University, Corum, Turkey. He is a Ph.D. student at the same university. His research areas include heat transfer, refrigeration systems, energy and exergy analyses.

Ali KILICARSLAN graduated from Middle East Technical University Gaziantep Campus in 1989 with BSME Degree and received MSME in 1994 and Ph.D. in 2000 from Istanbul Technical University. He is a Professor at the Department of Mechanical Engineering, Hitit University, Corum, Turkey and teaching Advanced Thermodynamics, Energy and Exergy Analysis of Thermal Systems, Thermodynamics, Refrigeration and Air Conditioning. He is The Director of The Graduate School of Natural and Applied Sciences. He is also Editor of "Hittite Journal of Science and Engineering". He worked as a Postdoctoral Investigator with Dr. Norbert Mueller at the Department of Mechanical Engineering, Michigan State University, USA in between 2003 and 2004. He worked for Michigan State University (Summer 2005) as a Visiting Professor, teaching the course "Refrigeration (ME 491)"at the Department of Mechanical Engineering. He made an experimental research on gasoline engines as a postdoctoral researcher at the Department of Mechanical Engineering, Central Michigan University, USA in summer 2013. His research focuses on Gas-Fired Pulse Combustion, Refrigeration, Air Conditioning and energy and exergy analyses of thermal systems.