EVALUATING ON EXERGY ANALYSIS OF REFRIGERATION SYSTEM WITH TWO-STAGE AND ECONOMISER

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Abstract

In this study, the first and second law analysis of vapor compression refrigeration cycle with two-stage and economiser were carried out. R134a was used in system as refrigerant. The necessary thermodynamic values for analyses were calculated by Solkane program. The coefficient of performance (COP), exergetic efficiency (ηex) and total irreversibility rate of the system in the different operating conditions were investigated for this refrigerant. As a result, the highest coefficient of performance value and exergy efficiency value was obtained in the condenser temperature 25 ^oC and evaporator temperature 5 ^oC in the refrigeration system with two-stage and economiser. The highest irreversibility value was obtained in the condenser temperature 45 ^oC and evaporator temperature -15 ^oC in the refrigeration system.

Keywords: Refrigeration system, COP, exergy analysis, two-stage.

1. Introduction

Single-stage compression process can give quite satisfactory results for the evaporation temperatures until -15 ^oC in vapor compression refrigeration cycles of condensation temperatures is not too high. However, the capacity of cooling cycle together with the coefficient of performance rapidly decreases in working conditions are very low evaporation temperatures. Compression process started with low suction pressure can reach for the same outlet pressure of condensation that it requires a higher compression ratio. As a result of this high compression ratio, the output pressure and temperature tends to rise even more. For these reasons, stage compression processes is used for to prevent excessive compressor outlet pressure and temperature and also to provide more efficient operation conditions. Stage compression usually provides with two or three series compression. Type of the refrigerant circulating in the system is the same.

Zhang et al. have developed a general methodology for the optimal synthesis in cascade refrigeration system to maximize the energy efficiency. Effectiveness of the method developed has made a research on strengthening for the cascade refrigeration system in an ethylene plant [1]. Ahamed et al. have made exergy analysis of vapor compression refrigeration systems used in various sectors. They have found that exergy depends on evaporating temperature, condensing temperature, sub-cooling and compressor pressure [2]. Lucia has worked on magnetic refrigeration which represents a safe cooling technology. His study, the first and second law analysis was made of the magnetic cooling and the results used for engineering applications [3]. Rezayan et al. have made thermoeconomic optimization and exergy analysis of the a CO_2/NH_3 cascade refrigeration cycle. They have determined the total annual cost of the system which includes costs of input exergy to the system and annualized capital cost of the system for the objective function. Results shown in their study that optimum values of decision variables may be found by trade-off between the input exergy cost and capital cost [4]. Shuxue et al. have made thermal analysis model based on the first and second law of thermodynamics. Their study shown that compressor has the greatest exergy loss about %77 of the total exergy. They said that exergy losses in heat pumps can be reduced by two-stage compression [5].

In addition to studies conducted so far in this study, the total irreversibility rate of the refrigeration system with two-stage and economiser has been calculated for different condenser and evaporator temperatures. Solutions have been proposed to for the reduce the irreversibilities.Thermodynamic values needed for the analysis have been calculated with the computer program, Solkane 7.0. Solkane 7.0 is a computer program that calculates all thermodynamic properties of 23 refrigerants involved in it. In addition, it contains 5 units of different cooling cycle and 2 different Rankine cycle. The programme is capable of calculation of thermodynamic properties in each point of the system depending on input values of these cycles. It can also display P-h and T-s diagrams that belong to the system. The computer programme Solkane 7.0 supports procedures that are performed in German, English, French, Spanish, Italian, Russian, Arabic and Chinese languages [9].

2. Cycle Description

Schematic diagram of vapor compression refrigeration cycle with two-stage and economiser is shown in Fig. 1. Cycle's temperature-entropy diagram is shown in Fig. 2. Description of the cycle are given in Table 1.

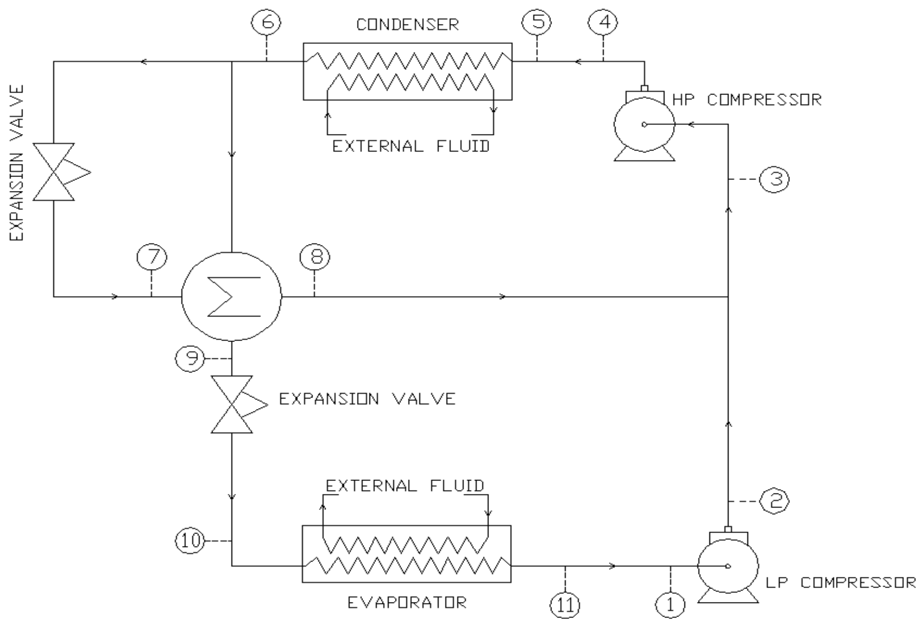

Fig. 1 Schematic diagram of vapor compression refrigeration cycle with two-stage and economiser

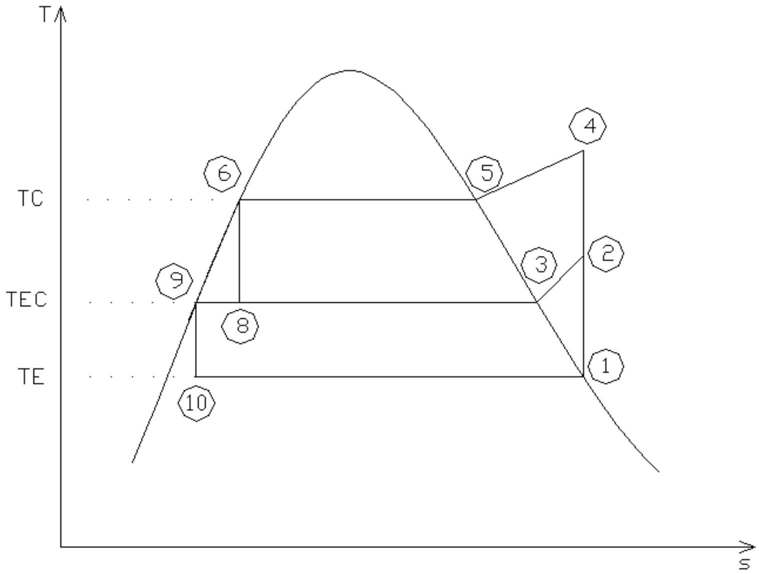

Fig. 2 The temperature-entropy diagram of the system in this work

Table 1. Cycle description

Point	Description
1	Low-pressure compressor input, (refrigerant inputs the low-pressure compressor as saturated vapor)
2	Low-pressure compressor output,(refrigerant is compressed to high pressure)
3	High-pressure compressor input, (mixing point of partial flow rates 2 and 8)
4	High-pressure compressor output, (refrigerant outputs from the high-pressure compressor as superheated vapor)
5	Condenser input, (refrigerant inputs the condenser as superheated vapor)
6	Condenser output, (refrigerant outputs from the condenser as saturated liguid)
7	Expansion valve output and same time economiser input, (refrigerant is liquid)
8	Economiser output, (refrigerant is liguid+vapor)
9	Expansion valve input, (refrigerant inputs the expansion valve as saturated liquid)
10	Evaporator input, (refrigerant inputs the evaporator)
11	Evaporator output, (refrigerant inputs the low-pressure compressor as saturated vapor)

Alternative refrigerant R134a used in this study don't contain chlorine, therefore; they have zero ozone depletion potential (ODP). Some physical properties of these alternative refrigerant are shown in Table 2.

Table 2. Physical properties of the refrigerant R134a

Refrigerant	**R134a**
Molecular Weight (g/mol)	102
Boiling Point at 1 atm (^{o}C)	–26,1
Critical Temperature (^{o}C)	101,1
Critical Pressure (kPa)	4060
Ozone Depletion Potential (ODP)	0
Global Warming Potential (GWP)	1300

3. FIRST AND SECOND LAW ANALYSIS

Constant values for first anad second law analysis are given below:

- All components of the system are working at a steady-state.
- The changes in kinetic and potential energies of the components that form the system have been ignored.
- The pressure losses along the pipelines have been ignored.
- Isentropic efficiency of LPC is η_{LPC}=%80.
- Isentropic efficiency of HPC is η_{HPC}=%80.
- Environmental temperature is T_0 = 25 ^oC.

According to the first law of thermodynamics, the refrigeration capacity of the system can be calculated as follows:

$$\dot{Q}_E = \dot{m}\ (h_{11} - h_{10}) \tag{1}$$

The low pressure compressor work load can be expressed as follows:

$$\dot{W}_{LPC} = \dot{m}\ (h_2 - h_1) \tag{2}$$

The high pressure compressor work load can be writen as follows:

$$\dot{W}_{HPC} = \dot{m}\ (h_4 - h_3) \tag{3}$$

Coefficient of performance (COP) of the refrigeration cycle can be calculated as follows [7-8]:

$$COP = \frac{\dot{Q}_E}{\dot{W}_{LPC} + \dot{W}_{HPC}} \tag{4}$$

Irreversibility rate of each component of the refrigeration cycle are calculated as follows [10-13]:

Evaporator:

$$I_E = E_{10} + \dot{Q}_E\left(1 - \frac{T_0}{T}\right) - E_{11} \tag{5}$$

The low pressure compressor (LPC):

$$I_{LPC} = E_1 - E_2 + W_{EL} \tag{6}$$

The high pressure compressor (HPC):

$$I_{HPC} = E_3 - E_4 + W_{EL} \tag{7}$$

Condenser:

$$I_C = E_5 - E_6 \tag{8}$$

Total irreversibility rate of the system is the total of irreversibilities in different components of the system:

$$I_T = I_E + I_{LPC} + I_{HPC} + I_{ECO} + I_C + I_{EV} \tag{9}$$

Exergetic efficiency can be calculated as follows [14]:

$$\eta_{ex} = \frac{\dot{Q}_E \left| \left(1 - \frac{T_0}{T_E} \right) \right|}{\dot{W}_{LPC} + \dot{W}_{HPC}} = \frac{COP_{vcr}}{COP_{rr}} \tag{10}$$

Where, COP_{vcr} is the coefficient of performance of vapor-compression cycle. COP_{rr} is the coefficient of performance of reversible refrigerant operating between T_o and T_E temperatures.

4. Results And Dıscussıon

Irreversibility values have been found depending on the evaporator temperature change in the vapor compression refrigeration cycle with two-stage and economiser and it is given in Fig. 3. Alternative refrigerant R134a has been used in the system and condenser temperatures have been kept constant. The highest total irreversibility rate of system for among all the working condition is found to be -15 ^{o}C for evaporator temperature, 45 ^{o}C for condenser temperature and about 700 Watt for total irrevesibility rate.

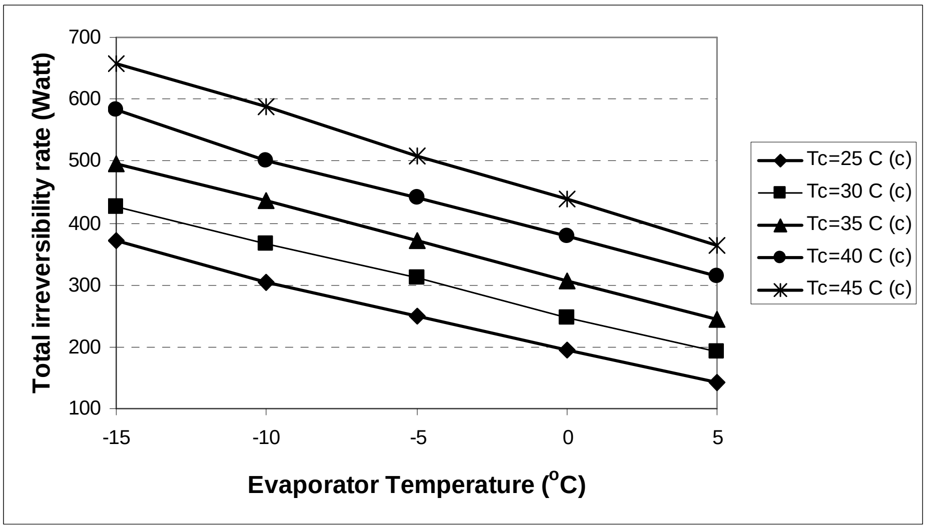

Fig. 3 Variation of total irreversibility rate with evaporator temperature

Exergy efficiencies obtained depending on the evaporator temperature change in the vapor compression refrigeration cycle with two-stage and economiser are given in Fig. 4. The condenser temperatures have been kept constant in the system. The highest exergy efficiency of system for among all the working condition is found to be 5 ^{o}C for evaporator temperature, 25 ^oC for condenser temperature and about %72 for exergy efficiency. Exergy efficiency is so high because of the fact that the vapor compression refrigeration cycle with two-stage and economiser has been used in the study and this study is theoretical. It is concluded that these results are acceptable because, the purpose of using such a refrigerant system is to increase coefficient of performance and exergy efficiency. The lowest exergy efficiency of system for among all the working condition is found to be -15 ^oC for evaporator temperature, 45 ^oC for condenser temperature and about %35 for exergy efficiency.

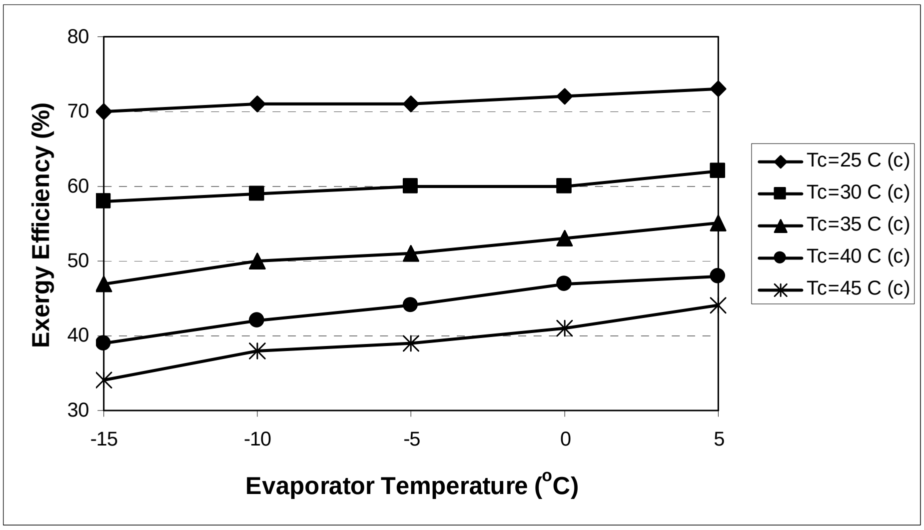

Fig. 4 Variation of exergy efficiency with evaporator temperature

Variation of COP with evaporator temperature in the vapor compression refrigeration cycle with two-stage and economiser are given in Fig. 5. The condenser temperatures have been kept constant in the system. The highest COP of system for among all the working condition is found to be 5 ^oC for evaporator temperature, 25 ^oC for condenser temperature and about 10,5 for COP.

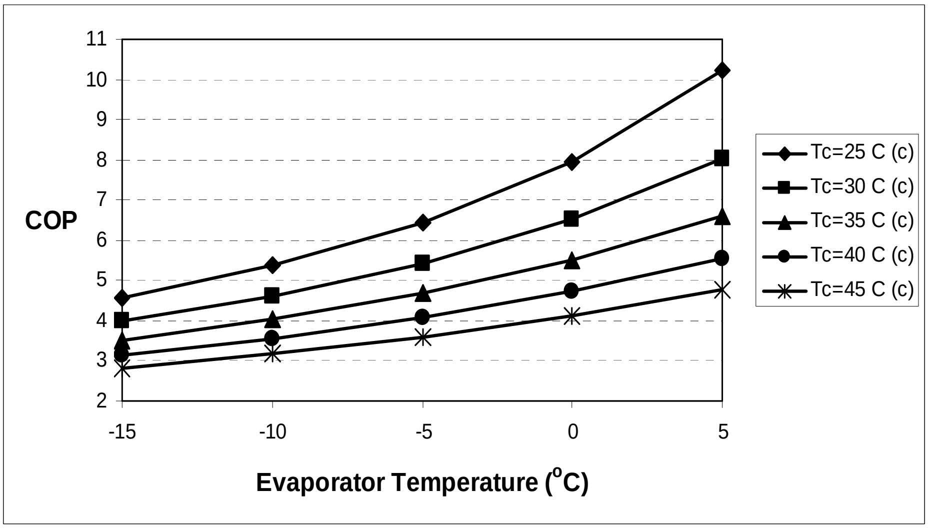

Fig. 5 Variation of COP with evaporator temperature

Variation of total irreversibility rate with exergy efficiency in the vapor compression refrigeration cycle with two-stage and economiser are given in Fig. 6. Evaporator temperatures have been changed and the condenser temperatures have been kept constant at

25 ^{o}C in the system. It is observed that exergy efficiency increase in direct proportion to the decreases of total irreversibility rate for this working condition in the system. The highest exergy efficiency value has been reached as %73 for this working condition.

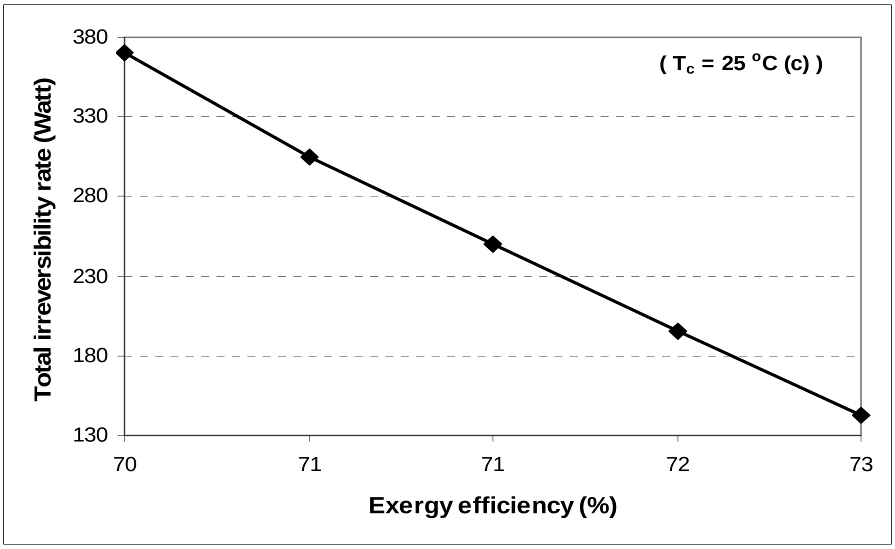

Fig. 6 Variation of total irreversibility rate with exergy efficiency

Variation of total irreversibility rate with exergy efficiency in the vapor compression refrigeration cycle with two-stage and economiser are given in Fig. 7. Evaporator temperatures have been changed and the condenser temperatures have been kept constant at 30 ^{o}C in the system. It is observed that exergy efficiency increase in direct proportion to the decreases of total irreversibility rate for this working condition in the system. The highest exergy efficiency value has been reached as %62 for this working condition.

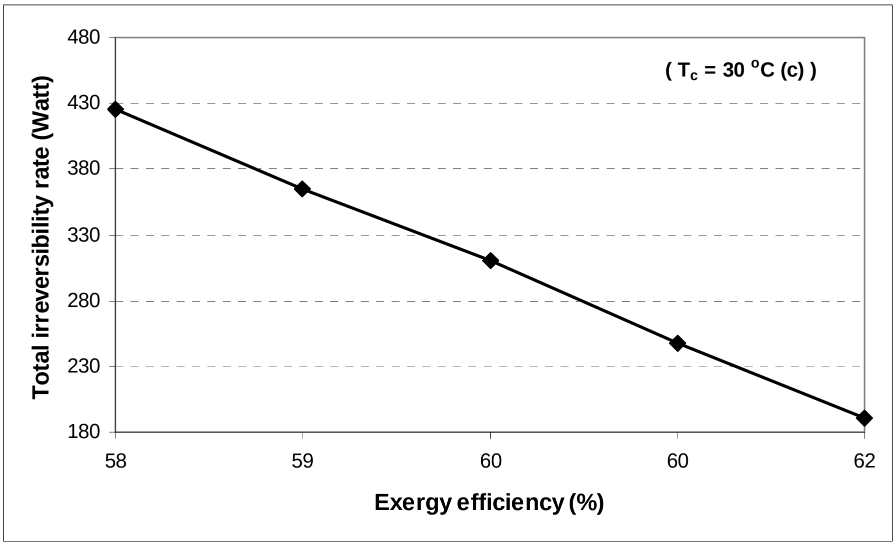

Fig. 7 Variation of total irreversibility rate with exergy efficiency

Variation of total irreversibility rate with exergy efficiency in the vapor compression refrigeration cycle with two-stage and economiser are given in Fig. 8. Evaporator temperatures have been changed and the condenser temperatures have been kept constant at 35 ^{o}C in the system. It is observed that exergy efficiency increase in direct proportion to the decreases of total irreversibility rate for this working condition in the system. The highest exergy efficiency value has been reached as %55 for this working condition.

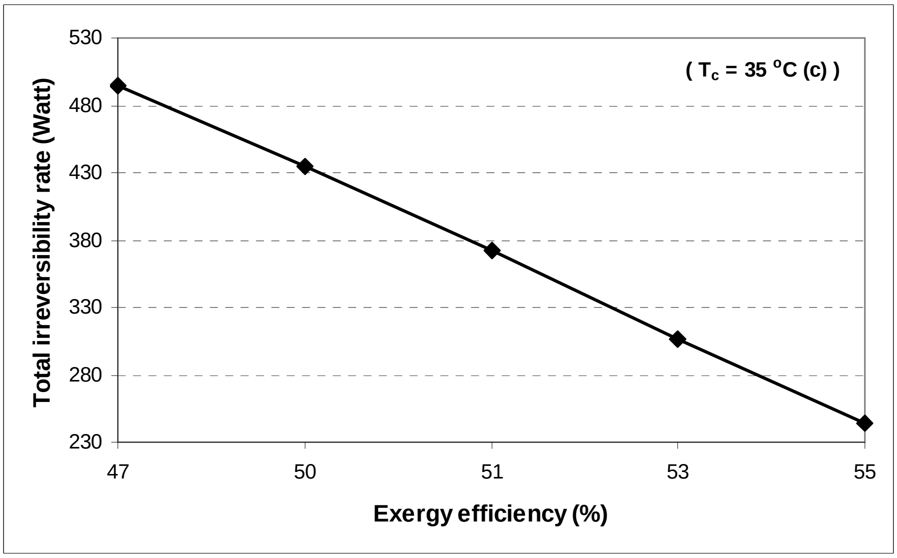

Fig. 8 Variation of total irreversibility rate with exergy efficiency

Variation of total irreversibility rate with exergy efficiency in the vapor compression refrigeration cycle with two-stage and economiser are given in Fig. 9. Evaporator temperatures have been changed and the condenser temperatures have been kept constant at 40 ^{o}C in the system. It is observed that exergy efficiency increase in direct proportion to the decreases of total irreversibility rate for this working condition in the system. The highest exergy efficiency value has been reached as %48 for this working condition.

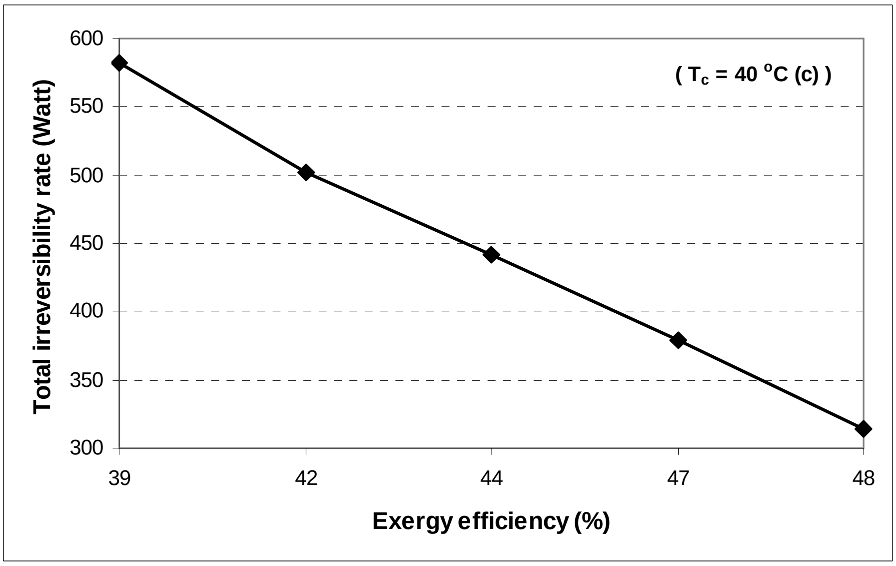

Fig. 9 Variation of total irreversibility rate with exergy efficiency

Variation of total irreversibility rate with exergy efficiency in the vapor compression refrigeration cycle with two-stage and economiser are given in Fig. 10. Evaporator temperatures have been changed and the condenser temperatures have been kept constant at 45 ^{o}C in the system. It is observed that exergy efficiency increase in direct proportion to the decreases of total irreversibility rate for this working condition in the system. The highest exergy efficiency value has been reached as %44 for this working condition.

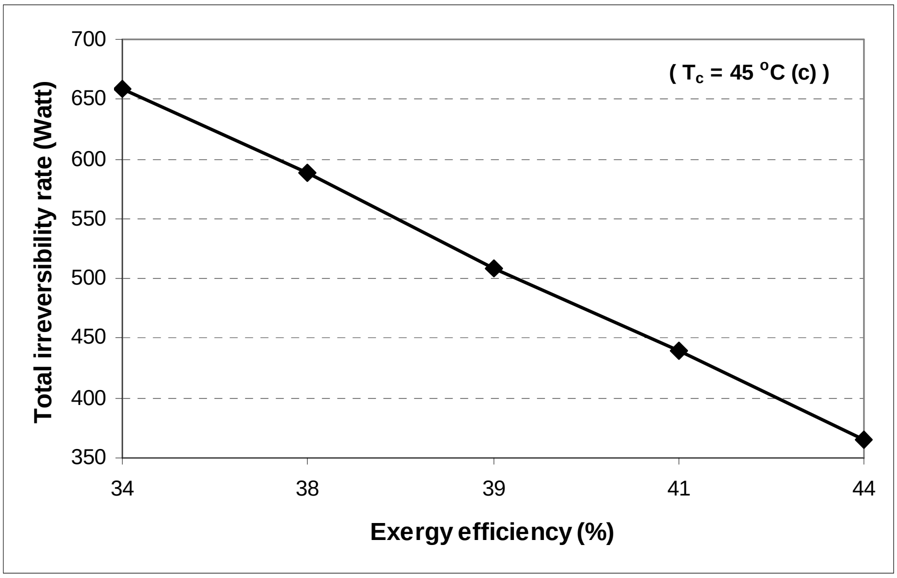

Fig. 10 Variation of total irreversibility rate with exergy efficiency

5. Conclusions

In this study, the first and second law analysis of vapor compression refrigeration cycle with two-stage and economiser were carried out. The coefficient of performance (COP), exergetic efficiency (ηex) and total irreversibility rate of the system in the different operating conditions were investigated. The highest total irreversibility rate of system for among all the working condition is found to be -15 ^{o}C for evaporator temperature, 45 ^{o}C for condenser temperature and about 700 Watt for total irrevesibility rate. The highest exergy efficiency of system for among all the working condition is found to be 5 ^{o}C for evaporator temperature, 25 ^{o}C for condenser temperature and about %72 for exergy efficiency.

Irreversibility in components of the system are based on pressure drop arising from phase change, the temperature difference between refrigerant fluid and refrigerated environment, heat tranfer and frictions. It is observed that total amount of irreversibility depends on evaporator and condenser temperature change. Accordingly, as the temperature difference between evaporator, condenser and ambient temperature extends, irreversibility value of system component and thus of entire system increases. To improve irreversibility in evaporator and condenser, enlargement the surface area and utilization of a very good conductive material can be recommended. But, these recommendations may affect system costs adversely. So, it’s concluded that the best way to improve irreversibility can be achieved with determination of optimum operation conditions. This study can be helpful for producers and engineers who produce or design refrigeration system two-stage and economiser.

6. References

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Nomenclature

COP	coefficient of performance
G	gravity (m/s2)
E	exergy (kW)
H	enthalpy (kJ/kg)
I	irreversibility rate (kW)
m	mass flow rate (kg/s)
Q	heat transfer rate (kW)
S	entropy (kJ/kg K)
T	temperature (^oC)
V	velocity (m/s)
W	work rate (kW)
Z	heigth (m)
ε	exergy (kJ/ kg)
η	efficiency

Subscript

C	condenser
E	evaporator
EV	expension valve
ex	exergy
HPC	high pressure compressor
EC	economiser

in	input
LPC	low pressure compressor
R	refrigerant
rr	reversible refrigeration
out	output
vcr	vapor compression refrigeration
0	reference condition
EL	electric