

Optimization through mathematical modelling of Irreversibility and other parameters in Simple VCR cycle for R-134a

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Abstract - In this paper, optimization of irreversibility and various other parameters is carried out through mathematical modeling for simple vapor compression refrigeration cycle using the properties of refrigerant R 134a. The study was carried out with the starting point as the exit of an evaporator assuming that the refrigerant leaves the evaporator as saturated vapor and compressed to some superheated temperature. Various values of change in exergy, Irreversibility, Second law efficiency, COP and ECOP are calculated for the whole cycle calculated for various evaporator temperatures and plotted to build a mathematical model to calculate either one by just knowing the value of the other. A relationship is established between the evaporator temperature and other parameters like COP, Total Exergy Change, Irreversibility, ECOP and second law efficiency for various evaporator temperatures for entire VCR cycle.

Key Words: VCR Cycle, Refrigerants, Exergy, Irreversibility, COP, ECOP, R-134a.

1. INTRODUCTION

Thermodynamics, as basically applied to heat engines, was concerned with the thermal properties of their 'working materials' such as steam in an effort to increase the efficiency and power output of engines. Thermodynamics later expanded to the study of energy transfers in chemical processes and heat and work. At that time Thermodynamics was limited to the First law of Thermodynamics only which was expanded to the Second law of Thermodynamics and the Zeroth law thereafter. Entropy and reversibility is the outcome of the Second law of Thermodynamics. Refrigeration cycles are the major outcome of the Second law of Thermodynamics. Vapor Compression Refrigeration Systems are the most commonly used in almost all refrigeration systems. These systems belong to general class of vapor cycles, in which the working fluid, popularly known as 'Refrigerant' undergoes the change of phase at least one time during the cycle. The cycle on which the actual Vapor Compression Cycle works is Reversed Carnot Cycle, which is also called as Evans-Perkins Cycle. Second law analysis is widely accepted as a useful tool in obtaining the improved understanding of the overall performance of any system and its components. Second law analysis also helps in taking account the important engineering decisions regarding design parameters of a system. Researchers carried out

second law studies of different thermal energy conversion systems describing various approach for second law analysis and its usefulness in a more simple and effective manner.

In the early days of study of exergy, an exergy method for compressor performance analyzed by J.A McGovern and S Harte [1] presented in 1995. The authors studied this to identify and quantify defects in the use of a compressor's shaft power. They claimed that this information could be used as the basis for compressor design improvements. The defects attributed to friction, irreversible heat transfer, fluid throttling, and irreversible fluid mixing and described, on a common basis, as exergy destruction rates and their locations were identified. They claimed the utility of the method for any type of positive displacement compressor. An analysis of an open reciprocating refrigeration compressor using R12 refrigerant experimented as an example. The results presented consist of graphs of the instantaneous rates of exergy destruction according to the mechanisms involved, a pie chart of the breakdown of the average shaft power wastage by mechanism, and a pie chart with a breakdown by location.

Ranendra Roy et al [2] analyzed multiple properties of multiple refrigerants and concluded that exergy loss of the system increases with the decrease in evaporator temperature. Exergy loss of the system increases when R134a and R290 are used other than CFC12. Maximum increase in exergy loss using R134a is 4.4% at condenser temperature of 40°C. It was a truly practical approach towards the analysis of second law properties numerically. They also concluded that the exergetic efficiency decreases with decrease in evaporator temperature. Maximum efficiency is achieved by using CFC12 and that differ from the efficiency of system using R134a is 3.1% at condenser temperature of 40°C. Shreekant Tare et al [3] during the study of exergy loss of multiple refrigerants in condenser of a VCR system established relationship between various properties of the refrigerants and calculated exergy loss at various pressures and degree of sub cool at the end of condensation process. The authors also formulated mathematical models to find out either properties just by knowing the other one and concluded the optimum parameters for the best possible combination of COP and exergy loss for specific refrigerant. While studying combined effect of degree of superheat at the entrance of compressor

and degree sub cooling at the end of condenser on performance of a VCR system, C.O.

Adegoke *et al* [4] came across many interesting conclusions with help of some graphs. Authors plotted graphs were in between exergy and COP, degree of superheat, degree of sub cool etc. Variation of Exergy with the Degree of Superheat: The authors observed that since less heat is “available” or absorbed by the evaporator in the superheated region, these results into a reduced COP. While working on the analysis of first and second law of a VCR system for different refrigerants. The first and second law analysis for vapor compression refrigeration system was analyzed thermodynamically by *Michel Louis et al* [5]. The performance of an experimental Vapor-Compression Refrigerating Machine (VCRM) using R410A was analyzed by the authors. They used exergy method to evaluate the irreversibility of each component by varying some working parameters. The dimensionless exergy loss rate of each component analyzed, emphasizing their contribution to the total exergy loss rate.

Theoretical assessment of the performance of R441a was done by *C.Prabha et al* [6] with standard parameters such as pressure ratio, volumetric cooling capacity (VCC), coefficient of performance (COP) and compressor input power. They obtained the results showing that the hydrocarbon refrigerant blend is suitable to be used as alternatives to R134 a as there is no mismatch in VCC. They also concluded that theoretically R441a has approximately the same VCC with about 5.2% higher COP at lower values of evaporator temperatures. Further they found that the compressor input power is also considerably reduced while using R441a. It was proved with the reported results that R441a is energy efficient and environment friendly alternative to phase out R134 a in domestic refrigerators.

Naushad A. Ansari et al [7] in his Theoretical Exergy Analysis of HFO-1234yf and HFO-1234ze as an Alternative Replacement of HFC-134a in Simple Vapor Compression Refrigeration System concluded that even though the values of performance parameters for HFO-1234yf are smaller than that of HFC-134a, but the difference is small, so it can a good alternative to HFC-134a because of its environmental friendly properties. HFO-1234ze can replace the conventional HFC-134a after having slight modification in the design as the performance parameters are almost similar. *Shreekant Tare et al* [8] in his second law analysis for compression process during a VCR cycle is done for number of refrigerants. Keeping value of degree of superheated in increasing order, their exergy change and change in COP calculated, plotted and analyzed at different absolute pressure at the end of compression process. The ultimate aim of any thermodynamic analysis is to reduce (energy) input or to enhance (energy) output. As exergy is the energy available for doing work, increase in exergy during any work input process can help to work the system more efficient way. As the study of the process was limited to COP of the

system and $d\psi$ only, other parameters like GWP, physical properties, availability, cost etc are not considered and only the outcome of second law of thermodynamics observed critically. Concluding R 134a to be fitted in the criterion most suitably amongst the set of refrigerants considered for observation.

R.S.Mishra et al [9] presents methods for improving first law and second law efficiency using new refrigerants: R134a, R290, R600a, R1234yf, R502, R404a and R152a and R12, R502. For exergy and energy analysis six type of vapor compression refrigeration system have been considered with using eco-friendly refrigerants with multi-evaporators, multiple compressors and multiple expansion valve with parallel and series with inter cooling and flash chambers. Numerical computational model have been carried out for the system. Performance of the system using R600 and R152a nearly matching same values under the accuracy of 5 % can be used in the system and difficulties detected with R600, R290 and R600a having flammable problems therefore safety measures are required using these refrigerants, therefore R134a is recommended for practical and commercial applications

2. Second Law analysis

2.1 Thermodynamic properties of R134a

| | |
|----------------------------|--------|
| Molar mass | 102.03 |
| Boiling point (°C) | -26.1 |
| Critical temperature (°C) | 122 |
| Critical pressure (bar) | 4.06 |
| Critical density (Kg / m3) | 515.3 |
| Ozone depletion potential | 0.05 |

2.2 System Description:-

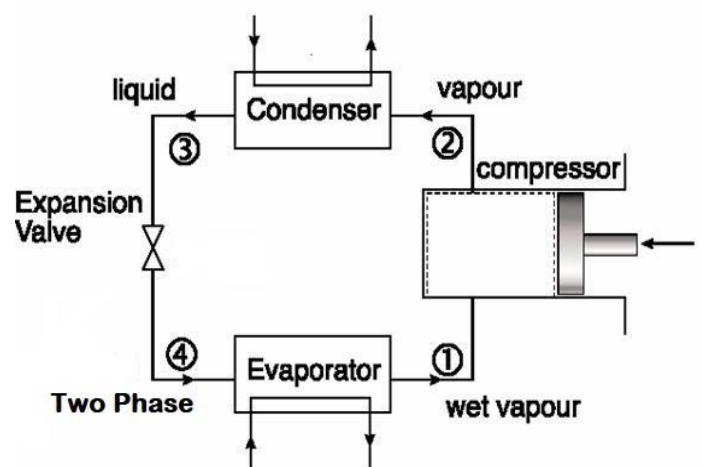


Figure (a)

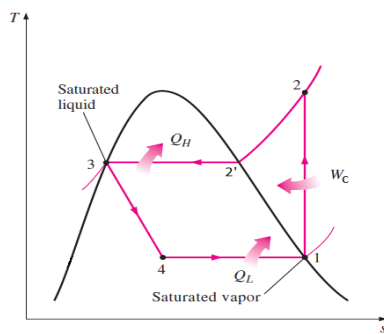


Figure (b)

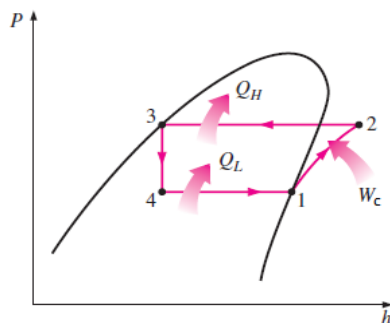


Figure (c)

Figure (a) shows the schematic diagram of a Simple vapor compression refrigeration system.

Figure (b) shows the t-s diagram of a Simple vapor compression refrigeration system

Figure (c) shows the p-h diagram of a Simple vapor compression refrigeration system.

a. COP = COP of vapor compression refrigeration system is an important criterion to analyze the performance. It represents refrigeration effect per unit compressor work.

$$COP = \text{Refrigeration effect} / \text{Work done}$$

$$= Q_L / W_c = (h_1 - h_4) / (h_2 - h_1)$$

b. Carnot COP (cCOP) = $T_e / (T_c - T_e)$

c. Second Law Efficiency (η_{II}) = $COP / cCOP$

d. Total Exergy Change (dEx) =

dEx = Sum of Exergy change of each component

i. Exergy Change in Compressor (de1) :

$$de1 = (h_{out} - h_{in}) - T_o (s_{out} - s_{in}) \quad (\text{kJ/kg})$$

$$de1 = (h_2 - h_1) - T_o (s_2 - s_1)$$

ii. Exergy Change in Condenser (de2) :

As condenser is a heat exchanger, the total change in exergy is the total sum of exergy change during the

process less the loss of exergy through latent heat rejection and sensible heat rejection.

$$de2 = (h_{out} - h_{in}) - T_o (s_{out} - s_{in}) - q_{rej} (1 - T_o / T_c) - C_p \log_n (T_{sup} / T_{sat}) * T_o \quad (\text{kJ/kg})$$

$$de2 = (h_3 - h_2) - T_o (s_3 - s_2) - (h_2 - h_3) * (1 - T_o / T_c) - C_{pv} * \log_n (T_2 / T_c) * T_o \quad (\text{kJ/kg})$$

iii. Exergy Change in Expansion Valve (de3):

As expansion process through an expansion valve generates lot of entropy, the exergy loss can be shown as under.

$$de3 = (h_{out} - h_{in}) - T_o (s_{out} - s_{in}) \quad (\text{kJ/kg})$$

$$de3 = (h_4 - h_3) - T_o (s_4 - s_3) \quad (\text{kJ/kg})$$

iv. Exergy Change in Evaporator (de4) :

As evaporator is a heat exchanger, the total change in exergy is the total sum of exergy change during the process in addition to the gain in exergy through latent heat addition and sensible heat addition.

$$de4 = (h_{out} - h_{in}) - T_o (s_{out} - s_{in}) + q_{add} * (1 - T_o / T_e) \quad (\text{kJ/kg})$$

$$de4 = (h_1 - h_4) - T_o (s_1 - s_4) + (h_1 - h_4) * (1 - T_o / T_e) \quad (\text{kJ/kg})$$

e. Irreversibility (I) = Total Work Done by the Compressor - Total exergy Change

$$I = W_c - dEx \quad (\text{KJ/Kg})$$

f. Exergy COP (ECOP) = Exergy Change in Evaporator / Exergy Change in Compressor

$$ECOP = de4 / de1$$

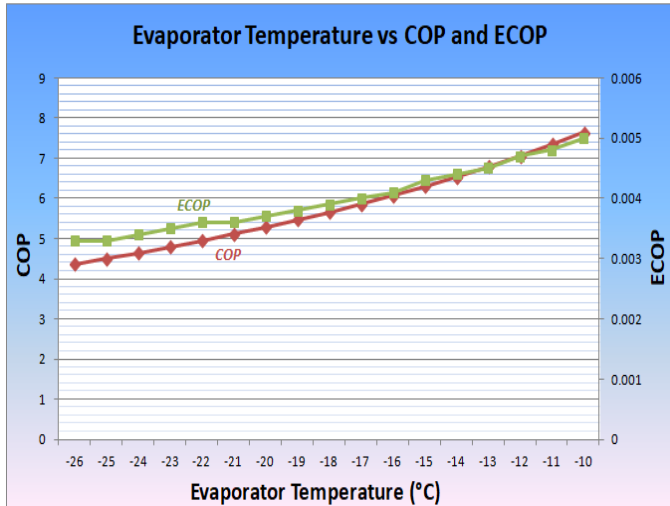
2.3 Assumptions:

To neglect the minor changes and to simplify calculations, following assumptions are being made.

- 1) Compression process is assumed to be isentropic.
- 2) The condenser temperature is assumed to be constant at 20°C.
- 3) Various evaporator temperatures are considered from -26°C to -10°C in the regular ascending interval of 1°C.
- 5) Atmospheric conditions are considered to be 101.3 kPa and 27°C.
- 6) Mass of refrigerant in circulation is assumed to be unit mass.
- 7) Condenser is water cooled.

3. Result and Discussion

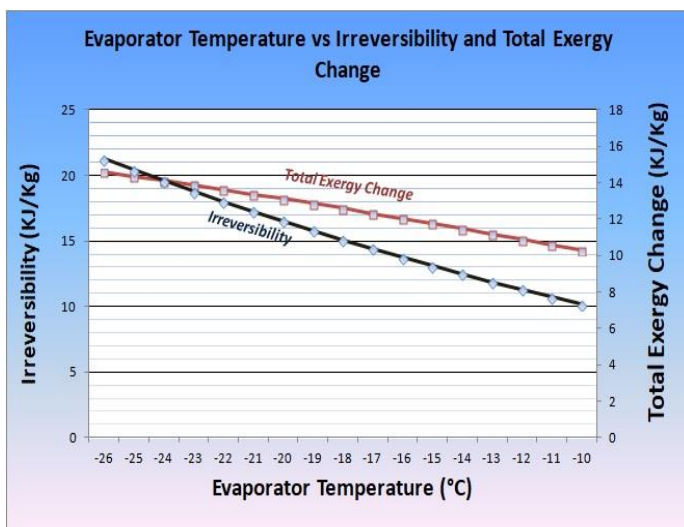
a. Variation of COP & ECOP with evaporator temperature:-



Graph 1

It is clear from graph 1 that as the evaporator temperature is increased, COP increases. This is due to the reason that COP is directly proportional to refrigerating effect, and inversely proportional to the compressor work input showing increase in the value of COP with respect to higher temperature of the condenser in the specified range. Similarly ECOP of the system also increases with increase in the evaporator temperature. This shows that the change of exergy in the evaporator is more predominant over the change in exergy in the compressor.

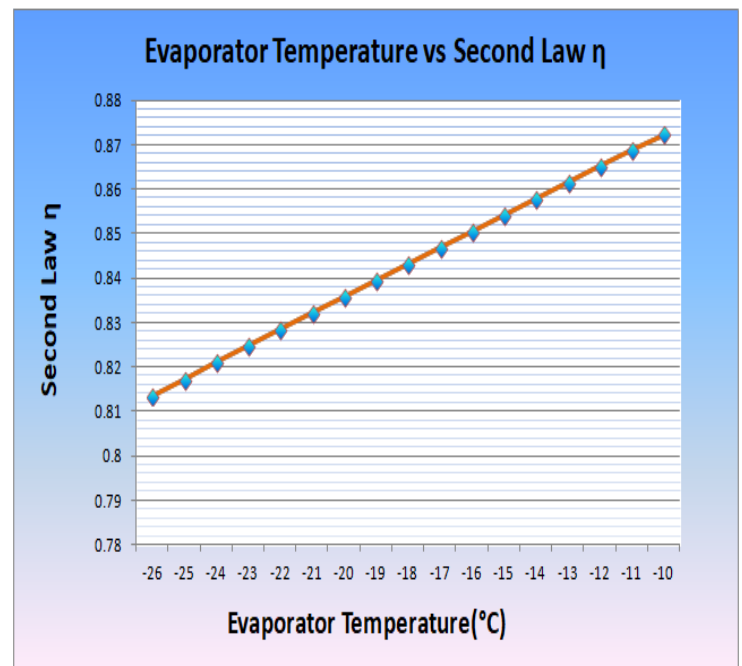
c. Variation of Irreversibility and Total Exergy change with evaporator temperature:-



Graph 3

Evaporator temperature which is function of evaporator effects the quantum of irreversibility. It is obvious that when evaporator temperature increases, irreversibility of the system as overall decreases and the system approaches towards reversible system although, it is impossible to achieve overall reversibility. Graph 3 gives clear indication about the irreversibility of the system with respect to evaporator temperature. May it be beneficial to keep the evaporator temperature at the higher side, requirement of the low and further lower temperature of the evaporator can be a limiting factor for this variable. The total exergy change of the system decreases with the increase in evaporator temperature but the slope of total exergy change is less steep than slope of irreversibility. This is due to the fact that with the increment of evaporator temperature work input decreases more than the total exergy change.

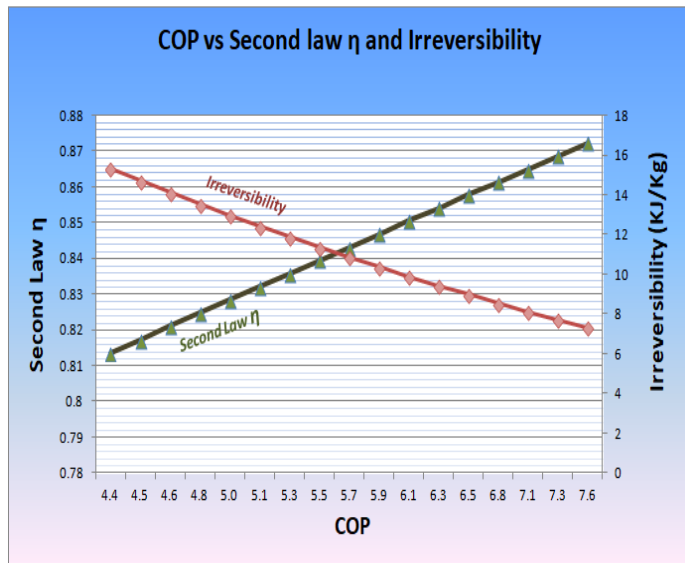
b. Variation of Second law η with evaporator temperature:-



Graph 2

It is seen in earlier graph that, as evaporator temperature increases, COP increases. As Carnot COP also increases, the ratio in the form of II law η would have remained the same, but the value of Carnot COP increases less steep than the COP and thus II law η increases sharply

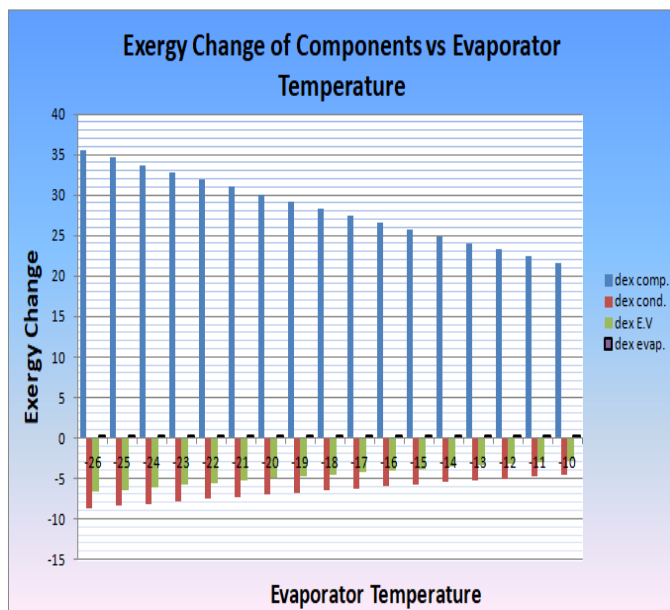
d. Variation of Second law η and Irreversibility with COP:-



Graph 4

As COP increases the numerator part of II law efficiency also increases predominantly over Carnot COP. II Law efficiency thus found to be function of COP. However irreversibility which is the difference of exergy change to the actual consumed, decreases with increase in COP, and less power is consumed during the cycle. This variation can be seen in graph 4.

e. Variation of exergy change in the components with evaporator temperature:-



Graph 5

Graph 5 indicates very interesting results which are of real concern with this paper. The exergy is gained in the compressor as the work is being added to the system in compressor. Similarly due to heat gain in the evaporator exergy gain is being observed. The highest loss of exergy is observed in condenser where heat is lost to cold fluid and the process is highly irreversible. Beyond this, the exergy loss decreases with increase in evaporator temperature. Similarly in expansion valve, where the liquid having no vapors, expand resulting in formation of mixture. Here entropy increases reasonably causing great quantum of irreversibility and exergy lost up to an extent.

4. Mathematical Models

A mathematical model is built to show polynomial relationship as under:

- i. COP and Evaporator temperature (in °C) (T_e):-

$$COP = 1.096987479 \cdot (10^{-7}) \cdot (T_e)^5 + 1.373640577 \cdot (10^{-5}) \cdot (T_e)^4 + 7.701925592 \cdot (10^{-4}) \cdot (T_e)^3 + 2.649887156 \cdot (10^{-2}) \cdot (T_e)^2 + 6.639287901 \cdot (10^{-1}) \cdot (T_e) + 12.2800747$$
- ii. Second Law Efficiency (η_{II}) and Evaporator temperature (in °C) (T_e) :-

$$\eta_{II} = 4.465328107 \cdot (10^{-10}) \cdot (T_e)^5 + 3.820314021 \cdot (10^{-8}) \cdot (T_e)^4 + 1.397877895 \cdot (10^{-6}) \cdot (T_e)^3 + 2.347247937 \cdot (10^{-5}) \cdot (T_e)^2 + 3.811366507 \cdot (10^{-3}) \cdot (T_e) + 9.091311842 \cdot (10^{-1})$$
- iii. ECOP and Evaporator temperature (in °C) (T_e):-

$$ECOP = -3.969198824 \cdot (10^{-10}) \cdot (T_e)^5 - 4.812653753 \cdot (10^{-8}) \cdot (T_e)^4 - 2.100698584 \cdot (10^{-6}) \cdot (T_e)^3 - 3.863370472 \cdot (10^{-5}) \cdot (T_e)^2 - 1.599137058 \cdot (10^{-4}) \cdot (T_e) + 5.591176478 \cdot (10^{-3})$$
- iv. Irreversibility (I) and COP:-

$$I \text{ (KJ/Kg)} = -2.856948064 \cdot (10^{-3}) \cdot (COP)^5 + 1.020089956 \cdot (10^{-1}) \cdot (COP)^4 - 1.500127673 \cdot (COP)^3 + 11.5733968 \cdot (COP)^2 - 48.69955206 \cdot (COP) + 99.64202261$$
- v. Second Law Efficiency (η_{II}) and COP:-

$$\eta_{II} = 1.204499495 \cdot (10^{-5}) \cdot (COP)^5 - 4.134298069 \cdot (10^{-4}) \cdot (COP)^4 + 5.894714966 \cdot (10^{-3}) \cdot (COP)^3 - 4.510483146 \cdot (10^{-2}) \cdot (COP)^2 + 2.008443177 \cdot (10^{-1}) \cdot (COP) + 0.436858952$$
- vi. Irreversibility (I) and Compressor work input (W_c):-

$$I \text{ (KJ/Kg)} = 8.941867691 \cdot (10^{-9}) \cdot (W_c)^5 - 1.472406439 \cdot (10^{-6}) \cdot (W_c)^4 + 8.072378114 \cdot (10^{-5}) \cdot (W_c)^3 + 4.573076963 \cdot (10^{-3}) \cdot (W_c)^2 + 2.217679024 \cdot (10^{-1}) \cdot (W_c) - 1.489896774 \cdot (10^{-1})$$
- vii. Total Exergy change (dEx) and Evaporator temperature (in °C) (T_e) :-

$$dEx \text{ (KJ/Kg)} = 1.48840229 \cdot (10^{-9}) \cdot (T_e)^5 + 8.68162715 \cdot (10^{-8}) \cdot (T_e)^4 + 2.720176417 \cdot (10^{-5}) \cdot (T_e)^3 - 1.784500899 \cdot (10^{-3}) \cdot (T_e)^2 - 4.639927782 \cdot (10^{-1}) \cdot (T_e) + 9.856829368$$

viii. Irreversibility (I) and Evaporator temperature (in °C) (T_e):-

$$I \text{ (KJ/Kg)} = 8.434468893 \cdot (10^{-10}) \cdot (T_e)^5 + 2.277319144 \cdot (10^{-7}) \cdot (T_e)^4 - 3.140560885 \cdot (10^{-5}) \cdot (T_e)^3 + 4.966369882 \cdot (10^{-3}) \cdot (T_e)^2 - 2.827085994 \cdot (10^{-1}) \cdot (T_e) + 3.954175521$$

5. Conclusion

The following conclusions are drawn from the graphs obtained during study.

1. With increase in the evaporator temperature, COP of the system increases with nominal increment in ECOP.
2. With increase in evaporator temperature, Carnot COP and COP both increases, but COP increases more which is numerator part of II law efficiency therefore II law efficiency increases with the increase in evaporator temperature
3. Exergy loss in condenser decreases with increase in evaporator temperature as less amount of heat is to be rejected to the cold fluid. Also there is exergy gain in compressor and evaporator, which decreases with increase in evaporator temperature. However, total exergy change decreases linearly with increase in evaporator temperature.
4. Irreversibility of the system in totality decreases with increase in evaporator temperature as heat rejection in condenser and throttling process in the expansion valve generates high entropy. However it depends on the requirement of the system whether to keep evaporator temperature high at the cost of increased value of reversibility.

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