International Journal of Mechanical Engineering (IJME)
Vol.1, Issue 1 Aug 2012 13-25

International Academy of Science, Engineering and Technology
Connecting Researchers; Nurturing Innovations

SIMULATION OF THREE STAGE VAPOUR COMPRESSION REFRIGERATION CYCLE WITH SINGLE EVAPORATOR AND FLASH INTERCOOLING

ASHUTOSH VERMA

ABSTRACT

The objective of this paper is to develop a novel approach for steady state simulation of multistage vapour compression refrigeration cycles in MATLAB simulink R 2010a. The multistage cycle consists of three compressors operating in series, three expansion valve, two flash intercoolers and single evaporator where refrigerant is R134 a. In this mathematical model, empirical relations for ideal process and polynomials generated for refrigerant R134a (CH_2FCF_3) are used to derive performance parameters of refrigeration cycle. Properties are calculated at all saturation and superheated states after that first law and second law analysis of mentioned refrigeration cycle is performed using state properties. Present work investigates the effects of varying pressure ratio, evaporating temperatures, and condensing temperatures and observes its response on the coefficient of performance, the second law coefficient of performance, second law efficiency and total energy losses. It has been observed that condensing and evaporating temperatures have strong effect on coefficient of performance of cycle. The temperature after the flash intercooler and refrigeration effect has been chosen as constant. The aim is to maximise coefficient of performance and minimize losses.

KEYWORDS: COP, Design and simulation, Efficiency, Exergy Loss.

INTRODUCTION

Simulation of refrigeration systems began to be an attractive topic of publications in the 1980s [1-3], was widely used to evaluate alternatives to CFCs in the 1990s [4,5], and still acts as an effective tool for design of refrigeration systems using environment-friendly working fluids such as carbon dioxide [6-8] in recent years. Models for different kinds of refrigeration systems, including residential air conditioners [2], multi-evaporator air conditioning systems [9], residential heat pumps [1], geothermal heat pumps [10], heat pumps for cold regions [11], automotive air conditioning systems [11,12], chillers [13,14], household refrigerators [4,14], auto cascade refrigeration systems [7], refrigeration systems in shipping containers [16], refrigeration systems using rejecters for performance enhancement [8,12], etc., were published. Simulation has been used for fault detection and diagnostics of HVAC and R systems [15]. The influence of oil on the heat transfer coefficient and the pressure loss of refrigerant flow and on the piston dynamics of hermetic reciprocating compressors used in refrigeration can also be simulated [18,19]. It is impossible to summarize simulation techniques for all kinds of refrigeration systems and their components in a size-limited paper. So only the models for the most important components in commonly used refrigeration systems and those for basic refrigeration systems will be introduced. Fig. 1 shows a basic refrigeration system including two subsystems. The first subsystem is the refrigerant cycle system, including at least a compressor, a condenser, a throttling device and an evaporator. In some actual refrigeration system, an accumulator, a receiver and a filter may be included. The second subsystem is the temperature- keeping system, including at least an envelope structure. In a household refrigerator, this subsystem may include a cabinet, a door seal and some foods inside the cabinet. In an air-conditioned room, this subsystem may include walls, windows, a door and some furniture inside the room. The models for all the components will be discussed. In the illustration of the models of throttling devices, only those for capillary tubes are given because capillary tubes are used widely.

CYCLE DESCRIPTION

Multistage refrigeration systems are widely used where ultra low temperatures are required, but cannot be obtained economically through the use of a single-stage system. This is due to the fact that the compression ratios are too large to attain the temperatures required to evaporate and condense the vapour. There are two general types of such systems: cascade and multistage. The multistage system uses two or more compressors connected in series in the same refrigeration system. The refrigerant becomes more dense vapour whilst it passes through each compressor. Note that a two-stage system can attain a temperature of approximately -65°C and a three-stage about – 100°C. Single-stage vapour-compression refrigerators are used by cold storage facilities with a range of +10°C to -30°C. In this system, the evaporator installed within the refrigeration system and the ice making unit, as the source of low temperature, absorbs heat. Heat is released by the condenser at the high-pressure side.

CYCLE LAYOUT

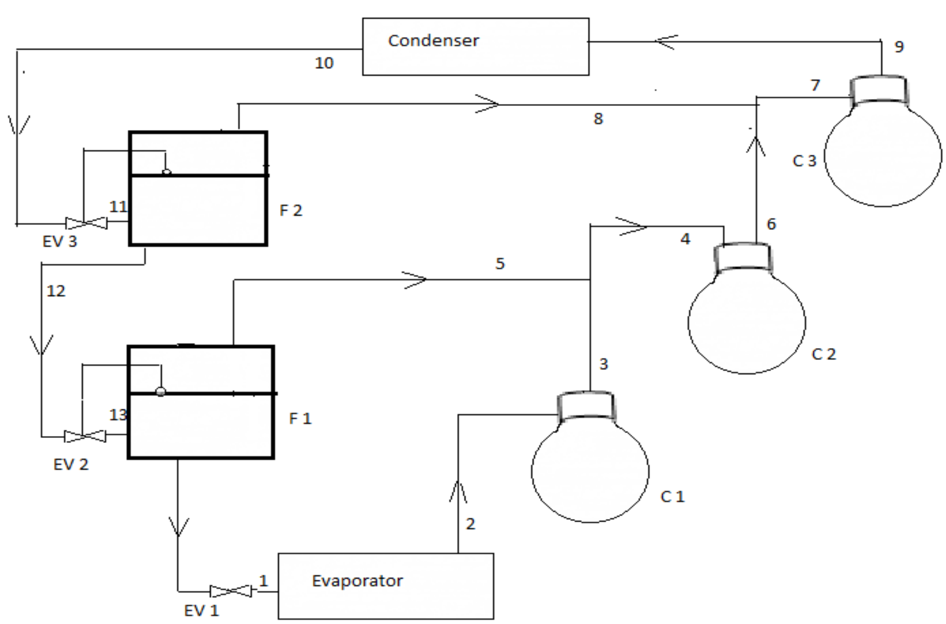


Figure 1

C_1, C_2, C_3 = Compressors

F_1 , F_2 = Flash Intercoolers

EV_1, EV_2 , EV_3 = Flash chamber

Pressure-Enthalpy representation of multistage Vapour Compression Refrigeration Cycle

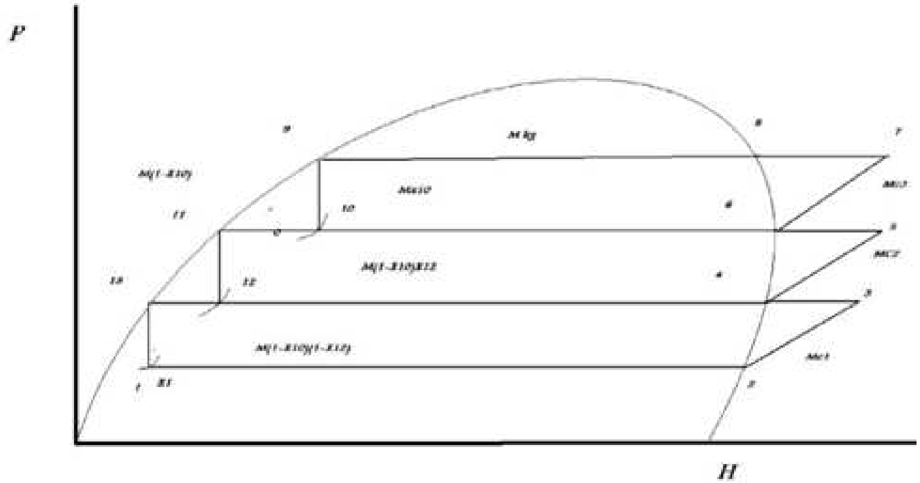


Figure 2

STAEADY STATE MATHEMATICAL MODEL

The mathematical model for "Compound compression with multiple expansion valves in series with flash chamber but no intercooling".

State properties at saturated state are calculated using some polynomials and ideal empirical relations for points at superheated state properties are calculated using basic relations and R134a refrigerant data for superheated region.

Empirical relation used for calculation of properties at saturation states.

1) Temperature of refrigerant in ^{o}C at desired Pressure can be obtained by using polynomial

$$T = p1\times p^5 + p2\times p^4 + p3\times p^3 + p4\times p^2 + p5\times p + p6$$

Where

$p1 = 5.047\times e\text{-}005\ ^{o}C/(Bar)^5$, $p2 = -0.0054231\ ^{o}C/(Bar)^4$, $p3 = 0.21273\ ^{o}C/(Bar)^3$,

$p4 = -3.7581,\ ^{o}C/(Bar)^{2,}$ $p5 = 32.06\ ^{o}C/(Bar)$, $p6 = -67.605\ ^{o}C$

Are coefficients of above equation.

2) Specific heat of vapour refrigerant At desired Pressure in J/kg-k can be

Obtained by using polynomial

$c_{pv} = p_1 \times p^6 + p_2 \times p^5 + p_3 \times p^4 + p_4 \times p^3 + p_5 \times p^2 + p_6 \times p + p_7$

where

p1 = -1.1606e-007J/kg-k-bar^6, p2 = 1.1339e-005 J/kg-K-bar^5,

p3 = -0.00040442 J/kg-K-bar^4, p4 = 0.0065761 J/kg-K-bar^3,

p5 = -0.049401 J/kg-K-bar^2, p6 = 0.19274 J/kg-K-bar, p7 = 0.62284 J/kg-k

Are coefficient of above equation.

3) Enthalpy of vapour refrigerant in kJ/kg can be calculated by

$h_v = p_1 \times p^4 + p_2 \times p^3 + p_3 \times p^2 + p_4 \times p + p_5$ kJ/kg

Where

T is in ^oC, p in Bar, p_1 is in $^oC/bar^5$, p_2 is in $^oC/bar^4$,p_3 is in $^oC/bar^3$

P_4 is in $^oC/bar^2$, p_5 is in ^oC/bar, p_6 is in ^oC.

Are coefficient of above equation

4) Enthalpy of liquid refrigerant at desired Pressure can be calculated using equation .

$h_f = p_1 \times p^4 + p_2 \times p^3 + p_3 \times p^2 + p_4 \times p + p_5$ kJ/kg

Where

p1 = 4.3265e-007 kJ/kg-bar^4, p2 = -6.2499e-005 kJ/kg-bar^3 ,

p3 = 0.0036085 kJ/kg-bar^2, p4 = -0.10636 kJ/kg-bar , p5 = 1.69822 kJ/kg

Are coefficient of above equation

5) Entropy of liquid refrigerant in kJ/kg-K can be calculated using relation

$s_f = p_1 \times p^9 + p_2 \times p^8 + p_3 \times p^7 + p_4 \times p^6 + p_5 \times p^5 + p_6 \times p^4 + p_7$ kJ/kg-K

Where

p1 = 2.0598e-009 kJ/kg-K-bar^9, p2 = -2.9893e-007 kJ/kg-K-bar^8,

p3 = 1.7327e-005 kJ/kg-K-bar^7, p4 = -0.00051214 kJ/kg-K-bar^6,

p5 = 0.0081777 kJ/kg-K-bar^5, p6 = -0.069165 kJ/kg-K-bar^4, p7 = 0.29917 kJ/kg-K

Are coefficient of above equation

6) Entropy of vapour refrigerant in kJ/kg-K can be calculated using relation

$s_v = p_1 \times p^9 + p_2 \times p^8 + p_3 \times p^7 + p_4 \times p^6 + p_5 \times p^5 + p_6 \times p^4 + p_7$ kJ/kg-K

Where

p_1 = -6.9538e-012 kJ/kg-bar^9 -K, p_2 = 1.2841e-009 kJ/kg-bar^8-K ,

p_3 = -9.9679e-008 kJ/kg-bar^5-K , p_4 = 4.2266e-00 kJ/kg-bar^4-K

p_5 = -5.197e-005 kJ/kg-bar^5-K , p_6 = 7.3462e-01 kJ/kg-bar^4-K

p_7 = 2.9489e-006 kJ/kg-K

Are coefficient of above equation

Where as in a case of three-stage refrigeration system with two intercoolers, intermediate flash intercooler pressures Px and Py in terms of inlet and outlet pressures (P_1 and P_2) have been obtained as

$$Px = (P_1^2 \times P_2)^{1/3}, \quad Py = (P_1 \times P_2)^{1/3}$$

For Second law Analysis

Exergy COP or the system

$$\text{ECOP} = \frac{(\lambda in - \lambda out)evaporator}{(\lambda in - \lambda out)c1 + (\lambda in - \lambda out)c2 + (\lambda in - \lambda out)c3}$$

Exergy Efficiency of the system

$$\text{Exergy efficiency} = \frac{W\,rev}{W\,act} = \text{COP / ECOP}$$

Exergy Loss in each Component , based on the following definition of exergy

$$E = (H\text{-}Ho) - To(S\text{-}So)$$

where kinetic and potential energy terms are excluded.

The exergy loss is calculated by making exergy balance for each component of the system and the final expressions are obtained.

1. **Exergy loss for the compressor & condenser can be expressed as**

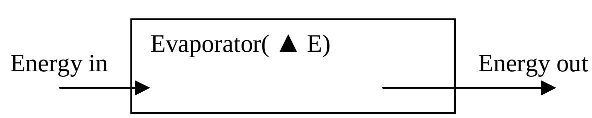
Considering Steady state conditions for control volume, exergy balance results are:

$$\blacktriangle E = [(Hin - Hout) - To(Sout - Sin) + W$$

2. **Exergy loss for the Evaporator**

Exergy loss for the condenser can be expressed as.

Considering Steady state conditions for control volume, exergy balance results are

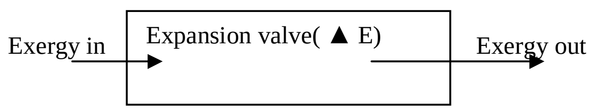


$$\blacktriangle E = We\,(1 - (To/Tr)) = (Hin - Hout)\,To/Tr + To(Sout - Sin).$$

3. Exergy loss for the Expansion Valve

Exergy loss for the Expansion valve can be expressed as.

Considering Steady state conditions for control volume, exergy balance results are

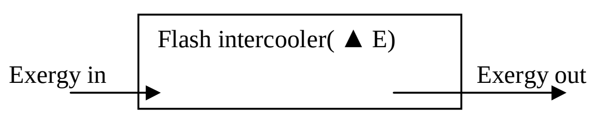


$$\blacktriangle E = [(Hin - Hout) - To(Sout - Sin)$$

4. Exergy loss for the Flash Intercooler Valve.

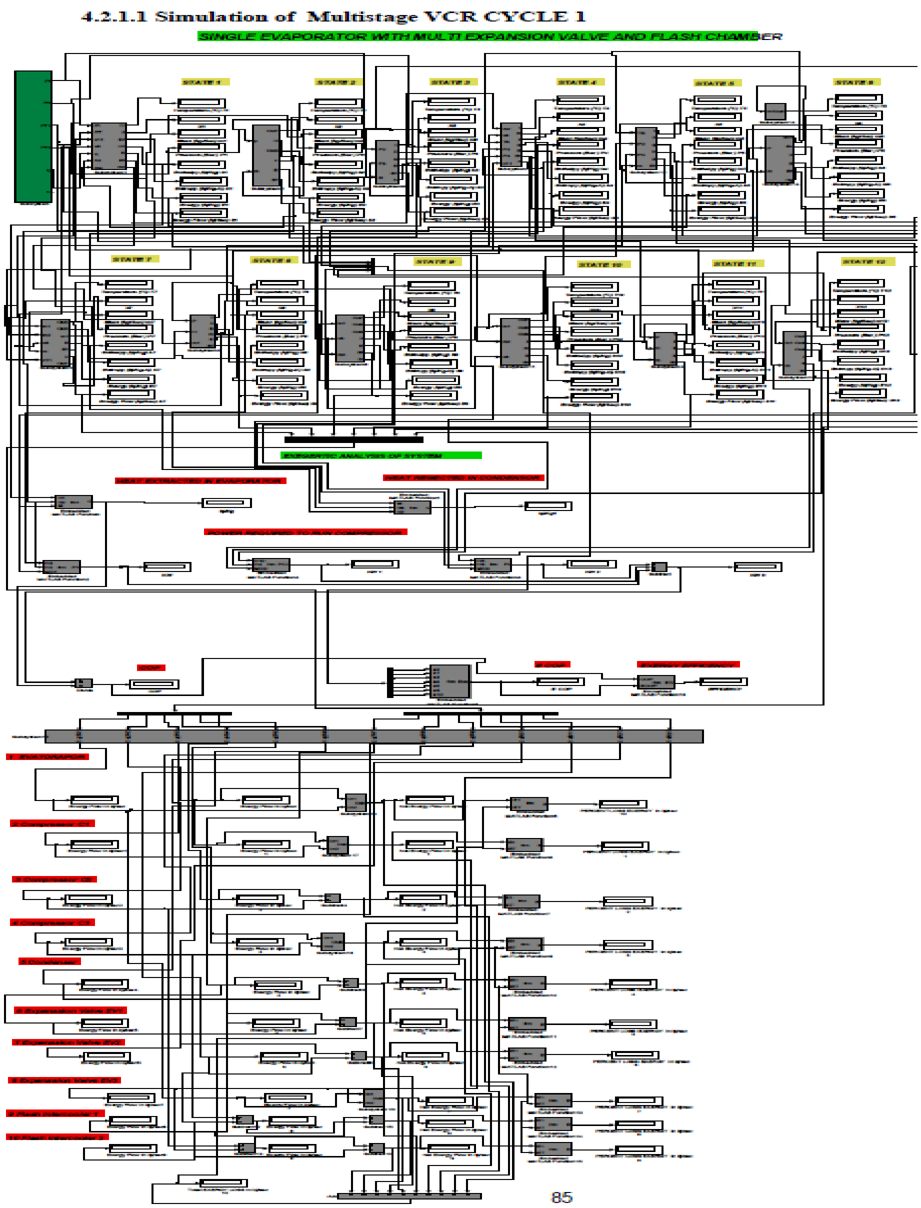
Exergy loss for the Flash intercooler can be expressed as.

Considering Steady state conditions for control volume, exergy balance results are



$$\blacktriangle E = [(H-Ho)-To(S-So)]in - [(H-Ho)-To(S-So)]out = Ein - Eout.$$

Model of Three Stage Vapour Compression Refrigeration Cycle With Single Evaporator And Flash Intercooling developed in MATLAB Simulink R 2010 a.



Graphs and 3d Surface

Results are derived on the basis of variation of following parameters in respond to COP , ECOP , Total Exergy loss & Exergy Efficiency.

1) Pressure ratio (Ph/Pl) from 250 to 400.

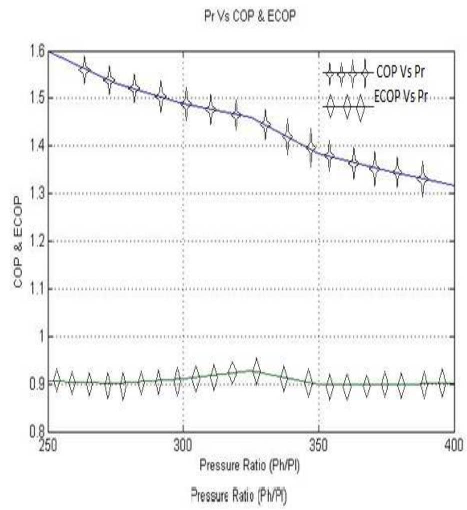


Figure 3

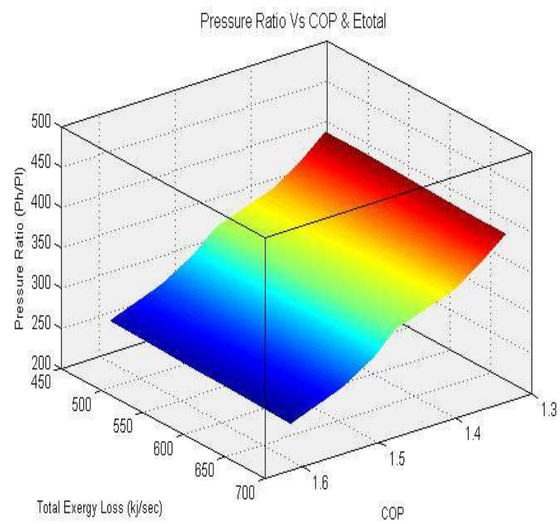


Figure 4

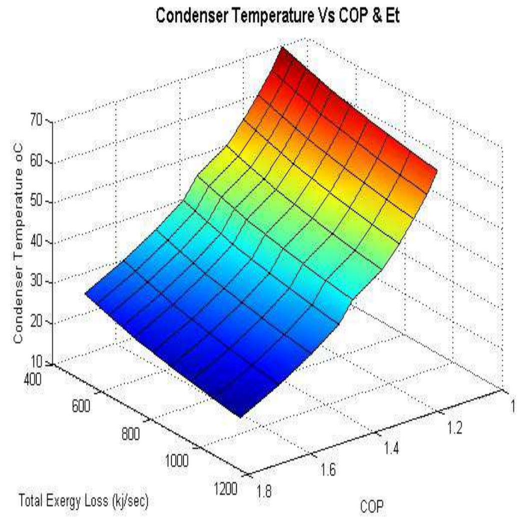


Figure 5

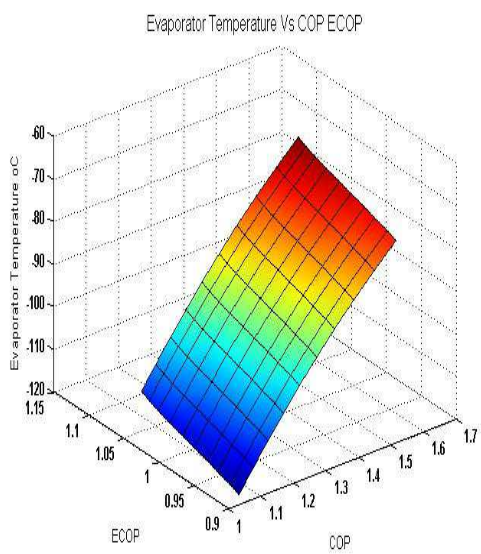


Figure 6

2) Evaporator temperature from -120 ° C to -70 ° C

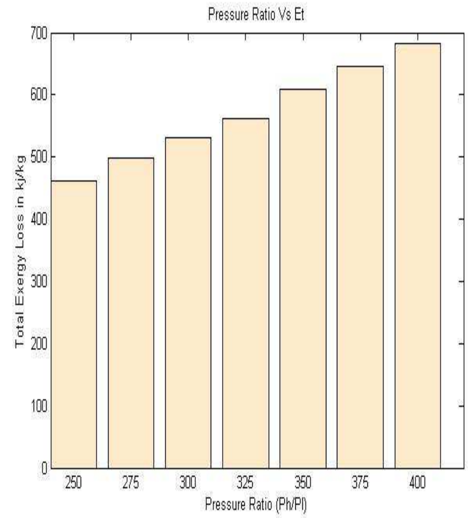


Figure 7

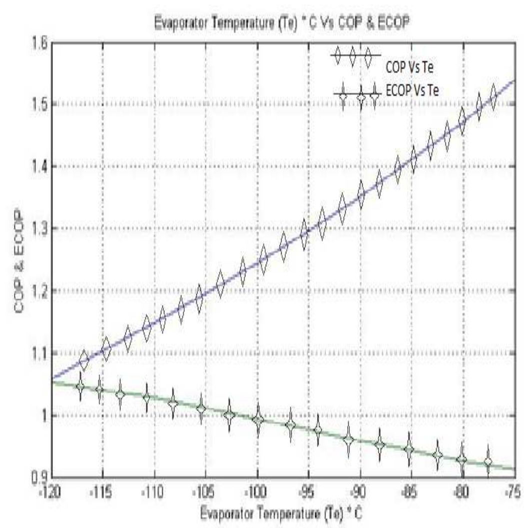


Figure 8

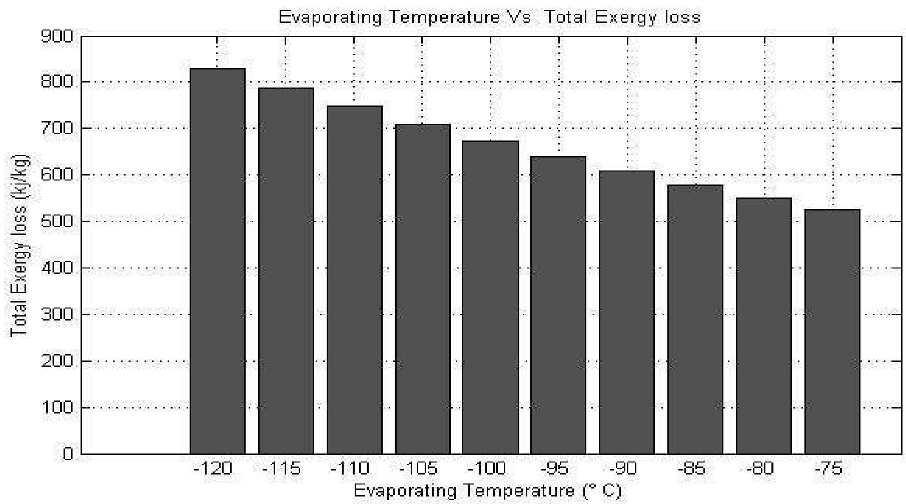


Figure 9

RESULTS AND DISCUSSIONS

As we are increasing pressure ratio COP decreases whereas ECOP follows a well define pattern, this is due to reason COP is inversely proportionally to Pressure ratio (compressor power), and directly proportional to refrigerating effect. ECOP for Refrigeration Cycle is the ratio of exergy flow rate across evaporator to the Summation of exergy flow rate across each compressor. On increasing pressure ratio exergy efficiency of the system increase this is due to reason that these systems are designed for higher pressure ratio therefore maximum efficiency is obtained at higher pressure ratio on further increasing pressure ratio there is sudden fall in second law efficiency. Exergy loss for the given refrigeration system is the loss of available energy which is due to irreversibility of system. On increasing pressure ratio total exergy loss of system increases this is due to increasing input power require to derive the compressor and refrigerating effect. There is increase in mass flow rate of refrigerant which will finally results into increase in disorder of system that is entropy generation. On decreasing evaporator temperature COP of the system increases this mainly due to increase in compressor power and COP is a function of refrigerating effect. As in the case of ECOP there is sudden fall with decrease in evaporator temperature. Study concludes the performance of discussed parameters with the variation of pressure ratio, evaporator temperature and condenser temperature, the behaviour of curves and their assertions. On the basis of these assertions an optimized condition for the working of particular type of system can be decided.

REFERENCES

1. J. Chi, D. Didion, A simulation of the transient performance of a heat pump, International Journal of Refrigeration 5 (3) (1982) 176-184.

2. W.E. Murphy, V.W. Goldschmidt, Cycling characteristics of a residential air conditioner modeling of shutdown transients, ASHRAE Transactions 92 (1A) (1986) 186-202.

1) J.W. MacArthur, E.W. Grald, Unsteady compressible two-phase flow model for predicting cyclic heat pump performance and a comparison with e×perimental data, International Journal of Refrigeration 12 (1) (1989) 29-41.

2) D.S. Jung, R. Radermacher, Performance simulation of a two evaporator refrigerator-freezer charged with pure and mixed refrigerants, International Journal of Refrigeration 14 (5) (1991) 254-263.

3) S.M. Sami, A. Dahmani, Numerical prediction of dynamic performance of vapor-compression heat pump using new HFC alternatives to HCFC-22, Applied Thermal Engineering 16 (8) (1996) 691-705.

4) T. Pfafferott, G. Schmitz, Modelling and transient simulation of CO2-refrigeration systems with Modelica, International Journal of Refrigeration 27 (1) (2004) 42-52.

5) S.G. Kim, M.S. Kim, Experiment and simulation on the performance of an auto cascade refrigeration system using carbon dioxide as a refrigerant, International Journal of Refrigeration 25 (8) (2002) 1093-1101.

6) D. Li, E.A. Groll, Transcritical CO2 refrigeration cycle with ejector-expansion device, International Journal of Refrigeration 28 (5) (2005) 7666-773.

7) R. Shah, A.G. Allen, C.W. Bullard, Dynamic modelling and control of multi-evaporator air-conditioning systems, ASHRAE Transactions 110 (PART 1) (2004) 109e119.

8) T. Chikahisa, H. Matsuo, T. Murayama, Investigations on compact and high-performance heat pumps for cold regions (2nd report, estimation of performance improvement by combining heat storage system with conventional GHP), Transactions of the Japan Society of Mechanical Engineers, Part B 62 (596) (1996) 1591-1598 (in Japanese).

9) J.M. SaizJabardo, W. Gonzales Mamani, M.R. Ianella, Modeling and experimental evaluation of an automotive air conditioning system with a variable capacity compressor, International Journal of Refrigeration 25 (8) (2002) 1157-1172.

10) C.Q. Tian, C.P. Dou, ×.J. Yang, ×.T. Li, Instability of automotive air conditioning system with a variable displacement compressor. Part 2: Numerical simulation, International Journal of Refrigeration 28 (7) (2005) 1111e1123.

11) F.W. Yu, K.T. Chan, Modelling of the coefficient of performance of an air-cooled screw chiller with variable speed condenser fans, Building and Environment 41 (4) (2006) 407-417.

12) M.W. Browne, P.K. Bansal, Transient simulation of vapour compression packaged liquid chillers, International Journal of Refrigeration 25 (5) (2002) 597e610.

13) P.G. Jolly, C.P. Tso, Y.W. Wong, S.M. Ng, Simulation and measurement on the full-load performance of a refrigeration system in a shipping container, International Journal of Refrigeration 23 (2) (2000) 112-126.

14) G. Grazzini, A. Rocchetti, Numerical optimisation of a twostage ejector refrigeration plant, International Journal of Refrigeration 25 (5) (2002) 621-633.

15) B. Yu, D.H.C. Van Paassen, S. Riahy, General modeling for model-based FDD on building HVAC system, Simulation Practice and Theory 9 (6e8) (2002) 387-397.

16) R. Radermacher, L. Cremaschi, R.A. Schwentker, Modelling of oil retention in the suction line and evaporator of air-conditioning systems, HVAC and Research 12 (1) (2006) 35-56.