Thermodynamic Analysis of Air Refrigeration cycle with Double Regeneration

Major project-II

Submitted to Delhi Technological University in partial fulfillment of the requirement for the award of Degree of

Master of Technology

In

Thermal Engineering

Rituraj

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UNDER THE SUPERVISION OF

Mr. P.V Ram Kumar
Associate professor

Mr.Naushad Ahmad Ansari
Assistant professor

Department of Mechanical Engineering
Delhi Technological University
New Delhi-110042

CERTIFICATE

DELHI TECHNOLOGICAL UNIVERSITY
(Formerly DELHI COLLEGE OF ENGINEERING)

Date:-__________

This is to certify that report entitled **"Thermodynamic Analysis of air refrigeration cycle with double regeneration"** by **Rituraj** in the requirement of the partial fulfillment for the award of Degree of **Master of Technology (M.Tech)** in **Thermal Engineering** at **Delhi Technological University**. This work was completed under my supervision and guidance. He has completed his work with utmost sincerity and diligence. The work embodied in this project has not been submitted for the award of any other degree to the best of my knowledge.

Mr. P.V. Ram Kumar

Associate Professor
Department Of
Mechanical
Engineering, DTU

Mr. Naushad Ahmad Ansari

Assistant Professor
Department Of
Mechanical
Engineering DTU

DECLARATION

I declare that the work presented in this thesis titled "**Thermodynamic Analysis of air refrigeration cycle with double regeneration**" ,submitted to Department of Mechanical Engineering, is an authentic record of my own work carried out under the supervision of Associate Professor **Mr. P.V. Ram Kumar** & Assistant Professor **Mr. Naushad Ahmad Ansari**, Department of Mechanical Engineering, Delhi technological university, Delhi.

This report does not, to the best of my knowledge, contain part of my work which has been submitted for the award of any other degree either of this university or any other university without proper citation.

Date:

Place : DTU ,Delhi **Signature of candidate**

ACKNOWLEDGEMENT

First of all, I would like to express my gratitude to God for giving me ideas and strengths to make my dreams true and accomplish this thesis.

To achieve success in any work, guidance plays an important role. It makes us put right amount of energy in the right direction and at right time to obtain the desired result. Express my sincere gratitude to my guide, Associate Professor **Mr. P.V. Ram Kumar** & Assistant Professor **Mr Naushad Ahmad Ansari** , Mechanical Engineering Department for giving valuable guidance during the course of this work, for his ever encouraging and timely moral support.

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RITURAJ

Roll No-2K14/THE/18

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NOMENCLATURE

COP	Coefficient of performance
HE	Heat exchanger
LMTD	Log mean temperature difference
ExDcomp	Exergy destruction in compressor
ExDcooler	Exergy destruction in cooler
ExDhe 1	Exergy destruction in first HE
ExDhe2	Exergy destruction in second HE
ExDcabin	Exergy destruction in cabin
ExDexpander	Exergy destruction in expander
ExDmixingch	Exergy destruction in mixing Chamber
ExDtotal	Total Exergy destruction
T_{exT}	Temprature at exit of turbine
Exdes	Exergy destruction
X_{dt}	Total Exergy destruction
RE	Refrigeration effect

SYMBOLS

Letters	Description	Unit
Q	Heat Energy	KW
U	Overall heat transfer coefficient	$W/m^2.K$
C_P	Specific heat at constant pressure	KJ/Kg.K
e	Effectiveness of heat exchanger	-
β	Pressure Ratio	-
T	Temperature	k
X	Exergy	KJ/Kg
R	Characteristic gas constant	KJ/Kg.K
W	Work	KJ
S	Entropy	KJ
x	Mass fraction	-
A	Area	m^2
η	Efficiency	-

ABSTRACT

With the growing concern of researchers to achieve low temperature and high COP in conventional refrigeration cycles, lots of modifications has been suggested and successfully implemented in the existing refrigeration cycles except air refrigeration cycle where very few of it is available in literatures. So, in the present study for achieving the given objective a double regenerated air refrigeration cycle is considered .Its thermodynamic analysis is done from the point of view of achieving low temperature refrigeration. The exergy destruction of each of the components in the system is determined. In this arrangement the cold air leaving the refrigerated space is first used to cool the air leaving the compressor. Also some fraction of chilled air leaving the turbine is extracted for further cooling of air leaving the first heat exchanger. A simulation analysis is performed by varying the parameter such as mass extracted, the effectiveness of heat exchangers & pressure ratio to get low temperature refrigeration. The simulation is also carried out by varying above parameters to get exergy destruction of each component in the system using EES. It is investigated that by adding another heat exchanger as a regenerator the cabin temperature decreases considerably but there is a slight decrease in the COP of the cycle compared to the conventional regeneration cycle.

The results of simulation are used to study the influence of various operating parameters such as effectiveness of heat exchangers, pressure ratio & percentage of mass extracted low temperature at the exit of turbine in order to get low temperature refrigeration. The results can be useful in the design of heat exchangers as well as control of other thermodynamic parameters to get optimum COP, as well as to get low temperature refrigeration.

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION TO AIR REFRIGERATION SYSTEM

This Machine is one of the earliest types of refrigerator. In air refrigeration cycle, air is used as a refrigerant. Since air does not change its phase therefore its heat carrying capacity is low .It contains a compression cylinder (air compressor) & expansion cylinder (air turbine).The work obtained in turbine are used to drive the compressor (hence external work required is reduced).The plant also contains cooler which contains a number of pipe through which air passes & this pipes are immersed in circulating cooling water. The cold air in cold chamber are direct contacted with cooling space (the atmospheric pressure) thus this system of refrigeration is also called open system of refrigeration (because in compression in next cycle fresh air is compressed).With invention of high speed passenger aircraft, jet aircraft and missiles has introduced the need of compact & simple refrigeration systems capable of high capacity with minimum weight. When weight of the system is taken into consideration, the air cycle system usually prove to be most efficient.

1.2 SIMPLE AIR REFRIGERATION CYCLE

This cycle works on reversed brayton cycle or Bell-Coleman cycle. It consists of a compressor an expander and a cooler (which contains a number of pipes immersed in circulating cooling water).Air is first compressed in a compressor due to to which its temperature increases. The compressed air is then cooled in cooler, the cooled air is expanded in turbine due to which its temperature decreases, and this low temperature air is used for refrigeration purposes.

This cycle has following advantages:-

a) It has very low weight/TR.
b) Since air is used as refrigeration, it can be directly sent to the refrigerated space eliminating the need of heat exchanger (i.e. evaporator).
c) Air is sent in the cabin at 1atm pressure thus it can also fulfill the need of maintaining atmospheric pressure in the cabin.

The open air system working on Bell-Coleman cycle has disadvantage that the air contacted with the cold chamber (space) or stuff to be preserved may collects moisture from such articles kept for preservation & this moisture freeze during expansion & there is possibility of valve chocking. Thus the same Bell-Coleman cycle may be worked with closed air system; the air may pass through the coils & extract heat from the cold chamber without coming direct contact in it.

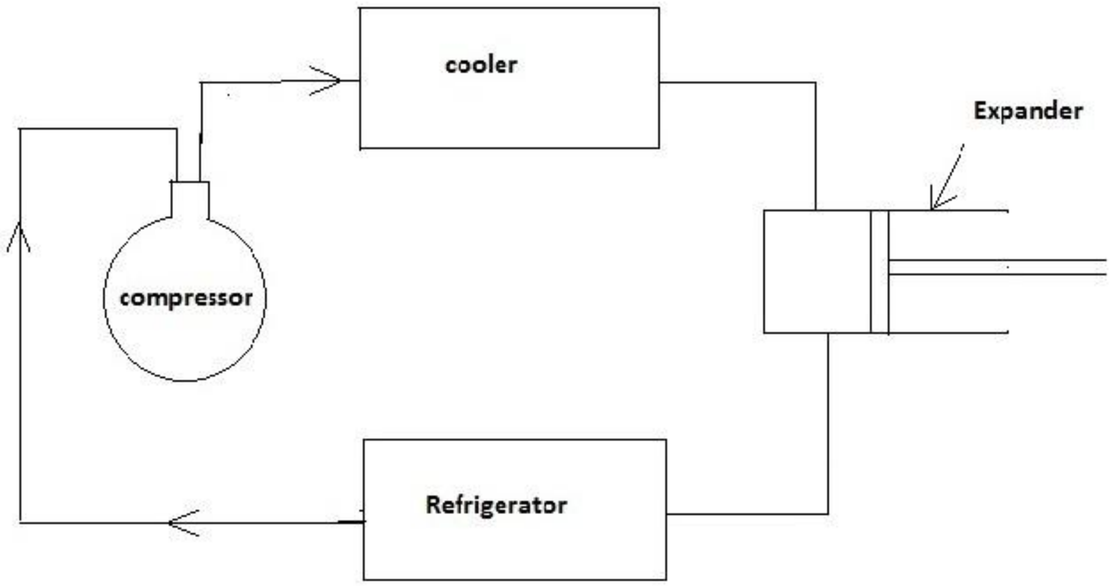

Fig: 1.1 Bell-Coleman Cycle

1.3 AIR REFRIGERATION CYCLE WITH SINGLE REGENERATION

In this system we use a heat exchanger to pre cool the air before entering into the turbine. Air is 1st compressed from state point 1 to 2. The hot compressed air is then cooled to atmospheric temperature in a heat exchanger of 100% efficiency, thus the temperature of air becomes equal to that of surrounding. This air is further cooled in an heat exchanger of effectiveness(ϵ_r) by using the cool air leaving the refrigerated space .The cooled air is then expanded in turbine due to which its temperature is decreased (as $\Delta H = f(T)$).This cooled air is then used to remove the heat from refrigerated space.

The circuit can be either opened towards ambient (so that a hot side heat exchanger is eliminated) or towards the refrigerated space (so that cold side heat exchanger is eliminated).Due to elimination of both side heat exchangers irreversibility related to heat transfer process can be eliminated & thus exergy loss can be prevented .Also frictional loss in the heat exchanger is prevented. The weight, volume& thus the cost of system is reduced. In the 2nd case i.e. elimination of evaporator leads to avoidance of frost. Also if the air is introduced at high speed then electric fan fitted on evaporator can be eliminated.

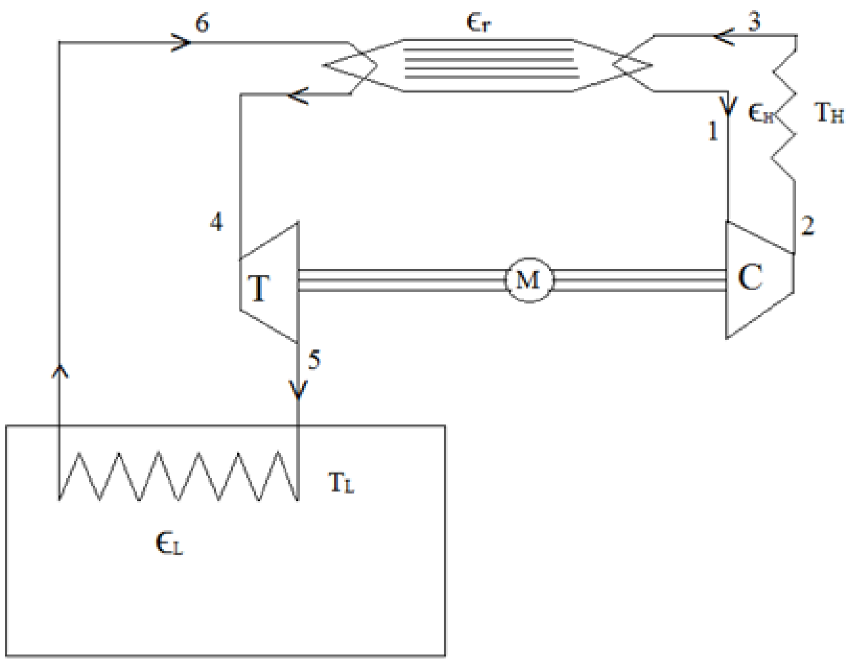

Fig: 1.2 Air refrigeration cycle with single regeneration[20]

1.4 AIR REFRIGERATION CYCLE WITH DOUBLE REGENERATION

In this system we are using 2 regenerators. Air is first compressed in a compressor; the compressed air is then cooled in a heat exchanger (of 100% efficiency) to environment temperature. This air is then cooled simultaneously in 2 heat exchangers of effectiveness e_1 & e_2 respectively. The cooling air comes from 2 sources:-

a) Air after leaving the refrigerated space is very cold & this air is used as a cooling fluid in 1st heat exchanger.
b) Air after leaving the turbine is at very low temperature & a part of it is bled to the 2nd heat exchanger.

The air leaving the refrigerated space & the air leaving 2nd heat exchanger are mixed together in a mixing chamber , this air is then send to 1st heat exchanger. Due to multiple regeneration temperature at the entry of refrigerated space is very low, Thus Refrigerated space (cabin) can be maintained at very low temperature.

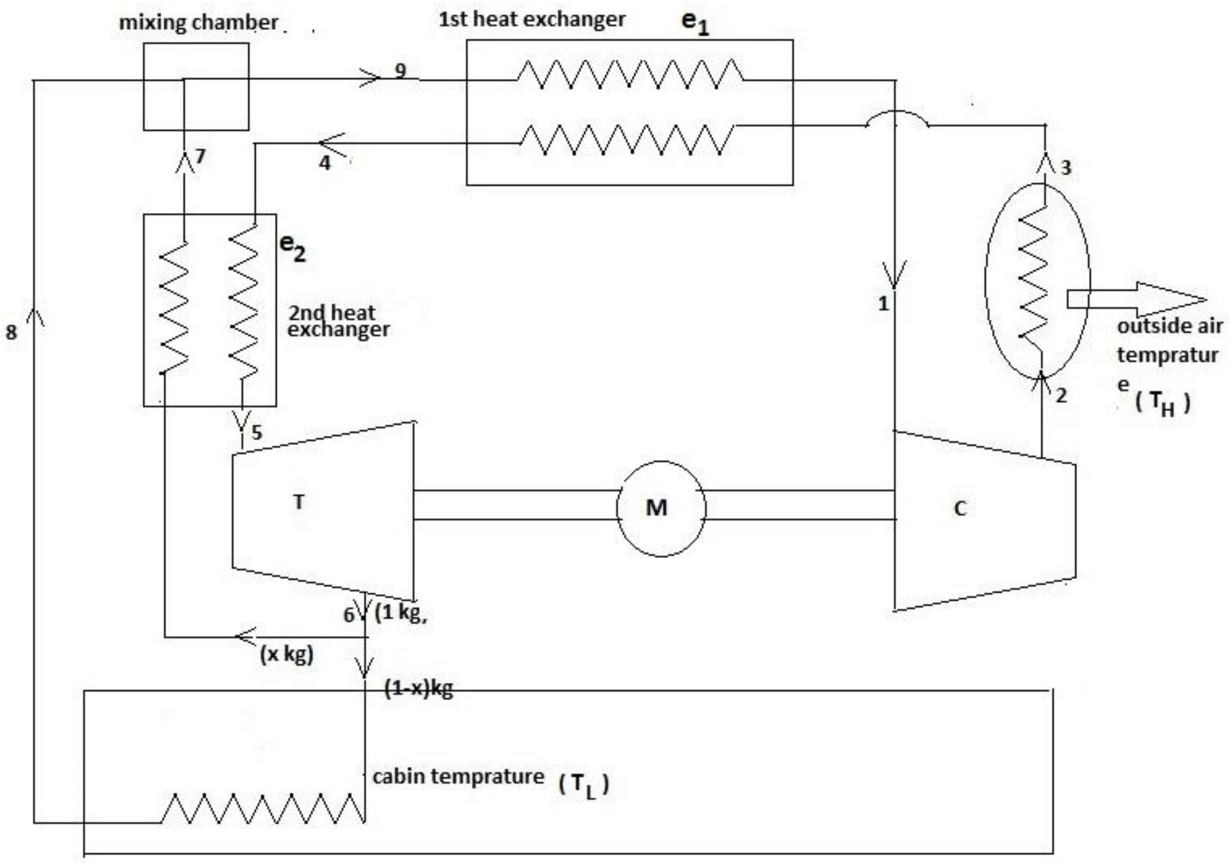

Fig: 1.3 Air refrigeration cycle with double regeneration.

CHAPTER 2

LITERATURE REVIEW

In recent years many work has been carried out on air refrigeration system to obtain low temperature refrigeration .In 2006 Shaobo Hou & Hefei Zhang carried out thermodynamic analysis of open reverse Brayton cycle with regeneration by using moist air & he also proved its feasibility via performance simulation. The cooling of wet air leaving the compressor is done by using pinch technology .The refrigeration mainly depends upon latent heat of water vapour & sensible heat of air. He stated that adoption of this system will make deep freeze easily & a very low temperature (-55°c) can be produced. [1]

In 1995 Blanchard, Klein &Agrawal applied the theory of finite-time thermodynamics to the fundamental Carnot refrigeration cycle. This paper will extend the concept to analyze the performance of an internally reversible Brayton refrigerator with the effect of real heat exchangers.[20]

In 1996 Gigiel has extensively analysed the various advantages of an regenerated air cycle over a conventional plant for the food freezing system, focusing on the enlarged range of accessible operating conditions, the improved temperature and humidity control, lower temperatures and high freezing rate attainable, as well as reduced capital and maintenance costs.[20]

In 1997 CHIH HU analyzed the heat exchanger effect on the power input of a gas refrigerator with a given cooling load. The finite-time thermodynamic model employed in a Brayton cycle coupled to a heat sink and a heat sauce along with both the heat exchangers. Both the heat source and heat sink have infinite heat capacity rates. Mathematical expressions for smallest power input and other important characteristics of the refrigerator at minimum power are obtained for the cycle. The results of this theoretical work present the base line criteria for use in the performance evaluation and design of such refrigeration cycles as well as for use in the performance comparisons with existing refrigerators.[10]

In 2006 Chen et al investigated the cooling load vs cop characterstics of regenerated air cycles with irreversibilities & heat transfer losses .He carried out Optimizations of air cycles using finite-time thermodynamics (FTT) & entropy-generation minimization.

In 2007 Luo et al. optimized the cooling-load and the COP of a irreversible air refrigeration cycle of simple type by searching for the optimum pressure-ratio of the compressor and the optimum distribution for heat conductance of the cold as well as hot side heat exchangers for the fixed total heat-exchanger inventory.[8]

In 2008 Hou and Zhang presented a paper on an enhanced system, using open reversed Brayton cycle with regeneration by using moist air for deep freeze cooled by using circulating water. In this open reversed Brayton cycle with regeneration using moist air for deep freeze cooled by circulating water, the air leaving the freeze chamber is used to cool water, and then this cooled water is further used to cool the wet air before entry to the turbine . So, that we can get lower wet air temperature before the turbine and higher COP.

In 2009 an irreversible cycle model of the quantum Brayton refrigeration cycle using an ideal Bose or Fermi gas as the working substance was established. Based on the theory of statistical mechanics and thermodynamic properties of ideal quantum gases, terminology for various important performance parameters such as the cooling rate, coefficient of performance as well as of power input, are obtained. The effect of the degeneracy of the quantum gases, the internal irreversibility of the working substance and the influence of finite-rate of heat transfer between the working substance and the heat reservoirs on the best possible performance of the cycle is investigated. By using numerical solutions, the cooling rate of the cycle is optimized for a set of specified parameters. The highest cooling rate and the corresponding parameters are calculated numerically.[13]

In 2010 Yue Zhang carried out many work on regenerated & non regenerated Brayton refrigeration cycle He concluded that the irreversibilities are coming mainly from the internal dissipation of the working material in the adiabatic processes and the finite-rate heat transfer in the regenerative and constant pressure processes.

In 2011 Zincan chen investigate the synthesis influence of the various irreversibility's, which especially include the finite-rate heat transfer, on the optimal performance of the quantum Brayton cryogenic refrigeration cycle. In the present paper, the optimal performance characteristics of the irreversible Brayton cryogenic refrigeration cycle composed of two adiabatic and two constant pressure processes are investigated. The working substance of the cycle consists of ideal Fermi or Bose gases. The general expressions of several important parameters such as the cooling rate, power input and coefficient of performance are derived. The performance characteristics of the cycle are revealed. The curves of the optimal relation between the cooling rate and the coefficient of performance are obtained. The optimal performances of the cycle are discussed in detail. The optimum criteria of some important parameters are obtained.[12]

In 2013 Zhang et al performed thermodynamic analysis on regenerated air cycle focusing on low temperature refrigeration. The effect of very high effective heat exchanger is shown exploring that it may double the air cycle COP. Low & high pressure configurations are compared.[15]

In 2013 using classical kays and London data the feasibility of very high regenerator effectiveness is explored. The effect of high regenerator effectiveness (0.95) is shown, showing that it may double the air cycle COP. A thermodynamic analysis is performed on regenerated air

Cycle focusing on low temperature refrigeration. Low and high pressure configurations (i.e. consisting of only one heat exchanger on the cold or hot cycle side) are compared. High pressure configuration, which are normally preferred in the case of low temperature refrigeration, has a lower efficiency, but the gap reduces as the effectiveness of regenerator is increased. prevention of a cold heat exchanger eliminates frosting troubles and power required for the circulation fan, so that the small COP decrease as suffered due to the high pressure cycle can easily be remunerated for and eventually yield higher average plant efficiency.

In 2103 N.Giannetti & A. Milazzo performed thermodynamic analysis on regenerated air cycle focusing on low temperature refrigeration. The effect of very high effective heat exchanger is shown exploring that it may double the air cycle COP. Low & high pressure configurations are compared.

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CHAPTER 3

THERMODYNAMIC ANALYSIS OF AIR REFRIGERATION SYSTEM

3.1 AIR REFRIGERATION CYCLE ASSUMPTIONS

i. Air is considered as an ideal gas.
ii. The specific heats (C_p &C_v) of air is assumed to be constant between the working range of temperatures.
iii. The analysis is made under steady state conditions.
iv. Heat losses & pressure drops in the tubing & the components are considered negligible.
v. System is internally reversible (i.e. there is no friction).

3.2 SIMPLE AIR REFRIGERATION CYCLE ANALYSIS:-

In actual systems, these heat exchangers have limited sizes, not substantially large. In particular, to achieve rates of heat transfer with limited sized heat exchangers requires the temperatures (Tt and T3) of the working fluid of the heat exchangers to be different from the temperatures (Ta and TL) of the surrounding thermal reservoirs, as shown in Fig 3.1. In order to find the higher bound for the performance characteristics and input power of actual Brayton refrigerators, let us consider an internally reversible Brayton refrigeration cycle.

An internally reversible Brayton refrigeration cycle is defined as the process in which the two heat transfer processes (Qa and Q_L) from the system to the heat sink and from the heat source to the system are the only irreversible processes in the cycle and in which both the heat exchangers have limited sizes. Since the system is internally reversible, it provides the highest coefficient of performance and also it requires the minimum power input as required by any actual refrigerator with the same amount of cooling load. To simplify our analysis we can consider the working fluid of the Brayton refrigeration cycle to be an ideal gas with specific heats constant at all temperatures. To analyze such an internally reversible Brayton refrigerator, we will apply Fourier's heat transfer law, the second and 1st law of thermodynamics, the log mean temperature differences definition, effectiveness of heat exchangers, and the definition of coefficient of performance of the refrigerator as follows.

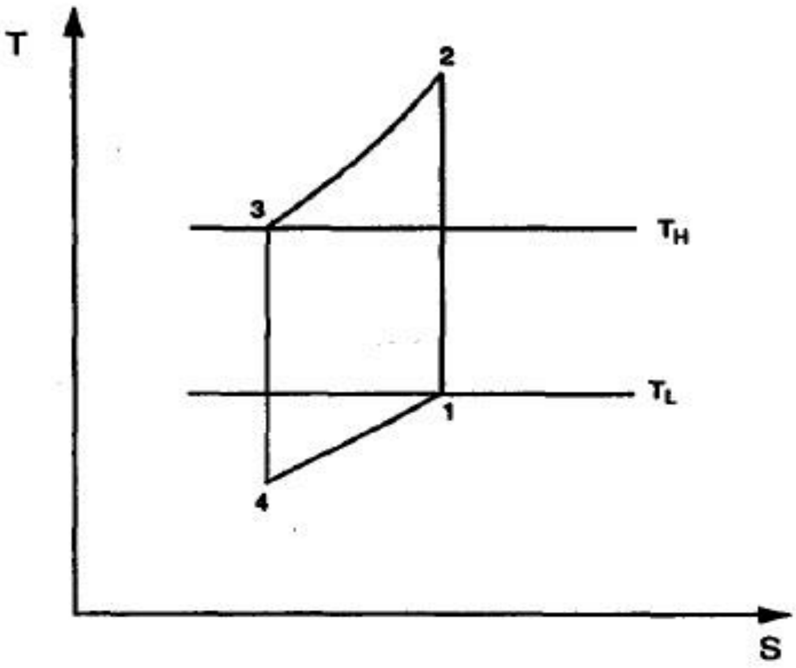

Fig 3.1 Reversible Brayton refrigerator with ideal heat exchangers[18]

3.3 GOVERNING EQUATIONS FOR SIMPLE SYSTEM

Heat transfer equation:-

For Cooler

$Q_H = U_H A_H (LMTD)_H$

$Q_H = mc_p(T_2 - T_3)$

For Refrigerator

$Q_L = U_L A_L (LMTD)_L$

$Q_L = mc_p(T_1 - T_4)$

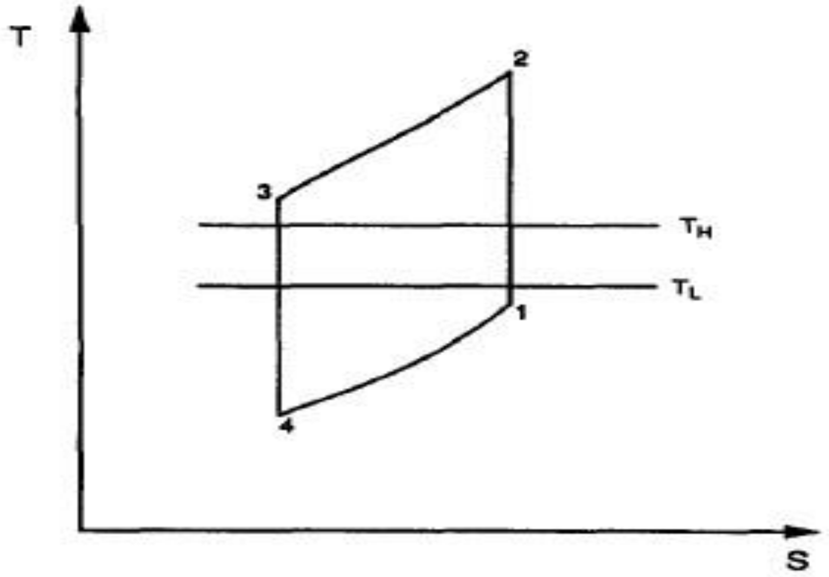

Fig 3.2 Endoreversible Brayton refrigerator with real heat exchangers.[18]

3.4 **GOVERNING EQUATIONS FOR REALISTIC SYSTEM**

$(LMTD)_H = \frac{(T_2-T_3)}{\ln[(T_2-T_H)/(T_3-T_H)}$,$(LMTD)_L = \frac{(T_1-T_4)}{\ln[(T_L-T_4)/(T_L-T_1)}$

For Cooler

$Q_H = e_H(Q_H)_{max}$

Where $(Q_H)_{max} = mc_p(T_2 - T_H)$

For Refrigerator

$Q_L = e_L(Q_L)_{max}$

Where $(Q_L)_{max} = mc_p(T_L - T_4)$

Effectiveness of 1st heat exchanger

$e_H = (T_2 - T_3)/(T_2 - T_H)$

Effectiveness of 2nd heat exchanger

$e_L = (T_1 - T_4)/(T_L - T_4)$

3.5 ANALYSIS OF AIR REFRIGERATION CYCLE WITH SINGLE REGENERATION:-

In this system we use a heat exchanger to pre cool the air before entering into the turbine . Air is 1st compressed from state point 1 to 2. The hot compressed air is then cooled to atmospheric temperature in an heat exchanger of 100% efficiency, thus the temperature of air becomes equal to that of surrounding. This air is further cooled in an heat exchanger of effectiveness(ϵ_r) by using the cool air leaving the refrigerated space .The cooled air is then expanded in turbine due to which its temperature is decreased (as $\Delta H=f(T)$).This cooled air is then used to remove the heat from refrigerated space.

The circuit can be either opened towards ambient (so that a hot side heat exchanger is eliminated) or towards the refrigerated space (so that cold side heat exchanger is eliminated).Due to elimination of both side heat exchangers irreversibility related to heat transfer process can be eliminated & thus exergy loss can be prevented .Also frictional loss in the heat exchanger is prevented. The weight, volume& thus the cost of system is reduced. In the 2nd case i.e. elimination of evaporator leads to avoidance of frost. Also if the air is introduced at high speed then electric fan fitted on evaporator can be eliminated.

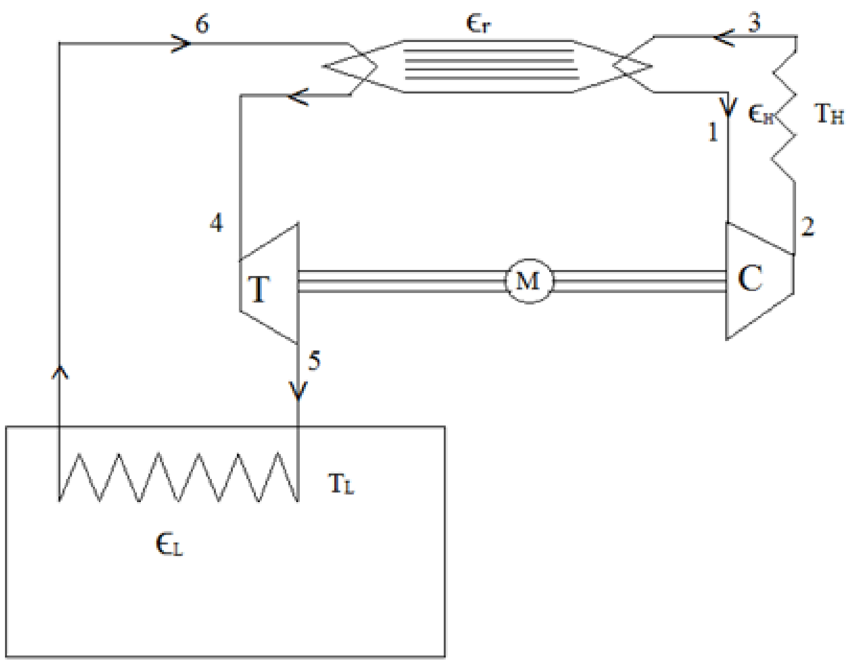

Fig 3.3 Air refrigeration cycle with single regeneration[20]

Air is continuously drawn from ambient temperature and pressure and sent to a regenerative heat exchanger, where it cools down before expanding in a turbine. Then it flows through a heat exchanger, where it absorbs the cooling load from the cold space and goes back to the other side of the regenerator. A compressor draws air from the system back to the ambient, which plays the role of the hot heat exchanger.

3.6 GOVERNING EQUATIONS FOR SINGLE REGENERATION

We can write Carnot's COP as:-

$$\eta_C = \frac{T_L}{(T_H - T_L)} \qquad \text{.............................(1)}$$

$$\theta = \frac{\dot{Q}_L}{\dot{m}c_p(T_H - T_L)} \qquad \text{.............................(2)}$$

Where $\theta = Dimensionless\ cooling\ load$

Temprature Ratio Can be Related to Compression Ratios ($\beta = \frac{P_H}{P_L}$) as

$$\frac{T_2}{T_1} = \beta^{\lambda_C} \quad \& \quad \frac{T_4}{T_5} = \beta^{\lambda_e} \qquad \text{.............................(3)}$$

Where $\lambda_c = \frac{\lambda}{\eta_p}$ & $\lambda_e = \lambda\eta_p$ ....(4)

η_p =polytropic efficiency & $\lambda = \frac{R}{C_P}$=0.2857 (Treating air as an ideal gas)

The Effectiveness Of Regenerator can be written as

$$\epsilon_r = \frac{T_3 - T_4}{T_3 - T_6} = \frac{T_1 - T_6}{T_3 - T_6} \qquad \text{.............................(5)}$$

In order to simplify the analysis, ideal heat exchangers are considered on the hot and cold Side ($\epsilon_H = \epsilon_L = 1$) so that $T_3 = T_H$ & $T_6 = T_L$

From equation (1) to (5) an explicit expression of compression ratio is obtained as

$$\beta = \left[\frac{\eta_c - \epsilon_r + 1}{\eta_c - \theta}\right]^{\frac{1}{\lambda_e}} \qquad \text{.............................(6)}$$

Equation 6 allows direct calculation of COP as a function of all desired parameters

$$\text{COP} = \frac{\dot{Q}_L}{\dot{m}C_p[T_1(\beta^{\lambda c} - 1) - T_5(\beta^{\lambda e} - 1)]}$$

$$=\frac{\theta}{(\epsilon_r+\eta_c)\left[\left(\frac{\eta_c-\in_r+1}{\eta_c-\theta}\right)^{\frac{\lambda_c}{\lambda_e}}-1\right]+\in_r-1-\theta} \qquad \ldots\ldots\ldots\ldots\ldots(7)$$

3.7 ANALYSIS OF AIR REFRIGERATION CYCLE WITH DOUBLE REGENERATION:-

In this system we are using 2 regenerators. Air is first compressed in a compressor; the compressed air is then cooled in a heat exchanger (of 100% efficiency) to environment temperature. This air is then cooled simultaneously in 2 heat exchangers of effectiveness e_1 & e_2 respectively. The cooling air comes from 2 sources:-

a) Air after leaving the refrigerated space is very cold & this air is used as a cooling fluid in 1st heat exchanger.
b) Air after leaving the turbine is at very low temperature & a part of it is bled to the 2nd heat exchanger.

The air leaving the refrigerated space & the air leaving 2nd heat exchanger are mixed together in a mixing chamber , this air is then send to 1st heat exchanger. Due to multiple regeneration temperature at the entry of refrigerated space is very low, Thus Refrigerated space (cabin) can be maintained at very low temperature.

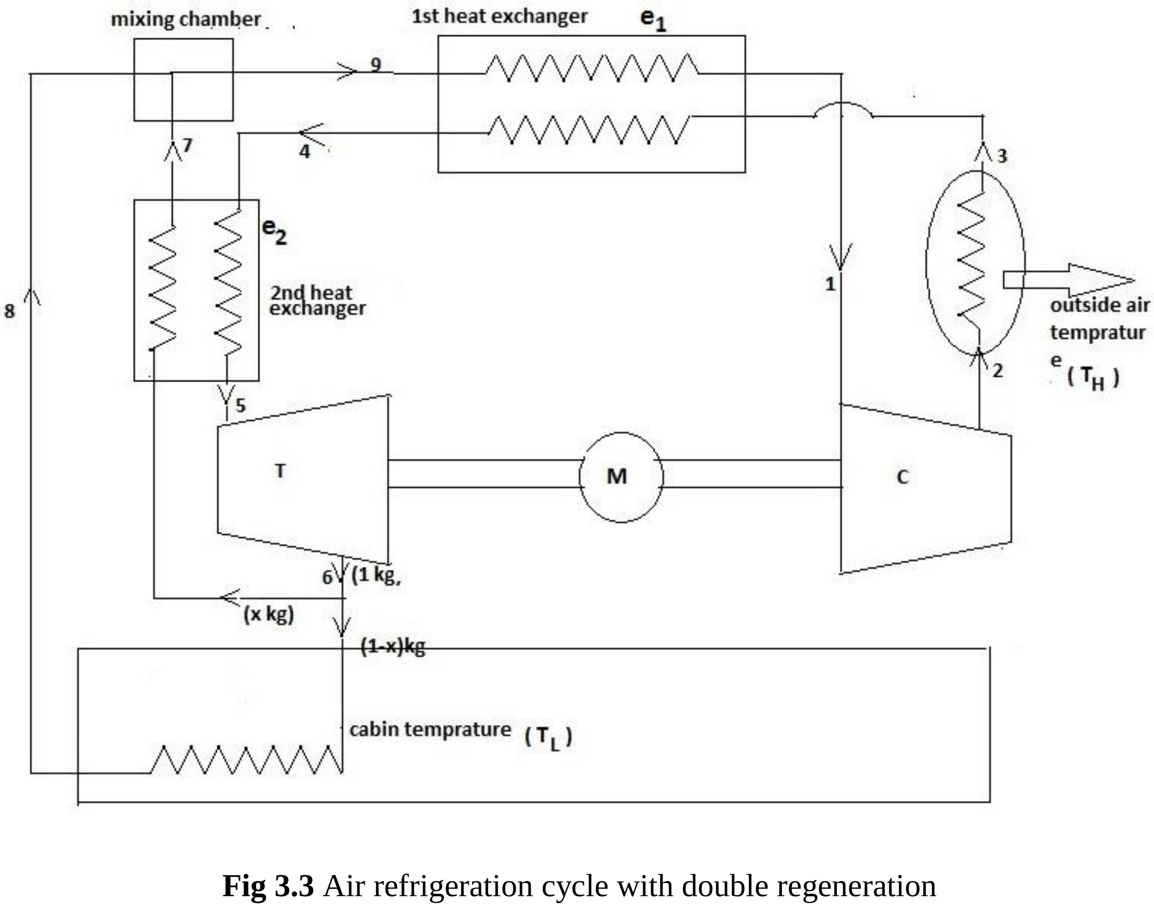

Fig 3.3 Air refrigeration cycle with double regeneration

3.8 GOVERNING EQUATIONS FOR DOUBLE REGENEATION

Pressure ratio of both compressor & expander can be written as

$$\beta = \frac{P_H}{P_L}$$

$$\alpha = \frac{charecterstics\ gas\ constant}{specific\ heat\ at\ constant\ pressure}$$

$$\alpha = \frac{R}{C_P}$$

$$\eta_P = Polytropic\ efficiency$$

For compression process

$$\alpha_c = \frac{\alpha}{\eta_p}$$

$$\frac{T_2}{T_1} = \beta^{\alpha_c}$$

For heat transfer to atmosphere (assuming 100% efficiency)

$$T_3 = T_H$$

For expansion process

$$\frac{T_5}{T_6} = \beta^{\alpha_e}$$

Effectiveness of 1st heat exchanger

$$e_1 = \frac{T_3 - T_4}{T_3 - T_9}$$

Energy balance of 1st heat exchanger

$$T_3 - T_4 = T_1 - T_9$$

Effectiveness of 2nd heat exchanger

$$e_2 = \frac{T_7 - T_6}{T_4 - T_6}$$

Energy balance of 2nd heat exchanger

$$T_4 - T_5 = x * (T_7 - T_6)$$

Energy balance of mixing chamber

$(1\text{-x})*T_8 + \text{x}*T_7 = T_9$

For refrigerated space i.e.Cabin

$T_8 = T_L$

Refrigeration effect can be written as

$\text{R.E} = (1\text{-x})*C_p * (T_8 - T_6)$

Work done on compressor

$W_c = C_p * (T_2 - T_1)$

Work done by expander

$W_T = \quad C_p * (T_5 - T_6)$

Net work input

$W_{in} = C_p * (T_2 - T_1) - C_p * (T_5 - T_6)$

Coefficient of performance can be written as

$$\text{COP} = \frac{(1-\text{x})*C_p* \ (T_8 - T_6)}{C_p*(T_2 - T_1) - C_p*(T_5 - T_6)}$$

3.9 EXERGY ANALYSIS OF AIR REFRIGERATION CYCLE WITH DOUBLE REGENERATION:

In the present work, a parametric study with various temperatures has been conducted to determine the performance evaluation of air refrigeration system with double regeneration. The following assumptions are made to simplify exergy analysis.

1. All components are assumed to be a steady flow and steady-state process.

2. The changes in the kinetic energy and the potential energy of the components are negligible.

3. The pressure drops and heat loss in the piping connecting the components are negligible.

4. Heat exchangers are internally reversible

5. Pressure drops in condenser and evaporator are neglected.

1) Exergy analysis of compressor

$S_{gen}=\Delta S_{sys} + \Delta S_{sur}$

$$s_{g1} = c_p \cdot \ln\left[\frac{T_2}{T_1}\right] - R \cdot \ln(b)$$

$X_{d1}=T_0 * S_{gen}$

2) Exergy analysis of cooler

$S_{gen}=\Delta S_{sys} + \Delta S_{sur}$

$$s_{g2} = c_p \cdot \ln\left[\frac{T_3}{T_2}\right] + \frac{c_p \cdot (T_2 - T_3)}{T_3}$$

$X_{d2}=T_0 * S_{gen}$

3) Exergy analysis of 1st Heat exchanger

$S_{gen}=\Delta S_{sys} + \Delta S_{sur}$

$$s_{g3} = c_p \cdot \ln\left[\frac{T_4}{T_3}\right] + c_p \cdot \ln\left[\frac{T_1}{T_9}\right]$$

$X_{d3}=T_0 * S_{gen}$

4) Exergy analysis of 2nd Heat exchanger

$S_{gen}=\Delta S_{sys} + \Delta S_{sur}$

$$s_{g4} = c_p \cdot \ln\left[\frac{T_5}{T_4}\right] + x \cdot c_p \cdot \ln\left[\frac{T_7}{T_6}\right]$$

$X_{d4}=T_0 * S_{gen}$

5) Exergy analysis of Turbine

$S_{gen}=\Delta S_{sys} + \Delta S_{sur}$

$$s_{g5} = c_p \cdot \ln\left[\frac{T_6}{T_5}\right] - R \cdot \ln\left[\frac{1}{b}\right]$$

$X_{d5}=T_0 * S_{gen}$

6) Exergy analysis of Refrigerator

$S_{gen}=\Delta S_{sys} + \Delta S_{sur}$

$$\dot{s}_{g6} = (1-x)\cdot c_p \cdot \ln\left[\frac{T_8}{T_6}\right] - \left[\frac{(1-x)\cdot c_p\cdot(T_8-T_6)}{T_8}\right]$$

$X_{d6}=T_0 * S_{gen}$

7) Exergy analysis of Mixing Chamber

$S_{gen}=\Delta S_{sys} + \Delta S_{sur}$

$$\dot{s}_{g7} = (1-x)\cdot c_p \cdot \ln\left[\frac{T_9}{T_8}\right] + x\cdot c_p \cdot \ln\left[\frac{T_9}{T_7}\right]$$

$X_{des7}=T_0 * S_{gen}$

8) Total Exergy destroyed=Exergy destroyed in compressor + Exergy destroyed in cooler + Exergy destroyed in 1^{st} heat exchanger + Exergy destroyed in 2^{nd} heat exchanger + Exergy destroyed in turbine + Exergy destroyed in refrigerator(cabin) + Exergy destroyed in mixing chamber

$E_{dt} = X_{d1} + X_{d2} + X_{d3} + X_{d4} + X_{d5} + X_{d6} + X_{d7}$

CHAPTER 4

RESULTS AND DISCUSSION

4.1 THERMODYNAMIC ANALYSIS

For performing thermodynamic analysis of complete cycle, certain state point values have been assumed such as:

Mass flow rate of working fluid (air) =1 kg/s

Outside air temperature, T_3=30˚C

Cabin temperature T_L= -30˚C

Pressure ratio, β=5

R=0.287 (KJ/Kg.k)

C_P=1.005

e_1=0.85

e_2=0.85,$\eta_p = 0.8$

x=0.4

The atmospheric temperature & pressure are assumed to be T_0=30˚C & P_0= 101.3 Kpa

Thermodynamic properties are calculated by utilizing all the equations & assumptions

IN SINGLE REGENERATION

T_{texs}=174.5 K

Cop=0.4548

IN DOUBLE REGENERATION

T_{texd}=148.7 K

Cop=0.4332

ExDcomp=34.99 KJ/kg

ExDcooler=54.14 KJ/kg

ExDhe1=2.161 KJ/kg

ExDhe 2=10.32 KJ/kg

ExDexpander=27.99 KJ/kg

ExDcabin=18.81 KJ/kg

ExDmixingch=0.0534 KJ/kg

ExDtotal=148.5. KJ/kg

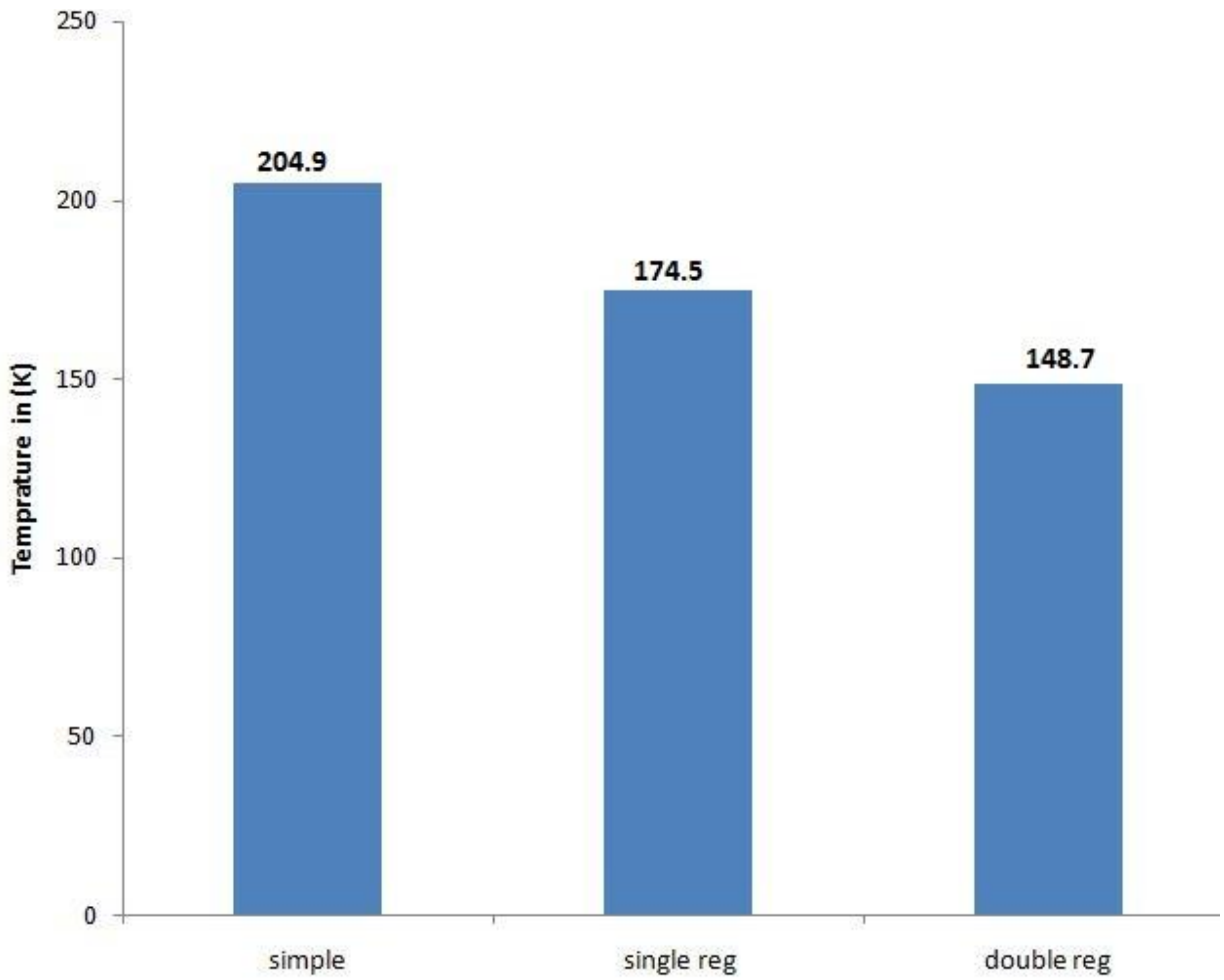

Fig 4.1 Temperature at exit of turbine (in K) in the three systems

It is clear from the above that to achieve deep freeze refrigeration system temperature at the exit of expander should be very low. In simple system (without using regeneration) this temperature is high so deep freeze is not possible. In double regeneration by controlling the mass fraction very low temperature can be produced. By extracting 40 % of mass for regeneration, temperature of (148.7 K) is produced at turbine exit, so deep freeze condition can be achieved inside the cabin.

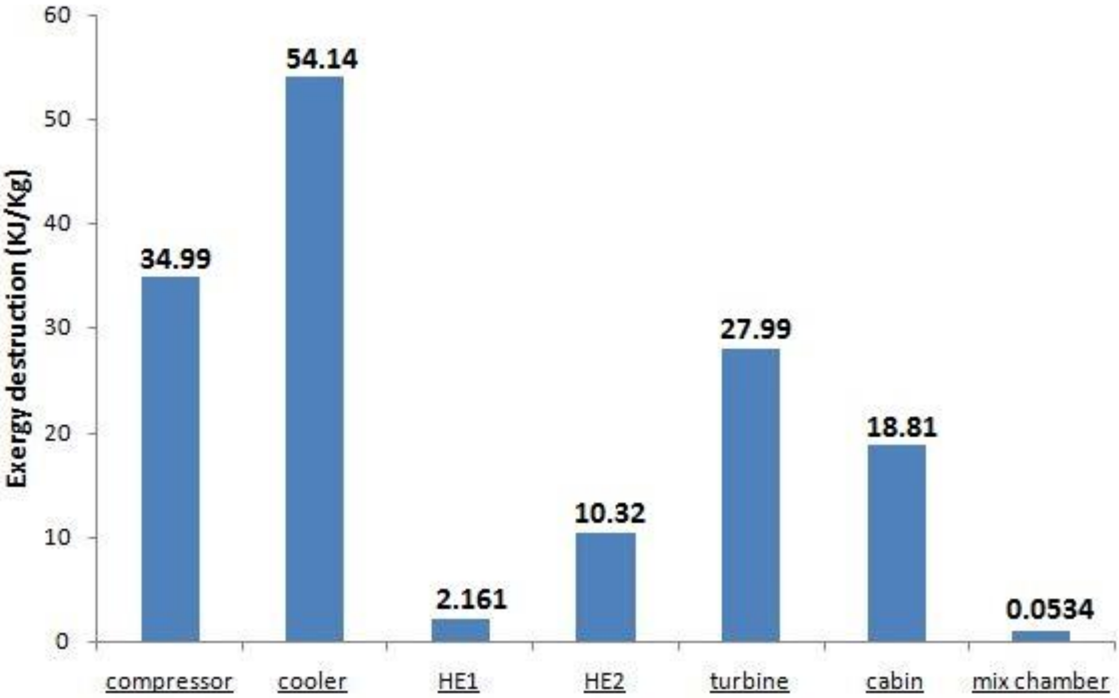

Fig 4.2 Exergy destruction (in KJ/Kg) of each component in system.

It is clear from the above figure that almost 36% of total exergy is wasted in cooler .Exergy destructed in cooler is highest (54.14) after that it is high in compressor (34.99). These components should be thus designed properly in order to enhance its exergy utilization.

The other components in decreasing order of exergy destruction are turbine, cabin, 2^{nd} heat exchanger, 1st heat exchanger & mixing chamber with exergy destruction values 27.99, 18.81, 10.32, 2.161 & 0.0534 KJ/Kg respectively

Although mixing chamber is highly irreversible but its exergy destruction value is coming low because we have neglected the pressure head loss in the connecting pipes.

4.2 VARIATION OF COP WITH DIFFERENT PARAMETERS

Variation of COP with pressure ratio

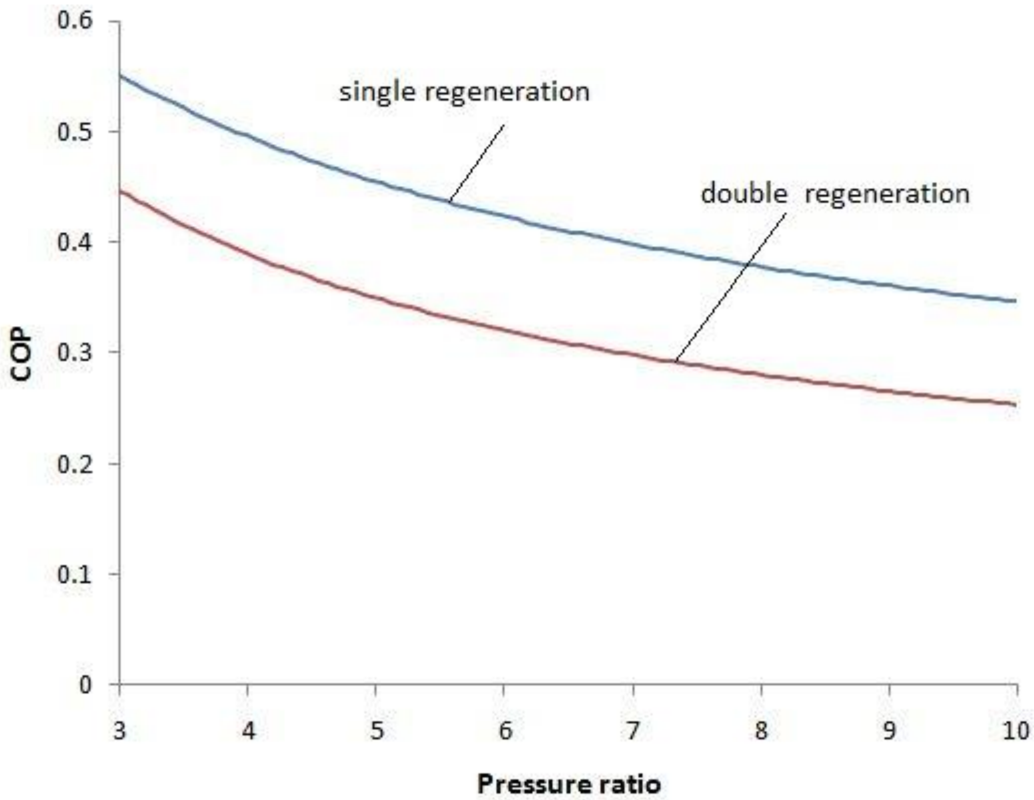

Fig 4.3 COP vs. pressure ratio for single & double regeneration system

From fig it is clear that COP decreases with increase in pressure ratio. In both single & double regeneration system the rate of decrease are almost same (∵ the slope is same).At pressure ratio 3 the value of COP is 0.55 in single regeneration system & 0.45 in double regeneration system. At high pressure ratio of 10 the value of COP is 0.39 in single regeneration system & 0.29 in double regeneration system .

Variation of COP with effectiveness of regenerator

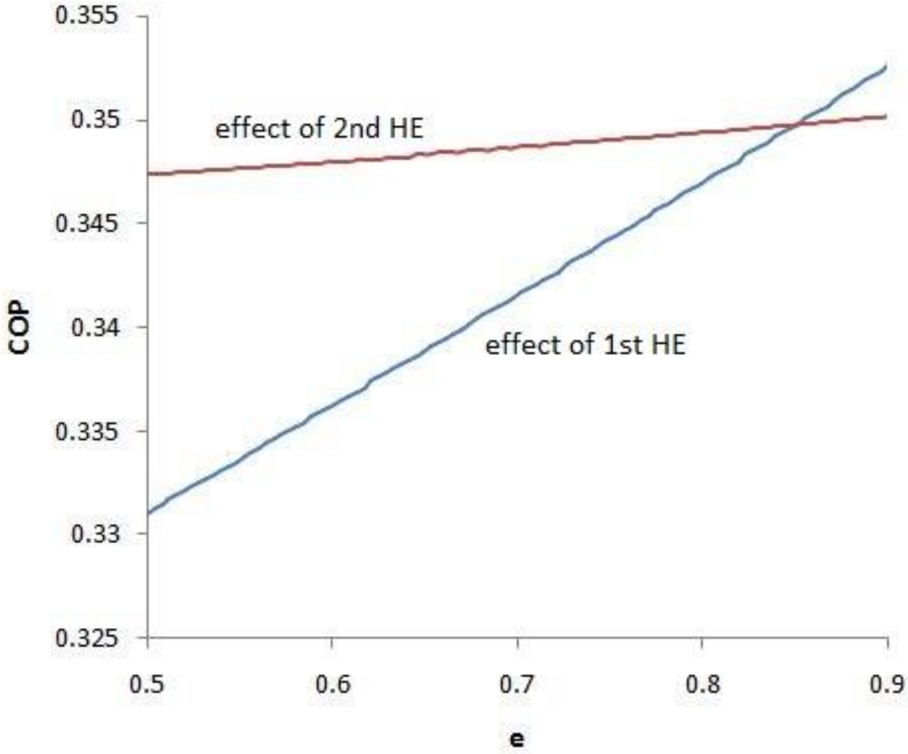

Fig 4.4 COP vs effectiveness of 1st & 2nd HE in double regeneration system

From the graph of COP vs effectiveness it can be seen that by increasing the effectiveness of either the 1st heat exchanger or the 2nd heat exchanger the COP increases. With increase of the effectiveness of first heat exchanger the COP increase at a high rate (linear increase). With increase of effectiveness of 2nd heat exchanger also the COP is increasing .However the rate of increase is negligible as compared the increase in first heat exchanger.

Variation of COP with percentage of mass extracted for 2nd regenerator

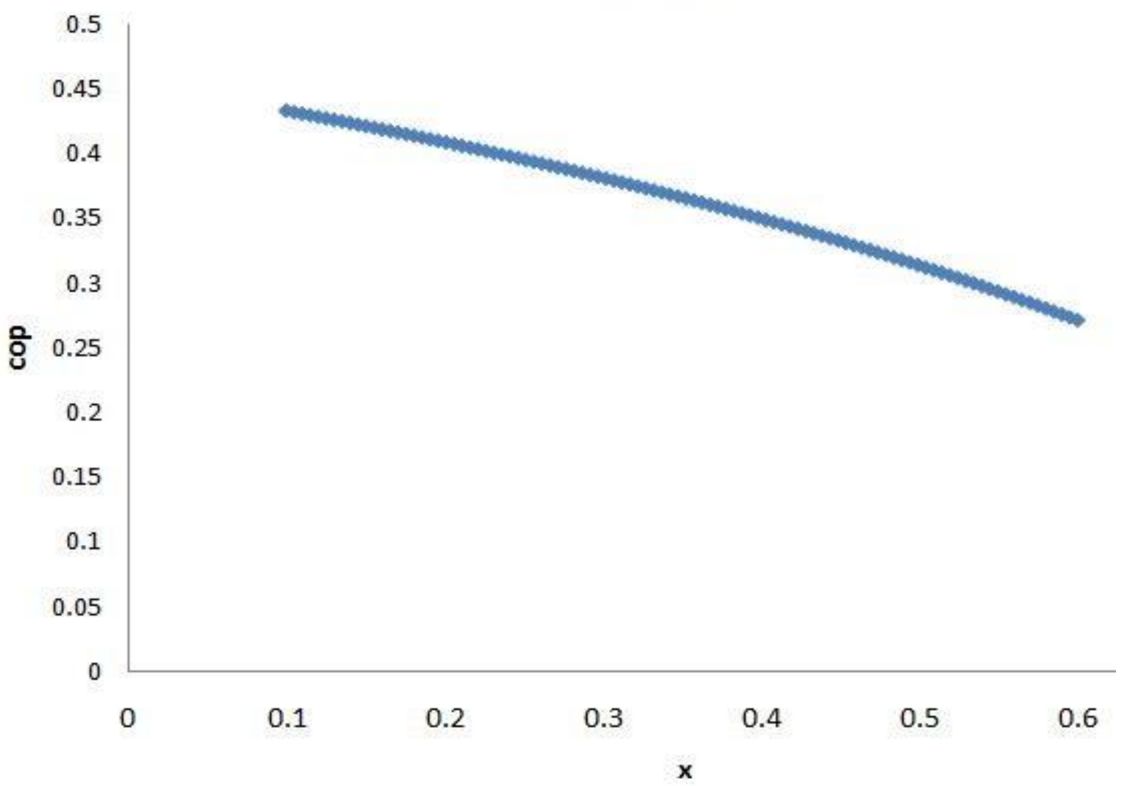

Fig 4.5 COP vs percentage of mass extracted for 2nd regenerator

From fig it is obvious that as the percentage of mass extracted in the 2nd regenerator goes on increasing the COP goes on reducing. However to obtain low temperature refrigeration more & more mass has to be extracted for the 2nd regeneration process. If 10 % of mass is extracted the COP is 0.4332 & If 60 % of mass is extracted the COP is 0.271.

4.3 VARIATION OF TURBINE EXIT TEMPRATURE WITH DIFFERENT PARAMETERS

Variation of Turbine exit temperature with effectiveness of regenerator

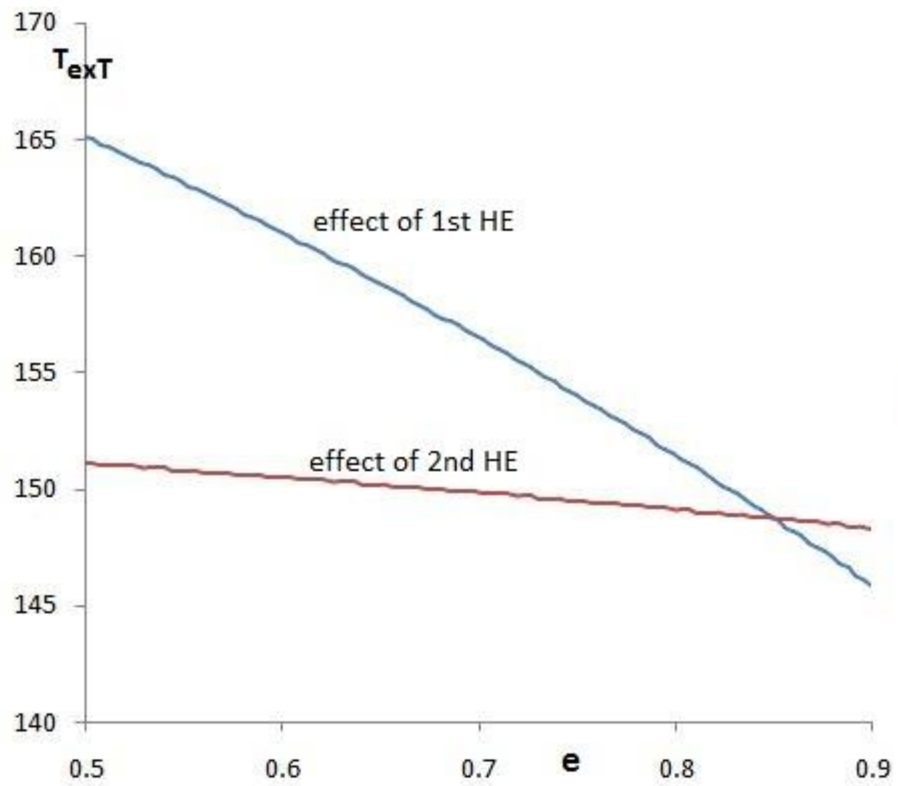

Fig 4.6 Turbine exit temperature vs effectiveness of heat exchanger

As the effectiveness of heat exchanger increases the turbine exit temperature decreases .With increase of effectiveness of 1st heat exchanger the rate of decrease in turbine exit temperature is much more as compared to that of 2nd heat exchanger(as the slope in 1st case is more steep).

Variation of Turbine exit temperature with percentage of mass extracted for second stage regeneration

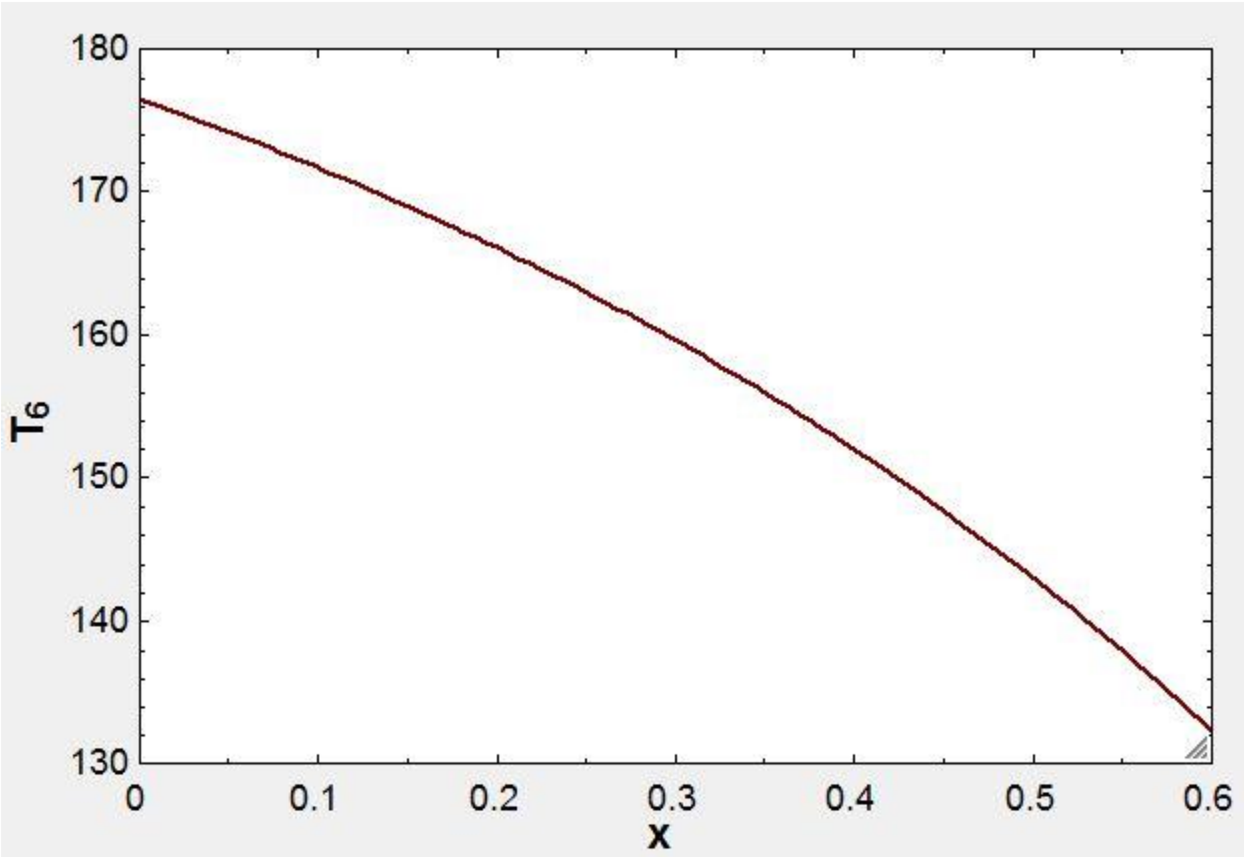

Fig 4.7 Turbine exit temperature vs percentage of mass extracted for 2nd heat exchanger

As the mass fraction extracted becomes more & more the turbine exit temperature decreases more & more rapidly. Although the numerical value of COP goes on decreasing but the decrease in turbine exit temperature will make deep freeze possible.

If 10 % of mass is extracted the turbine exit temperature is 170.1 k & If 60 % of mass is extracted the turbine exit temperature is 133.1 k.

4.4 VARIATION OF EXERGY DESTRUCTION OF THE ENTIRE COMPONENT WITH DIFFERENT PARAMETERS

Variation of Exergy destruction of all the components with effectiveness of HE 1

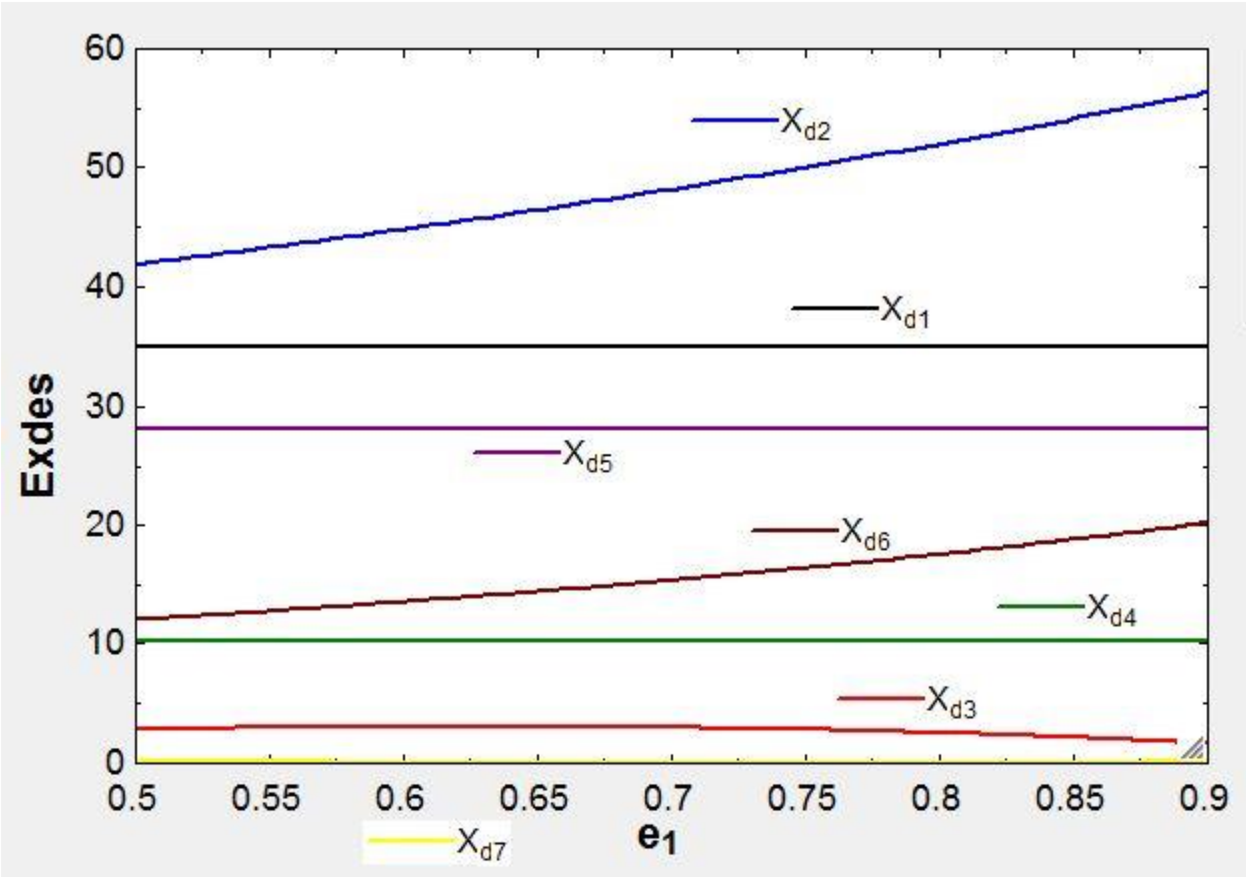

Fig 4.8 Exergy destruction of individual components vs effectiveness of first HE

From the above figure it is clear that as the effectiveness of 1st heat exchanger increases the exergy destruction in each component shows different trend.

In compressor, heat exchanger 2 & turbine exergy destruction remains constant with the change in effectiveness of 1st heat exchanger.

The rate of increase of exergy destruction is highest in cooler, after that it is in cabin & then heat exchanger 1 & mixing chamber respectively.

Although mixing chamber is highly irreversible but its exergy destruction value is coming low because we have neglected the pressure head loss in the connecting pipes.

<u>Variation of Exergy destruction of all the components with effectiveness of HE</u> 2

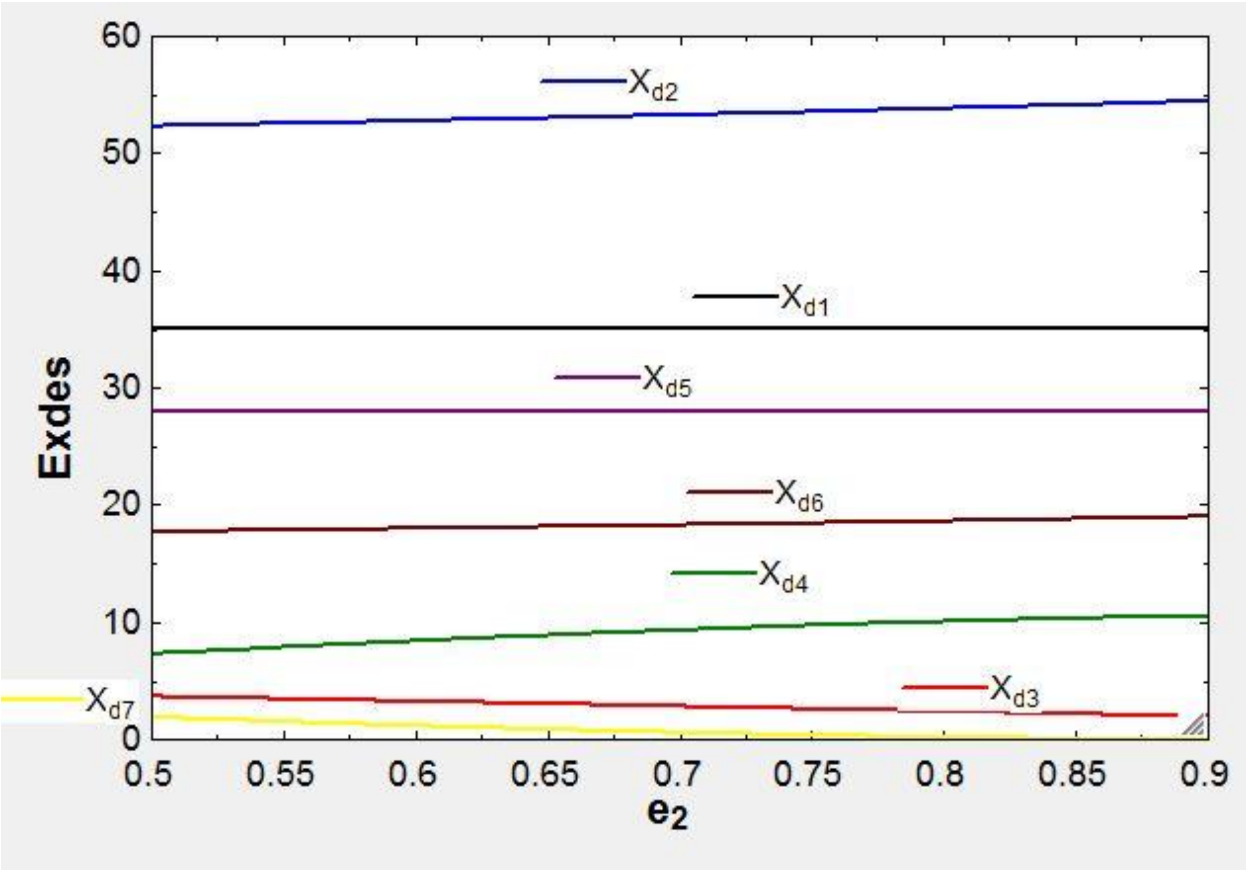

Fig 4.9 Exergy destruction of individual components vs effectiveness of second HE

From the above figure it is clear that as the effectiveness of 2nd heat exchanger increases the exergy destruction in 1st heat exchanger & mixing chamber goes on reducing, In compressor & turbine it is constant & in remaining components it goes on increasing

The rate of increase of exergy destruction is highest in cooler, after that it is in cabin & heat exchanger 2 respectively.

Although mixing chamber is highly irreversible but its exergy destruction value is coming low because we have neglected the pressure head loss in the connecting pipes.

Variation of Exergy destruction of all the components with percentage of mass extracted for 2nd state regeneration

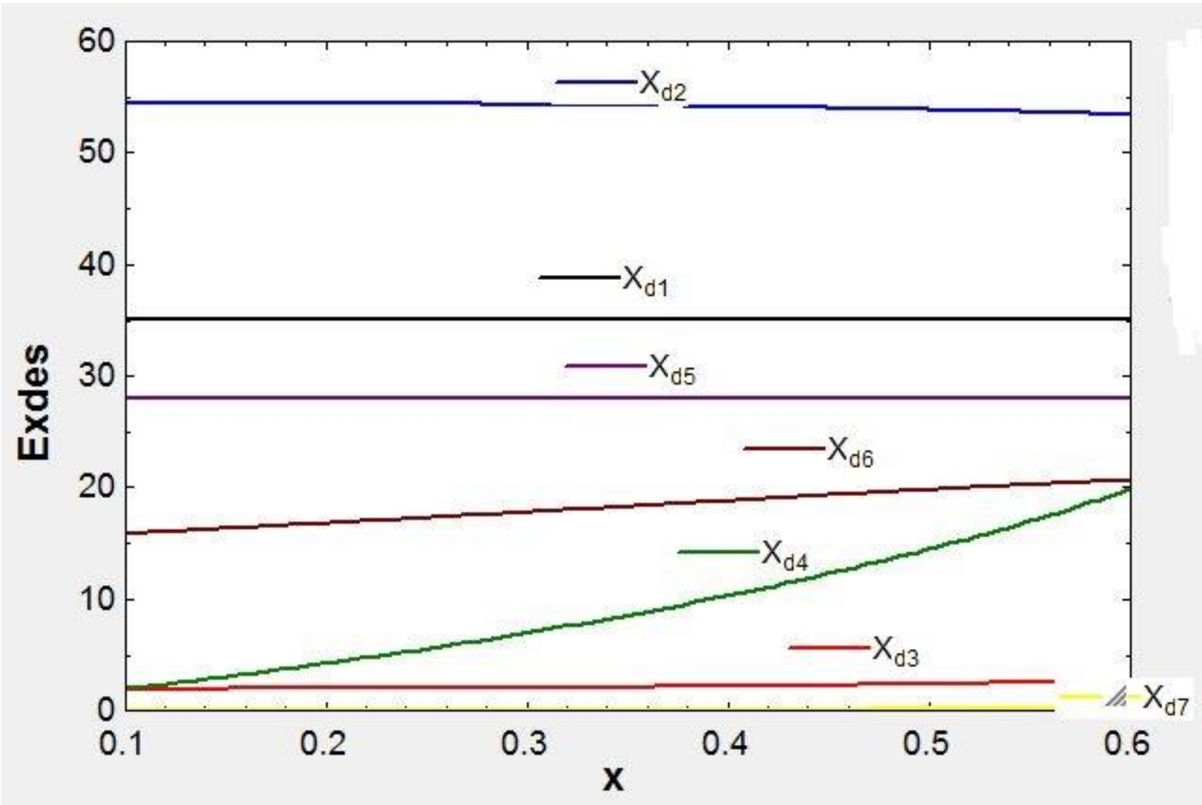

Fig 4.10 exergy destruction of individual components vs percentage of mass extracted

From the above figure it is clear that as the effectiveness of 1st heat exchanger increases the exergy destruction in each component shows different trend.

The exergy destruction in compressor & turbine remains constant.

The exergy destruction in HE 1 , HE 2, cabin & Mixing chamber increases. The rate of increase is highest in heat exchanger 2.

The exergy destruction in cooler decreases.

Variation of Exergy destruction of all the components with change in pressure ratio.

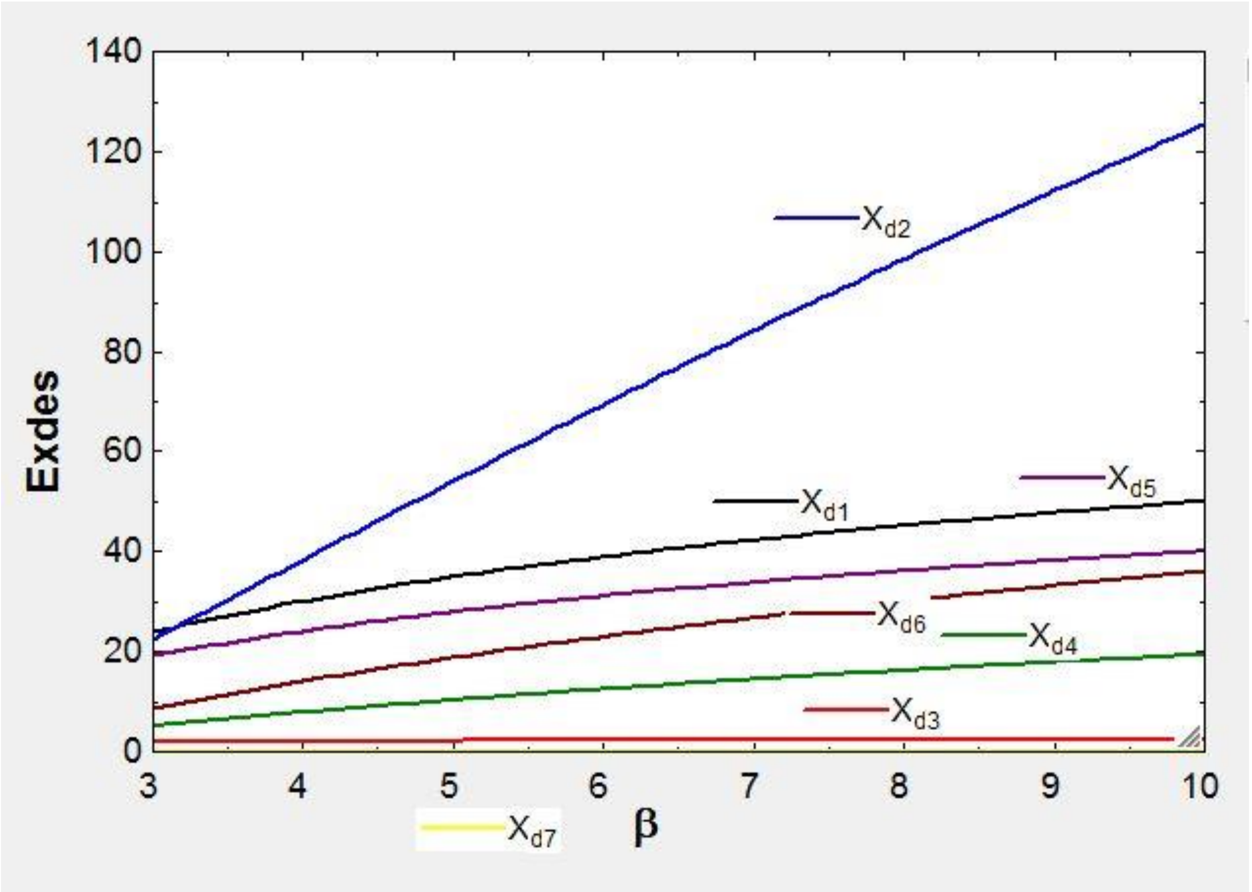

Fig 4.11. Exergy destruction of individual components vs pressure ratio

With increase in pressure ratio exergy destruction in each component goes on increasing.The increase is highest in cooler followed by compressor, turbine ,cabin , HE 2, HE 1 & mixing chamber respectively. Since in cooler heat is transferred to atmosphere so it is highly irreversible process & as the pressure ratio increases this irreversibility goes on increasing so highest exergy destruction is observed.

Although mixing chamber is highly irreversible but its exergy destruction value is coming low because we have neglected the pressure head loss in the connecting pipes.

4.5 VARIATION OF TOTAL EXERGY DESTRUCTION WITH DIFFERENT PARAMETERS

Variation of Total Exergy destruction with effectiveness of HE 1

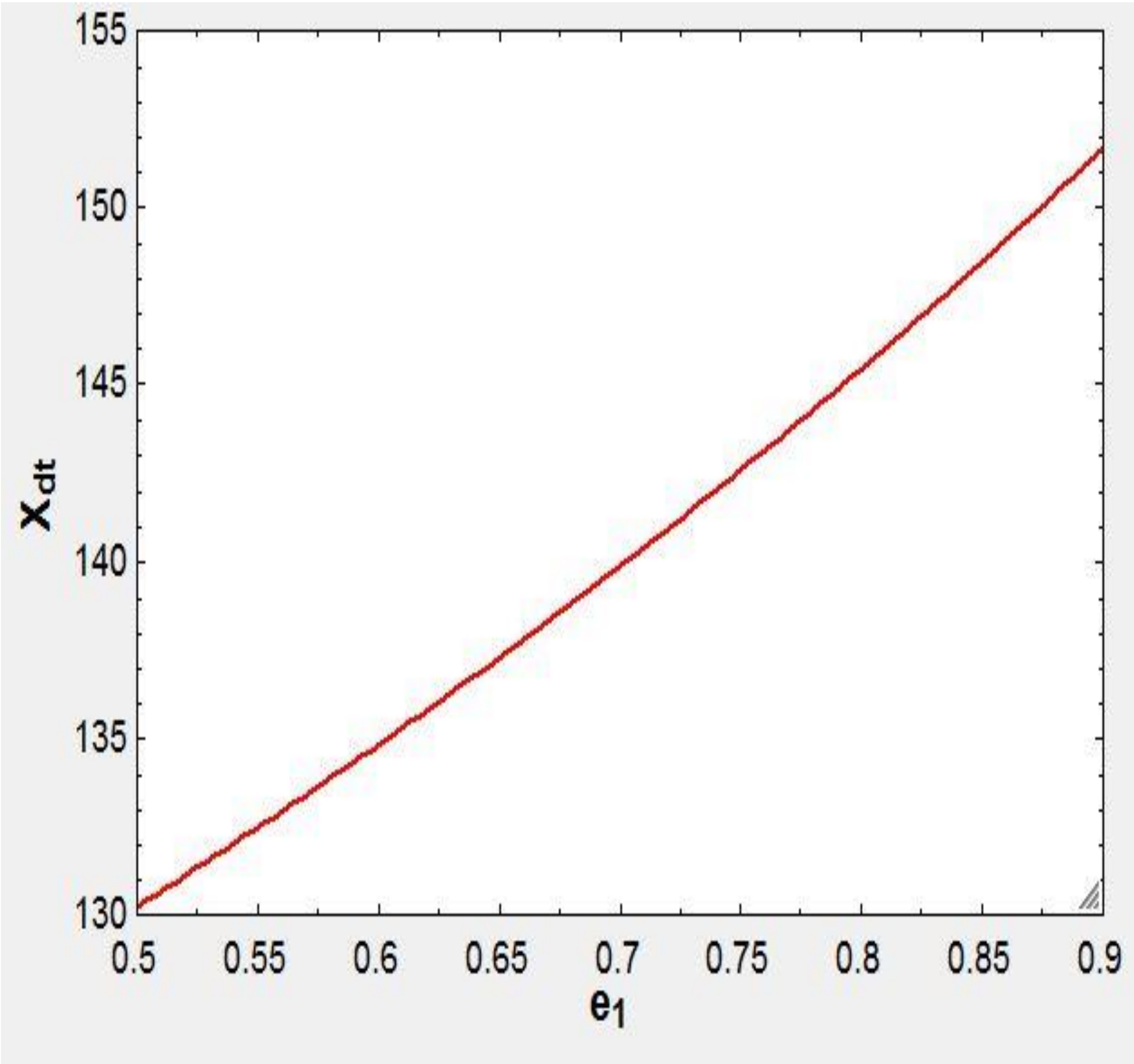

Fig 4.12 Total exergy destruction(X_{dt}) vs effectiveness of first HE

The graph shows us the values of change in total exergy destruction due to the change in effectiveness of first heat exchanger. From graph it is obvious that as the effectiveness of heat exchanger increases the total exergy destruction goes on increasing. It can be seen from graph that up to a certain value of effectiveness of heat exchanger (almost 0.62) there is linear increment in the value of total exergy destruction ,after that the rate of increase in exergy destruction value increases.When the effectiveness of heat exchanger is 0.5 the value of total exergy destruction is 130.4 (KJ/Kg) & when the effectiveness of heat exchanger is taken to be 0.9 total exergy destruction is 152.3 (KJ/Kg).

Variation of total Exergy destruction with effectiveness of HE2

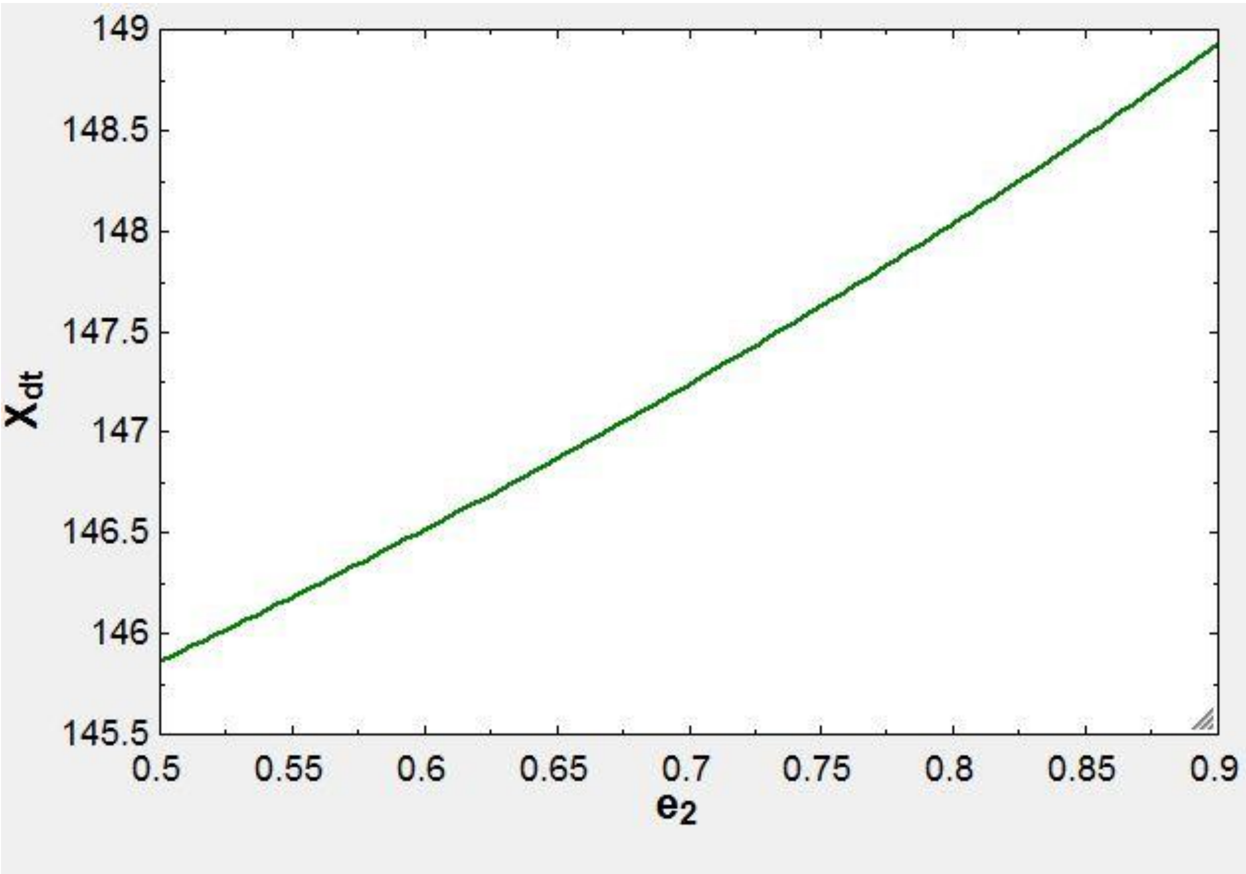

Fig 4.13 Total exergy destruction(X_{dt}) vs effectiveness of second HE

The graph shows us the values of change in total exergy destruction due to the change in effectiveness of second heat exchanger. From graph it is obvious that as the effectiveness of heat exchanger increases the total exergy destruction goes on increasing. When the effectiveness of heat exchanger is 0.5 the value of total exergy destruction is 145.8 (KJ/Kg) & when the effectiveness of heat exchanger is taken to be 0.9 total exergy destruction is 148.3 (KJ/Kg).

Thus due to the increase in effectiveness of second heat exchanger the total exergy destruction value increases by small amount only.

Variation of total Exergy destruction with mass fraction extracted for 2nd HE

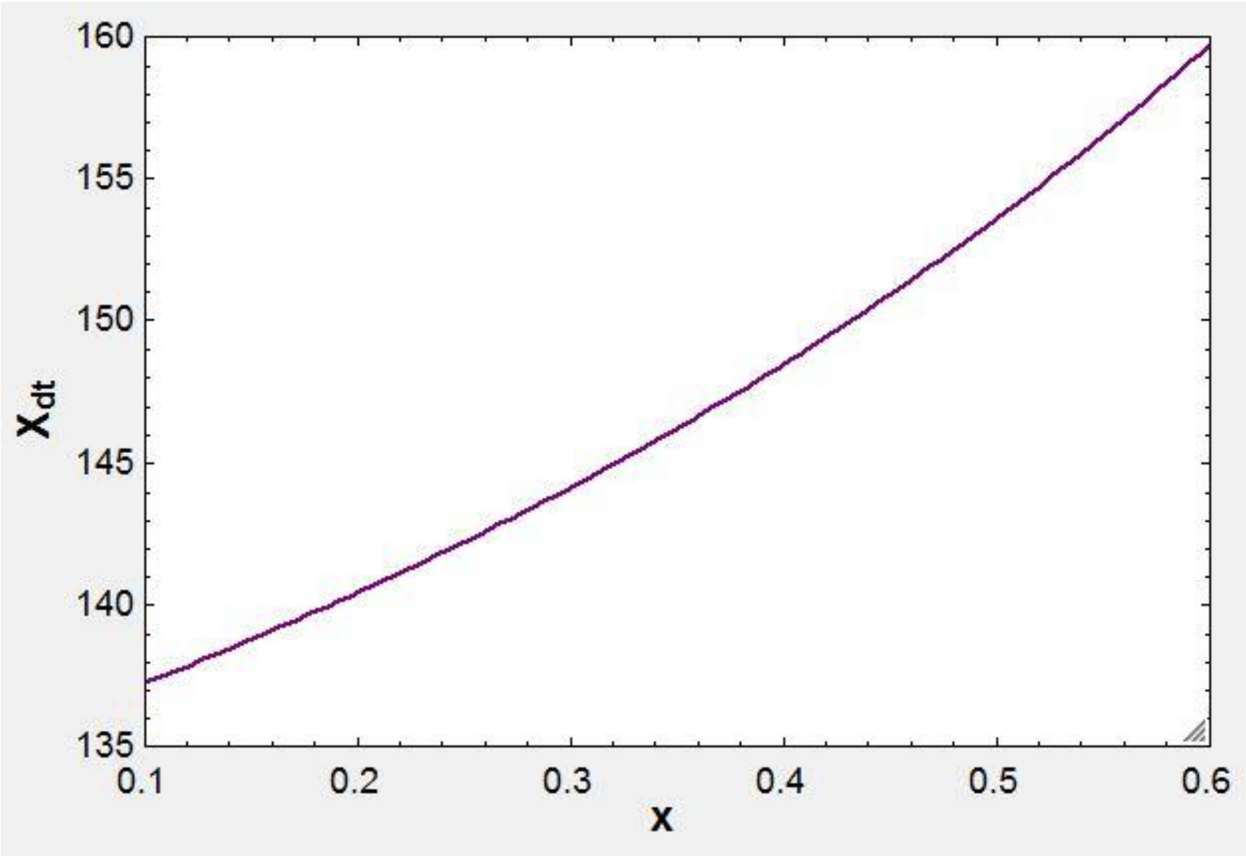

Fig 4.14 Total exergy destruction(X_{dt}) vs fraction of mass extracted

If the fraction of mass extracted for second state regeneration increases the total exergy destruction value increases. To maintain low temperature in the cabin the temperature at the exit of turbine should be very less therefore more mass has to be extracted & hence exergy destruction will be high.

If 10% of mass is extracted exergy detruction is 137.5 (KJ/Kg), the rate of exergy destruction goes on increasing with increase in percentage of mass extracted. At 60 % its value is 160(KJ/Kg).

Variation of total Exergy destruction with pressre ratio

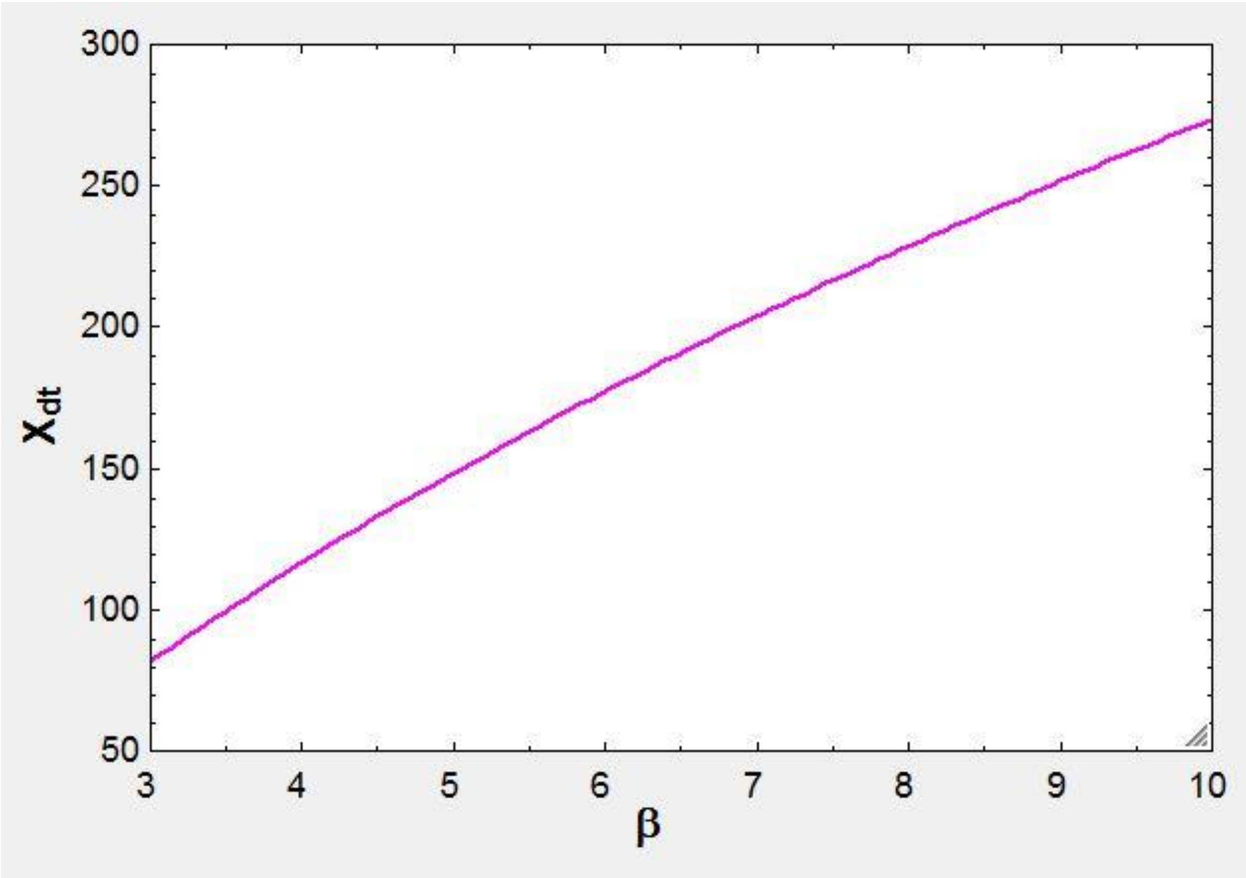

Fig 4.15 Total exergy destruction(X_{dt}) vs pressure ratio

The graph shows us the values of change in total exergy destruction due to the change in pressure ratio. From graph it is obvious that as the pressure ratio increases the total exergy destruction goes on increasing. It can be seen from graph that as the pressure ratio increases the rate of increase of exergy destruction goes on reducing.

When the pressure ratio is 3 the value of total exergy destruction is 81.3 (KJ/Kg) & when the effectiveness of heat exchanger is taken to be 0.9 total exergy destruction is 269.6 (KJ/Kg).

Thus it is observed that change in pressure ratio has huge impact on total exergy destruction as compared to other operating parameters.

CHAPTER 5

CONCLUSION

A double regenerated air refrigeration cycle is studied. Its thermodynamic analysis is done. Also the exergy destruction of each of the components in the system is calculated. It is observed that by using double regeneration the temperature at the exit of turbine (expander) reduces. As this temperature reduces the low temperature refrigeration is possible. It is observed that as the effectiveness of heat exchangers goes on increasing the temperature at the exit of turbine goes on reducing. Also as the mass extracted for second state regeneration increases the turbine exit temperature goes on reducing.

The COP of the system will decrease with increase in fraction of mass extracted & will increase with increase in effectiveness of either of the heat exchangers.

The exergy destruction in the compressor, Heat exchanger 2 & turbine remains constant on increasing the effectiveness of 1^{st} Heat exchanger. The exergy destruction increases in cooler & cabin while it reduces in Heat exchanger 1 & mixing chamber.

The exergy destruction in the compressor & turbine remains constant on increasing the effectiveness of 2nd Heat exchanger. The exergy destruction increases in cooler, heat exchanger 2& cabin while it reduces in Heat exchanger 1 & mixing chamber.

The exergy destruction in the compressor & turbine remains constant on increasing the fraction of mass extracted for second state regeneration. The exergy destruction increases in Heat exchanger 1, heat exchanger 2 &cabin while it reduces in cooler & mixing chamber.

On increasing the pressure ratio the exergy destruction in each component goes on increasing. The results can be useful in the design of heat exchangers as well as control of other thermodynamic parameters to get optimum COP, as well as to get low temperature refrigeration.

With addition of second heat exchanger although we are getting very low temperature at the exit of turbine ,but COP value is decreasing by a small amount .In future we can use high efficient heat exchangers to get high COP.

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