THERMODYNAMIC ANALYSIS OF THREE STAGES CASCADE VAPOUR COMPRESSION REFRIGERATION SYSTEM FOR BIOMEDICAL APPLICATIONS

R. S. Mishra ,
Mechanical Engineering Department, Delhi Technological University, Delhi-110042, India
professor_rsmishra@yahoo.co.in

Abstract-

Montreal Protocol was signed on 16-september1987 under the auspices of the united nations Environment Programme on substances that depleting ozone layer was phase out the consumption and production of ozone depleting substances within a specified time frame both for developed and developing countries and to provide aid to developing countries in terms of technology transfer, and incremental costs for implanting non ODS technologies Similarly Kyoto Protocol in Kyoto(1997) aims at the reduction and control of green house gases(GHG) emissions. This paper mainly focus on alternative refrigerants for replacing CFC-12 which contains global warming (GWP=1500) and ozone depleting potential(GWP=0055) containing chlorine sustenance . The main critical issue in the field of green technologies is to develop the relationship between ODP and GWP and suggest new and alternative refrigerants which do not damage ozone layer and not to increase global warming. The Numerical computation have been carried out using first and second law thermodynamic analysis (i.e. Energy and Exergy Analysis) of two and three stages cascade vapor refrigeration system of 10 ton capacity for seven eco-friendly refrigerants such as R-1234yf and R-1234ze in high temperature circuit, and R134a , R-404a, R-407C, R-502, propane(R-290), isobutene(R-600a), butane (R-600)) in lower temperature circuit in two stage and above refrigerants in intermediate circuit in three stage system. The performance parameters such as COP, EDR, exegetic efficiency, have been predicted.

Keywords*- Sustainable Technologies, Green (eco friendly) Technologies, Sustainable Development, Alternative Refrigerants, First law and Second law Analysis, Energy and Exergy analysis ,Vapour compression cascade refrigeration system, Irreversibility analysis .*

INTRODUCTION

Thermodynamic processes in refrigeration system releases large amount of heat to the environment. Further, heat transfer between the system and the surrounding environment takes place at a finite temperature difference, which is a major source of irreversibility for the cycle and also responsible for the system performance degradation. The losses in the cycle need to be evaluated considering respective individual thermodynamic processes by applying first and second laws. Energy analysis (First Law) is still the most commonly used method in the analysis of thermal systems which only concern with the conservation of energy, and gives no information on how, where, and how much the system performance is degraded. On the other hand, second law is used to describe the quality of energy of materials. The first law optimization results in maximizing the coefficient of performance (COP) thus providing maximum heat removal from minimum power input; while the second law optimization is used for maximizing the exergy efficiency and minimizing entropy generation within the system, hence providing maximum cooling for the smallest distraction of available energy (exergy).

LITERATURE REVIEW:

The exergy method, known as the second law analysis calculates the exergy loss caused by irreversibility which is an important thermodynamic property that measures the useful work that can be produced by a substance or the amount of work needed to complete a process. Thus exergy analysis is powerful tool in the design, optimization, and performance evaluation of energy systems. The principles and methodologies of exergy analysis are well established [1-5]. An exergy analysis is usually aimed to determine the maximum performance of the system and identify the sites of exergy destruction. The evaluation of system performance using second law of thermodynamics is based on exergy principles. Agrawal [1] investigated that exergy is always decreases due to thermodynamic irreversibility. Exergy analysis of a complex system can be performed by analyzing the components of the system separately. Identifying the main sites of exergy destruction shows the direction for potential improvements. An important objective of exergy analysis for system that consume work such as refrigeration, liquefaction of gases, and distillation of water is to find the minimum work required for a certain desired result [6]. There are several studies on the exergy analysis of refrigeration system [7, 8]. Bejan [8] showed that the exergetic efficiency decreases as the refrigeration temperature decreases. He offered two simple models to explain this trend. In his model, thermodynamic imperfections are explained largely by the heat transfer irreversibility. The behaviour of two stage compound-cycle with flash intercooling, using refrigerant R22 has been investigated by Nikolaidis and Probert [9] using exergy method. A computational model based on the exergy analysis is presented by Yumrutas et. al [10] for the investigation of the effects of the evaporating and condensing temperatures on the pressure losses, exergy losses, second law of efficiency, and the COP of a vapour compression cycle.

RESEARCH GAPS IDENTIFIED AND OBJECTIVE OF PRESENT INVESTIGATIONS

In the present research work, energy and exergy analysis is performed on three stages cascade vapour compression refrigeration system in which high temperature circuit ecofriendly refrigerants (R1234ze and R1234yf) and ethane in lower temperature circuits and other ecofriendly refrigerants have not been analyzed in detail . The objective of the present study mainly deals with the utility of ecofriendly refrigerants particularly in the intermediate temperature circuit and R1234ze and R1234yf refrigerants. The expressions for the exergy losses (lost works) for the individual processes of the cycle as well as the coefficient of performance (COP) and second law efficiency for the entire cycle has been obtained. Effect of variation of condensing and evaporating temperatures on exergy losses, second law efficiency and COP has been investigated. The concept of structure coefficient (coefficient of structure bond) is used to explain the relation between irreversibilities of system & its components in the three stages and two stage cascade systems.

It should be noted that the above discussion is based on the three stage cascade ideal vapor-compression refrigeration cycle which does not take into account the frictional pressure drop and heat loss in the system, slight internal irreversibility during the compression of the refrigerant vapour, or non-ideal gas behaviour (if any).To facilitate the thermodynamic analysis, simplified schematic of the present three stage vapour compression cascade system where Qcond is the heat rejection rate from the condenser, Qeva is the heat input rate(cooling load) to the evaporator and Wcomp is the work input required to the compressor. This paper mainly deals with thermal analysis of two and three stages cascade refrigeration systems for bio medical applications

FIRST LAW ANALYSIS FOR FINDING COP OF THREE STAGES CASCADE VAPOUR COMPRESSION REFRIGERATION SYSTEM:

For the thermodynamic analysis of the industrial ice plant working on vapour compression system (shown in Figure 3), the principles of mass conservation, first and second laws of thermodynamics are applied to each component of the system. Each component can be treated as control volume with inlet and outlet streams, heat transfer and work interactions. For the system, the mass conservation is governed by following equation:

$$m_i - m_o = 0 \quad (1)$$

where m is the mass flow rate and suffix i and o represents inlet and outlet of the component.

The first law of thermodynamics yields the energy balance of each component of the system as follows:

$$(mh)_i - (mh)_o + [Q_i - Q_o] + W = 0 \quad (2)$$

The overall energy balance of the system requires that the sum of the evaporator, condenser and compressor must be zero. The vapour compression system is assumed to be in steady state condition and further if the pump work for brine solution circulation and the environmental heat losses are neglected, the energy balance for the entire system can be written as

$$Q_{cond} = Q_{eva} + W \quad (3)$$

The energy balance equations of various components of a vapour compression system are given below: The cooling load in the high temperature evaporator 1 is given by

$$m_{R1} = Q_{eva1} / (h_1 - h_4) \quad (4)$$

Mass flow rate of circulated refrigerant in first circuit can be calculated as

$$m_{R1} (h_1 - h_4) = m_{R2} (h_6 - h_7) \quad (5)$$

For the compressor (by convention, the work done by compressor is presumed to be negative);

$$W_{comp1} = - m_{R1} (h_2 - h_1) / \eta_{m1} \eta_{e1} \quad (6)$$

where η_{m1} and η_{e1} are mechanical and electric motor efficiencies respectively.

Heat transfer rate in the condenser (zone B) is given by

$$Q_{cond} = m_{R1} (h_2 - h_1) \quad (7)$$

Coefficient of performance of refrigeration is

$$COP_1 = Q_{eva1} / W_{comp1} \quad (8)$$

For state point 4 dryness fraction and specific entropy can be represented as:

$$x_4 = (h_3 - h_{f1}) / (h_1 - h_{f1}) \quad (9)$$

where h_{f1} is the enthalpy of saturated liquid refrigerant at evaporator pressure.

Specific entropy is:

$$s_4 = s_{f1} + x_4 (s_1 - s_{f1}) \quad (10)$$

where s_{f1} is the entropy of saturated liquid refrigerant at evaporator pressure.

The energy balance equations of various components of a vapour compression system are given below:

Mass flow rate of circulated refrigerant in intermediate circuit2 can be calculated as

$$m_{R2} * (h_5 - h_8) = m_{R3} * (h_{10} - h_{11}) \quad (11)$$

For the compressor (by convention, the work done by compressor is presumed to be negative);

$$W_{comp2} = - m_{R2} (h_6 - h_5) / \eta_{m2} \eta_{e2} \quad (12)$$

where η_{m2} and η_{e2} are mechanical and electric motor efficiencies respectively.

Heat transfer rate in the cascade condenser2 in intermediate circuit is given by

$$Q_{cond2} = m_{R2} (h_6 - h_7) \quad (13)$$

The cooling load in the evaporator 2 is given by

$$Q_{eva2} = (h_5 - h_8) \quad (14)$$

Coefficient of performance of refrigeration is

$$COP_2 = Q_{eva2} / W_{comp2} \quad (15)$$

For state point 8 dryness fraction and specific entropy can be represented as:

$$X_8 = (h_7 - h_{f2}) / (h_5 - h_{f2} \quad (16)$$

where h_{f2} is the enthalpy of saturated liquid refrigerant at evaporator pressure.

Specific entropy is:

$$S_8 = s_{f2} + x_8 (s_5 - s_{f2}) \quad (17)$$

where s_{f2} is the entropy of saturated liquid refrigerant at evaporator pressure.

The energy balance equations of various components of a vapour compression system are given below:

Mass flow rate of circulated refrigerant can be calculated as

$$m_{R3} = Q_{eva3} / (h_9 - h_{13}) \quad (18)$$

For the compressor (by convention, the work done by compressor is presumed to be negative);

$$W_{comp3} = - m_{R3} (h_{10} - h_9) / \eta_{m3} \eta_{e3} \quad (19)$$

where η_{m3} and η_{e3} are mechanical and electric motor efficiencies respectively.

Heat transfer rate in the cascade condenser is given by

$$QCond3 = mR3\,(h10 - h11) \quad (20)$$

Coefficient of performance of refrigeration is

$$COP3 = Qeva3/Wcomp3 \quad (21)$$

For state point 12 dryness fraction and specific entropy can be represented as:

$$X12 = (h3 - hf3) / (h9 - hf3) \quad (22)$$

where hf3 is the enthalpy of saturated liquid refrigerant at evaporator pressure.

Specific entropy is:

$$S12 = sf3 + x12\,(s9 - sf3) \quad (23)$$

where sf3 is the entropy of saturated liquid refrigerant at evaporator pressure.

SECOND LAW ANALYSIS FOR FINDING EXERGETIC EFFICIENCY and EXERGY DISTRUCTION RATIO OF THREE STAGES CASCADE VAPOUR COMPRESSION REFRIGERATION SYSTEM:

Second law analysis is a relatively new concept, which has been used for understanding the irreversible nature of real thermal processes and defining the maximum available energy. The second law analysis is based on the concept of exergy, which can be defined as a measure of work potential or quality of different forms of available energy relative to the environmental conditions. In other words, exergy can be defined as the maximum theoretical work derivable by the interaction for an energy resource with the environment. Exergy analysis is applied to a system describes losses both in the components of the system and for the system as a whole. With the help of exergy analysis the magnitude of these losses or irreversibility's and their order of importance can be understood respectively. With the use of irreversibility, which is a measure of process imperfection, the optimum operating conditions can be easily determined. It is possible to say that exergy analysis throw an insight to indicate the possibilities of thermodynamic improvement for the process under consideration. The formulation for exergy analysis is described below:

The difference of the flow availability of a stream and that of the same stream at its restricted dead state is called flow exergy ( ) and by ignoring chemical exergy terms, flow exergy is given by

$$= (h - T0s) + V2/2 + gZ - (h0 - T0s0) \quad (24)$$

Ignoring the potential and kinematic energy terms, Eq. (10) becomes

$$= (h - T0s) - (h0 - T0s0) \quad (25)$$

The exergy balance equation is given by

$$Ew = EQ + (m\ )i - (m\ )o + T0Sgen \quad (26)$$

In equation (26) the term T0Sgen is defined as the irreversibility (I) and can be written as:

$$I = T0Sgen \quad (27)$$

The above exergy analysis formulation has been performed on each component of the vapour compression system and corresponding irreversibility of each component is calculated. Using this formulation described below:

By carrying out an exergy-rate balance for the compressor, the irreversibility rate in intermediate circuit :

$$IA1 = W + E1 - E2 \quad (28)$$

The exergy-rate balance for the evaporator1 :

$$ID1 = E4 - E1 - EQ1D \quad (29)$$

where, for the evaporator1

$$EQ1D = QC\,(T0 - Tc)/Tc \quad (30)$$

The exergy rate balance for the condenser1, is given by

$$IB1 = E2 - E3 \quad (31)$$

The exergy-rate balance in the throttling valve1, (zone C) is given by

$$IC1 = E3 - E4 \quad (32)$$

The total irreversiblitities of the system components in the primary circuit is expressed as

$$It1 = IA1 + IB1 + IC1 + ID1 \quad (33)$$

By carrying out an exergy-rate balance for the compressor, the irreversibility rate in intermediate circuit :

$$IA2 = W + E5 - E6 \quad (34)$$

The exergy-rate balance for the evaporator (zone D):

$$ID2 = E8 - E5 - EQ2D \quad (35)$$

where, for the evaporator

$$EQ2D = QC\,(T0 - Tc)/T \quad (36)$$

The exergy rate balance for the condenser, (zone B) is given by

$$IB2 = E6 - E7 \quad (37)$$

The exergy-rate balance in the throttling valve, (zone C) is given by

$$IC2 = E7 - E8 \quad (38)$$

The total irreversiblitities of the system components is expressed as

$$It2 = IA2 + IB2 + IC2 + ID2 \quad (39)$$

By carrying out an exergy-rate balance for the compressor, the irreversibility rate :

$$IA3 = Wc2 + E9 - E10 \quad (40)$$

The exergy-rate balance for the evaporator (zone D):

$$ID3 = E12 - E9 - EQ3D \quad (41)$$

where, for the evaporator

$$EQ3D = QC\,(T0 - Tc)/Tc \quad (42)$$

The exergy rate balance for the condenser, (zone B) is given by

$$IB3 = E10 - E11 \quad (43)$$

The exergy-rate balance in the throttling valve, (zone C) is given by

$$IC3 = E9 - E12 \quad (44)$$

Efficiency defect (k) of kth component of the system may be expressed as fractions of input which are lost through irreversibility

$$k = Ik/W \quad (45)$$

where Ik is the irreversibility rate of the kth component of the system under consideration.

The total irreversiblitities of three stages cascade system components is expressed as

$$It = IA1 + IB1 + IC1 + ID1 + IA2 + IB2 + IC2 + ID2 + IA3 + IB3 + IC3 + ID3 \quad (46)$$

The relative irreversibility of the kth component of plant is = Ik/It (47)

Structural coefficients are used in the study of the system structure, optimization of plant components and product pricing in multi-product plants. The change of local irreversibility rates and exergy fluxes in relation to the overall plant's irreversibility rate is effectively expressed by the coefficient of structural bonds (CSB) which is defined by

$$\sigma k, i = [\ It/\ xi]/[\ Ik/\ xi] \quad (48)$$

where xi is the ith parameter of the system which produces the changes in kth component.

The effect of a change in xi on the system would be to alter the rate of exergy input while leaving the output constant. This acceptance confirms to the usual practice of specifying a plant in terms of its output rather than its input. From the exergy balance of the system

$$EIN = EOUT + It \quad (49)$$

But EOUT = constant, thus

$$EIN = \ It \quad (50)$$

As seen from equation (50) changes in the irreversibility of the system are equivalent to changes in the exergy input. In general, the ratio of the rates of exergy output to exergy input is less than unity. This ratio denotes the degree of thermodynamic perfection of the process and is called the rational efficiency(R).

$$R = EOUT/EIN \quad (51)$$

Therefore cascade three stage rational efficiency R, plant = 1 – k (52)

RESULTS AND DISCUSSIONS:

Actually exergy is the maximum possible useful work that can be obtained from working substance with reference to environment. Exergy destruction occurred in the process can be obtained by difference between exergy entering and exergy leaving plus exergy associated with heat transfer from the source maintained at constant temperature and is equal to work obtained by the Carnot engine operating between temperature and surrounding temperature (Tamb) which is equal to the maximum reversible work that can be obtained from heat energy (Q) plus mechanical work transfer or from the system. After calculating exergy destruction in each component in the vapour compression refrigeration system one can find out the total exergy destruction in the system which is the sum of exergy destruction in the compressor, evaporator, throttling valve and condenser. First law performance can be measured in term of coefficient of performance (COP) but second law performance can be measured in terms of exergetic efficiency. Exergetic efficiency is the ratio of exergy of product to the exergy of fuel. Where as eta(II) = (Exergy of product /exergy of fuel) *100 i.e. Etasecond = 1- (ED+EL/EF) .Exergy of product represents the desired result produced by the system. In case of vapour compression refrigeration system the exergy of product is the heat abstracted from the evaporator and exergy of fuel is the actual input work of compressor. Exergy of fuel is the sum of exergy of product plus exergy destruction plus exergy loss. Therefore exergy of fuel in the vapour compression refrigeration system can be represented in terms of Ef= (Exergy of product + Exergy destruction + Exergy loss) Table-1, gives the various alternatives of ecofriendly refrigerants to be used in high temperature circuit and also ecofriendly refrigerants in the secondary circuit known as low temperature circuit in two stages cascade refrigeration system . Similarly in the three stages cascade refrigeration systems ethane has freezing point of -89°C has been used in the lower temperature circuit and comparisons have been made using other ecofriendly refrigerants in primary high temperature circuit as well as intermediate temperature circuit. The maximum COP was attained using R1234ze in high temperature circuit in two stage cascade refrigeration system

Table-1: Variation of THERMANCE PERFORMANCE PARAMETERS using eco-friendly refrigerants in the two stages and three stage cascade refrigeration system using R1234yf and R1234ze in high temperature circuits and ethane refrigerant in lower temperature circuit and following refrigerants in intermediate temperature circuit.(APPROACH1=10 , TcascadeEva1=263K, APPROACH2=10 , TcascadeEva2=233K, TCond=313K,TEva=188K,QLoad=QEVA, =35.167 kW, Cooling capacity= 10 Tons)

Name of Primary Refrigerant	High temperature circuits	High temperature circuits	High temperature circuits	High temperature circuits	High temperature circuits	High temperature circuits
Name of Secondary Refrigerant	COPR1234yf	COPR1234ze	EDRR1234yf	EDRR1234ze	II of R1234yf	II of R1234ze
R410a	0.9948	1.041	2.296	2.548	0.2401	0.2660
R134a	0.9947	1.041	2.295	2.547	0.2398	0.2663
R404a	0.9885	1.034	2.249	2.502	0.2269	0.2519
R717	0.9922	1.038	2.276	2.529	0.2345	0.2603
R744	0.9841	1.031	2.217	2.469	0.2184	0.2425
R290	0.9944	1.041	2.292	2.545	0.2391	0.2655
R600	0.9957	1.042	2.302	2.555	0.2421	0.2688
R123	0.9964	1.043	2.307	2.559	0.2435	0.2704
R125	0.9911	1.037	2.268	2.521	0.2321	0.2577
R600a	0.9948	1.041	2.296	2.548	0.240	0.2666
R407c	0.9604	1.004	2.036	2.289	0.1795	0.1994

while in three stage cascade it comes out to be 0.7048 using R134a in intermediate circuit.

Similarly from Table2 gives , exergy destruction ratio (EDR) increases with increasing evaporator and condenser temperatures where as second law efficiency (exegetic efficiency) decreases with increasing condenser and evaporator temperatures. Maximum EDR occurs in R-404a where as lowest occurs in R-407c. If we use hydrocarbons of zero ODP and zero GWP the lower ODP occurs using propane (R-290) and higher occurs by using isobutene (R-600a) as second alternative refrigerants, where as exegetic efficiency of vapour compression system using R-407c of GWP is 1530) is higher and lower by using R-410a refrigerant which has GWP of 1730. The next alternative refrigerant is R-134a refrigerant has zero ODP with 1300 GWP. The exegetic efficiency of vapour compression system using hydrocarbon as refrigerants (i.e. R-290, R-600, R-600a), which are slightly lower exegetic efficiency can be first alternative than R-407c. Although Table-2 gives the idea regarding the ratio of exergy destruction in condenser, evaporator, throttling valve and compressor (system components) to the total exergy destruction in the system using alternative refrigerants. The exergy destruction in the condenser is lower by using R-407c of 5.93% and higher 61.9% in case of using R-410a refrigerant in the vapour compression refrigeration system. However. one can use hydrocarbon (R-290, R-600 and R-600a), the propane is best eco friendly refrigerant due to zero ODP & zero GWP

However, next alternative is to use R-134a of zero ODP and 1300 GWP which requires larger

Table-2: Variation of THERMANCE PERFORMANCE PARAMETERS using eco-friendly refrigerants in the three stage cascade refrigeration system using R1234yf and R1234ze in high temperature circuits and ethane refrigerant in lower temperature circuit and following refrigerants in intermediate temperature circuit. APPROACH1=10 , TcascadeEva1=263K, APPROACH2=10 , TcascadeEva2=233K, TCond=313K, TEva=188K, QLoad=QEVA=35.167 kW, Cooling capacity= 10 Tons)

Primary Refrigerants	R1234ze	R1234yf	R134a	R410a	R290	R600a	R404a	R600	R407c
Secondary Refrigerants	ETA_ SECOND	ETA_ SECOND	ETA_ SECOND	ETA_ SECOND	ETA_ SECOND	ETA_ SECOND	ETA_ SECOND	ETA_ SECOND	ETA_ SECOND
R717	0.4111	0.3999	0.4105	0.4030	0.4083	0.4109	0.3915	0.4162	0.3831
R744	0.4021	0.3913	0.4015	0.3943	0.3993	0.4019	0.3833	0.4070	0.3757
R410a	0.41121	0.4008	0.4115	0.4040	0.4092	0.4119	0.3924	0.4172	0.3844
R404a	0.4081	0.3970	0.4075	0.4001	0.4033	0.4095	0.3887	0.4131	0.3809
R134a	0.4124	0.4011	0.4118	0.4043	0.4095	0.4122	0.3926	0.4175	0.3847
R290	0.4122	0.4009	0.4116	0.4041	0.4093	0.4120	0.3924	0.4173	0.3845
R600a	0.4123	0.4010	0.4117	0.4042	0.4093	0.4122	0.3926	0.4175	0.3846
R600	0.4136	0.4022	0.4130	0.4054	0.4107	0.4134	0.3937	0.4187	0.3857
R407c	0.3982	0.3877	0.3976	0.3906	0.3955	0.3981	0.3797	0.4030	0.3723
R123	0.4143	0.4030	0.4137	0.4061	0.4114	0.4142	0.3944	0.4195	0.3864
R125	0.4081	0.3971	0.4075	0.4002	0.4053	0.4080	0.3888	0.4131	0.3810
R152a	0.4135	0.4022	0.4129	0.4054	0.4107	0.4134	0.3937	0.4187	0.3857

size of compressor. it was observed that first law efficiency of vapour compression refrigeration system increases with increasing evaporator temperature and also decreases with increasing condenser temperature. The COP of R-407c is higher for condenser temperature of 313K with ambient temperature of 298K however R-407c has GWP of 1530 is the best option for using refrigerants in the vapour compression system where as the next option is using R-134a (of GWP is 1300). Utilization of hydrocarbon due to Kyoto protocol and Montreal protocol restricts the use of CFCs, HFCs etc due to emission of green house gas and temperature change due to global warming and increasing level of CO2. Ammonia (R-717) is also unsuitable for small refrigeration systems due to toxic and flammability nature. The use of hydrocarbon (in the vapour compression refrigeration system) such as propane (R-290) refrigerant which has zero ODP and zero GWP is most suitable in the vapour compression refrigeration systems because it has more COP than other hydro carbon such as butane (R-600) and isobutene (R-600a) which can also a next alternatives.

CONCLUSIONS AND RECOMMENDATIONS:

Montreal protocol protects the ozone layer, Hydrofluro carbons (HFCs) have been developed as substitutes for ozone depleting substances. R-407c (of GWP1530) refrigerants is first alternative for replacing R-22 in the 4.75KW vapour compression refrigeration system for domestic application due to its first and second law performances. R-134a (of GWP=1300) is the next alternative for replacing R-22 in the above system. Although they have zero depletion potential and they are producer of green house gases and are subjected to limitation and reduction commitments under United Nations Framework Convection on Climate change (UNFCC). These performance parameters have been computed for 35.167KW capacity of cascade system by varying condenser temperatures in the range from 303 K to 333K and first cascade evaporator temperatures in the range from 248 to 268K and second cascade evaporator temperatures in the range from 243 to 218K and using ethane in lower temperature in which evaporator temperatures in the range from 188 to 213K . It was observed that R1234ze in high temperature circuit and R 134a in lower temperature circuit is a best alternative than R1234yf in high temperature circuit and R 134a in lower temperature circuit .Although thermal performance of two and three stages cascade refrigeration systems are slightly higher using R290 and R600and R600a in lower/ intermediate circuits but have flammable problems. There for R-290 eco friendly refrigerant is recommended for replacing R-22 in the vapour refrigeration system for domestic applications and next alternative refrigerants (R-600 (butane) and R-600a (isobutene) and blends of R-290/R-600a) are recommended based on first law (energy analysis) performances (COPs) and second law (exegetic efficiency, exergy destruction Ratio (EDR) , etc) thermodynamic performances for domestic applications.

LIMITATIONS OF THE PRESENT WORK:

With the entry into force of Kyoto protocol on 16-2-2005 ,developed countries have already planning and implanting rational measures intended to contribute towards meeting green house gas reduction targets during the first commitment period of Kyoto protocol (2008-2012) . These countries have also started together with developing countries to size up projects that qualify under Kyoto clean development mechanism. All governments will work together over next few years to decide on future intergovernmental action on climate change. In this light, it is vital that there should be continuous work on the replacement option for ozone depleting substances in the way that serve the aim of the Montreal protocol and UNFCC The developing countries, the conversion of CFCs to alternate is still a major issue. The detailed exergic-economic analysis is also required to be adopt the green technology for reducing global warming and ozone depletion

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Nomenclature

A	Area (m2)
Approach	Difference between cascade condenser and cascade evaporator temperature
c	Constant
EQD	Exergy rate, (kW)
h	Specific enthalpy, (kJ/kg)
hf	Enthalpy of saturated liquid refrigerant, (kJ/kg)
I	Irreversibility rate, (kW)
mR	Mass flow rate of refrigerant, (kg/s)
P	Pressure, (bar)
Q0	Heat transfer rate, (kW/m2K)
r	Compression ratio
s	Specific entropy, (kJ/kg K)
sf	Entropy of saturated liquid, (kJ/kg K)
T	Temperature, (K)
W	Work input, (kW)
X	Dryness fraction

Greek Symbols (all dimensionless

ηm	Mechanical efficiency
ηe	Electric motor efficiency
σk,I	Coefficient of structural bonds
R	Rational efficiency
k	Efficiency defect

Abbreviations

COMP	Compressor
COND	Condenser
COP	Coefficient of performance
CSB	Coefficient of structural bond
EVAP	Evaporator
t	total