International Journal of Research in Engineering and Innovation (IJREI)
journal home page: http://www.ijrei.com
ISSN (Online): 2456-6934

Experimental & theoretical validation of thermodynamic performance of vapour compression refrigeration system using ecofriendly refrigerants

R.S. Mishra

Department of Mechanical & Production Engineering, DTU Delhi-110042

Abstract

In this paper, thermal model consisting of energy-exergy- analysis for variable speed compressor vapour refrigeration system is developed for using ecofriendly refrigerants was developed. The experiments were conducted on the system for several days for variable brine mass flow rates in the secondary circuit of evaporator and water is flowing in the secondary circuit of condenser. The developed model validates experimental results well. The developed model can predict the thermal performances of VCRS along with performance parameters including condenser and evaporator pressures (Bar), condenser and evaporator temperatures, variations in three LMTDs (because condenser consists of vapour and liquid parts), Evaporator and condenser loads, work done by the compressors, evaporator and condenser heat transfer coefficients (w/m^{2}K). This model predicts the similar behaviour for other ecofriendly refrigerants. It is found that the performance of HFO refrigerants is slightly less than HFC (R-134a) refrigerant. The performance of R1234ze is higher than R1234yf but lower than R134a.
Keywords: Thermodynamic Performances, Vapour compression refrigeration system, Thermal Model Validation

1. Introduction

The HFC-134a was identified as having a high global warming potential (GWP) of 1,430 and hence needs to be replaced by more environmentally friendly refrigerant. Original equipment manufacturers (OEMs) and suppliers to adopt an alternative refrigerant with a GWP of 150 or less by the year 2012. The European Union's F-gas Regulation No 842/2006 became law on 4 July 2006 and many of the requirements came into force on 4th July 2007. The F-Gas regulations will phase out the use of HFC-134a in automotive air conditioning systems for all new models beginning in 2011. In anticipation, extensive research is being carried out to develop new low global warming potential fluids to support the refrigeration and air conditioning industry. HFO-1234yf, which has a 100 year GWP of 4 as compared to that of CO2 could be used as a "near drop-in replacement" for HFC134a [1].

1.1 Vapour Compression Refrigeration Systems

Refrigeration is a technology which absorbs heat at low temperature and provides temperature below the surrounding by rejecting heat to the surrounding at higher temperature. Simple vapour compression refrigeration system which consists of four major components compressor, expansion valve, condenser and evaporator in which total cooling load is carried at one temperature by single evaporator but in many applications like large hotels, food storage and food processing plants, food items are stored in different compartment and at different temperatures. Therefore there is need of multi evaporator vapour compression refrigeration system. The systems under vapour compression technology consume huge amount of electricity, this problem can be solved by improving performance of system. Performance of systems based on vapour compression refrigeration technology can be improved .The performance of refrigerator is evaluated in term of COP which is the ratio of refrigeration effect to the net work input given to the system. The COP of vapour compression refrigeration system can be improved either by increasing refrigeration effect or by reducing work input given to the system. It is well known that throttling process in VCR is an irreversible expansion process. Expansion process is one of the main factors responsible for exergy loss in cycle performance because of entering the portion of the refrigerant flashing to vapour in evaporator which will not only reduce the cooling capacity but also increase the size of evaporator. This problem

Corresponding author: R. S. Mishra
Email Id: hod.mechanical.rsm@dtu.ac.in

can be eliminated by adopting multi-stage expansion where the flash vapours is removed after each stage of expansion as a consequence there will be increase in cooling capacity and reduce the size of the evaporator. In the vapour compression refrigeration systems, the major operating cost is the energy input to the system in the form of mechanical work (i.e. compressor work).Thus any method of increasing coefficient of performance is advantageous so long as it does not involve too heavy an increase in other operating expenses, as well as initial plant cost and consequent maintenance. Since the coefficient of performance of a vapour compression refrigeration system is the ratio of refrigerating effect to the compressor work, therefore the coefficient of performance can be increased either by increasing the refrigerating effect or by decreasing the compressor work.

Several methods are available for improving first law efficiency in the terms of coefficient of performance (COP) of vapour compression refrigeration systems as given below.

By introducing the flash chamber between the expansion valve and the evaporator. However the refrigerating effect and coefficient of performance and the power required are similar as that of a simple vapour saturation cycle when the flash chamber is not used. Thus the use of flash chamber has no effect on the thermodynamic cycle. The only effect resulting from the use of flash chamber is the reduction in the mass of refrigerant flowing through the evaporator and hence the reduction in the size of evaporator. By using the accumulator or pre cooler. When the accumulator is used in the vapour compression refrigeration system, the refrigerating effect, coefficient of performance, and power required to run the compressor is same as the simple saturation cycle. The accumulator or pre cooler is used only to protect the liquid refrigerant to flow into the compressor and thus dry compression is always used. By subcooling the liquid refrigerant by the vapour refrigerant. We know that subcooling the liquid refrigerant by the vapour refrigerant, the coefficient of performance of cycle is reduced.

By subcooling the liquid refrigerant leaving the condenser by liquid refrigerant from the expansion valve. In this process, the mass of refrigerant required in the heat exchanger is exactly equal as the mass of flash and that forms in the simple saturation cycle. Since the COP of this modified cycle and the power required to derive the compressor is same as that of simple saturation cycle. Therefore, this arrangement of subcooling the liquid refrigerant has no advantage because this method of subcooling of thermodynamically same as the simple saturation cycle. A liquid suction heat exchanger is used to sub-cool the liquid refrigerant from the condenser by exchanging heat with cold suction vapour. The subcooling would increase the refrigerating effect per kg of refrigerant. Also the suction vapour gets superheated, and it ensures that no liquid droplets should enter the compressor, thereby preventing any damage to the compressor valve. But at the same time, the compressor work would increase. However, there may be some improvement in COP of the cycle. Chopra et.al. [2]. The performance of vapour compression refrigeration system, be improved by a little consideration in compression in refrigerant a reduction of compressor work very closed to saturated vapour line. This can be achieved by compressing the refrigerant in a more stages with intermediate intercooling. It is economically only where the pressure ratio is considerable as would be the case when very low evaporator are desired or when high condenser temperature may be required. Therefore compound compression is generally economical in the large refrigeration plants. The refrigerating effect can be increased by maintaining the condition of refrigerant in more liquid state at the entrance to the evaporator. This can be achieved by expanding the refrigerant very close to the saturated liquid line. It was observed that by subcooling the refrigerant and by removing the flashed vapour as they are during multi stage expansion, the expansion can be brought closed to the liquid line. Chopra et.al. [3] Kumar et al. [4] did energy and exergy analysis of vapour compression refrigeration system by the use of exergy-enthalpy diagram. They did first law analysis or energy analysis for calculating the coefficient of performance and exergy analysis for evaluation of various losses occurred in different components of vapour compression cycle using R11 and R12 as refrigerants. Bolaji et al. [5] had done experimentally comparative analysis of R32, R152a and R134a refrigerants in vapour compression refrigerator. They reached to the conclusions that R32 shows lowest performance whereas R134a and R152a showing nearly same performance but best performance was obtained of system using R152a. Yumrutas et al. [6] carried out exergy analysis based investigation of effect of condensing and evaporating temperature on vapour compression refrigeration cycle in terms of pressure losses, COP, second law efficiency and exergy losses. Variation in temperature of condenser as well as have negligible effect on exergy losses of compressor and expansion valve, also first law efficiency and exergy efficiency increase but total exergy losses of system decrease with increase in evaporator and condenser temperature. Spatz and Motta [7], focused on replacement of R12 with R410a through experimental investigation of medium temperature vapour compression refrigeration cycles. In terms of thermodynamic analysis, comparison of heat transfer and pressure drop characteristics, R410a gives best performance among R12, R404a and R290a. Mastani Joybari et al. [8] performed experimental investigation on a domestic refrigerator originally manufactured to use of 145g of R134a.They concluded that exergetic defect occurred in compressor was highest as compare to other components and through their analysis it has been found that instead of 145g of R134a if 60g of R600a is used in the considered system gave same performance which ultimately result into economic advantages and reduce the risk of flammability of hydrocarbon refrigerants. Reddy et al. [9] performed numerical analysis of vapour compression refrigeration system using R134a, R143a, R152a, R404A, R410A, R502 and R507A and discussed the effect of evaporator temperature, degree of subcooling at condenser outlet, superheating of evaporator outlet, vapour liquid heat exchanger effectiveness and degree of condenser temperature on COP and exergetic efficiency. They reported that evaporator and condenser temperature have significant effect on both COP and exergetic efficiency and also found that R134a has the better performance while R407C has poor performance in all respect. Mishra [10] concluded that The first law efficiency in terms of coefficient of performance COP and second law efficiency in terms of exergetic efficiency of HFC-134a and HFO- 1234ze is almost same having a difference of 5.6%, which decreases with the increase in evaporator temperature, whereas it is 14.5-15% higher than HFO-1234yf. Hence HFO-1234yf can be a good drop-in' replacement of HFC-134a. At the higher value of evaporator temperature and HFO-1234ze can be a good replacement after certain modification. From the irreversibility or exergy destruction viewpoint, worst component is condenser followed by compressor, throttle valve, evaporator and liquid vapour heat exchanger, the most efficient component. Total efficiency defect is more for HFO-1234yf followed by HFO-1234ze and HFC-134a, but the difference is small. HFC-134a gives higher COP and exergetic efficiency than HFO-1234yf but lesser value than HFO- 1234ze. Hence, it can be concluded that even though the values of performance parameters for HFO-1234yf are smaller than that of HFC-134a, but the difference is small, so it can a good alternative to HFC-134a because of its environmental friendly properties. HFO-1234ze can replace the conventional HFC-134a after having slight modification in the design as the performance parameters are almost similar. Esbri, et al. [11] experimentally analysed HFO-1234yf as a drop-in replacement for HFC-134a in a vapour compression system and summarized as, the cooling capacity of HFO-1234yf is about 9% lower than that of HFC-134a, which diminishes with the use of internal heat exchanger. Volumetric efficiency was about 5% less than that obtained with HFC-134a. From the irreversibility or exergy destruction viewpoint. Reaser et al. [12] investigated and compared the thermophysical properties of HFO-1234yf to those of HFC-134a and R410a to determine the drop-in replacement potential of HFO-1234yf and concluded that properties were similar to that of HFC-134a and not much similar to that of R410a. Mishra [13] carried out detailed energy-exergy analysis of vapour compression cascade refrigeration systems and found that the worst component is condenser followed by compressor, throttle valve, evaporator and liquid vapour heat exchanger, the most efficient component. Total efficiency defect is more for HFO-1234yf followed by HFO-1234ze and HFC-134a, but the difference is small. Increase in ambient state temperature has a increasing (positive) effect on second law efficiency in terms of exergetic efficiency and exergy destruction ratio which was computed based on exergy of fuel or based on exergy of product (EDR). When exergy destruction ratio (EDR) reduced, then exergetic efficiency increases. Therefore HFO-1234yf gives lesser values of exergetic efficiency whereas HFO-1234ze gives approximately similar values.4. HFC-134a gives higher COP and exergetic efficiency than HFO-1234yf but lesser value than HFO- 1234ze. However reverse trend is seen when effectiveness of heat exchanger is increased from 0 to 1. Hence, it can be concluded that even though the values of performance parameters for HFO-1234yf are smaller than that of HFC-134a, but the difference is small, so it can a good alternative to HFC-134a because of its environmental friendly properties. HFO-1234ze can replace the conventional HFC-134a after having slight modification in the design as the performance parameters are almost similar. Mishra [14] evaluated the performance of HFO-1234yf theoretically and showed that it had 2-9% less capacity and 2-7% less COP than HFC-134a. Also HFO-1234yf had similar lubricant miscibility and polymer compatibility as that of HFC-134a.

Literature survey emphasizes that HFO-1234yf and HFO-1234ze can be a promising alternative to HFC134a.Secondly, it has been observed that in most of the studies referred above, the analysis of the systems is based on first law of thermodynamics i.e. estimating coefficient of performance. In this study a more comprehensive exergy approach is followed, based on both first and second laws of thermodynamics. It is a powerful tool in the design and performance evaluation of the systems and allows an explicit presentation of thermodynamic processes by quantifying the effect of irreversibility occurring during the processes. Exergy balance applied to processes tells us how much of the exergy input to the system has been consumed (irreversibly lost) by the system. This analysis takes into account all the losses appearing in the refrigeration system, for calculating exergetic efficiency. The various parameters calculated are COP, exergetic efficiency, exergy destruction and efficiency defects. Effects of degree of sub-cooling, liquid vapour heat exchanger effectiveness and dead state temperature are also computed.

2. Experimentation

In this paper, the experimental procedure for operating the test rig has been developed as shown in Fig.1. For the development of experimental test rig the various components has been used for vapor refrigeration system, the configuration of the components and system is as per computational design input. Following components and instruments has been used for development of test rig.

Figure 1: Vapour compression refrigeration system model

2.1 Compressor

A hermetically sealed model no. KCE419HAG B130, serial – DBKA 36610 reciprocating compressor is used as shown in table 1. The low pressure and temperature vapor refrigerant from evaporator is drawn into the compressor through inlet or suction valve, where it is compressed to a high pressure and temperature. This high pressure and temperature vapor refrigerant is discharged into the condenser through the delivery or discharge valve as shown in Fig 2

Figure 2: Hermetically sealed reciprocating compressor

Table.1: Compressor specifications

S.No.	Items	Specification
1	Power Input	245 watts
2	Input Current	1.6 amps
3	Displacement	5.79
4	Weight	10.2 kg
5	Oil Charge	510
6	Wiring	RSIR

2.2 Pressure Gauge

Pressure gauges are mounted at salient point in the setup to measure pressure. In this setup following two types of pressure gauges are used As Shown in Fig .3.

S.N	Pressure gauge	Range
1	Suction pressure gauge.	Range -30 to 24.6 kg/cm^2
2	Discharge pressure gauge.	Range 0 to 35 kg/cm^2

2.3 Temperature meter

In this setup there are four temperature meters is used to measure brine, water inlet & outlet temperature and is to be used to find out the result as shown in Fig.4.

Figure 3: Pressure gauge

Figure 4: Temperature meter

S.No.	Items	Specification
1	Range	-50°c to 80°c
2	Accuracy	+-1
3	Resolution	0.1
4	Using environment	20 to 85% RH
5	Power supply	BatteryLR$$,1.5V

2.4 Watt meter

A wattmeter of 0-500 watt Range is used in the experimental setup to determine the load of setup which is used to calculate the performance of experimental setup as shown in Fig. 5

Figure: 5 Watt meter

2.5 Hand Shut Off Valve

Hand shut off valve are provided to stop or allow the refrigerant flows as desired to run the setup in vapor compression cycle as shown in Fig. 6.

Figure 6: Hand Shut Off Valve

Figure 7: Flow meter/Rota meter

2.6 Flow meter / Rota meter

It is fitted in liquid line in between dryer & expansion valve (capillary tube) to determine the mass flow rate of refrigerant during experiment for calculations as shown in Fig.7 (Range 0.1 to 1 LPM).

Figure 8: Drier used in VCRS

Figure 9: Capillary tube used in VCRS

2.7 Drier

It is fitted between the condenser outlet and evaporator inlet. It contain silica gel to observe the moisture from the refrigerant acts as a drier as shown in Fig.8.

2.8 Capillary tube

It is placed between condenser outlet and evaporator inlet. Capillary tube is used to reduce the pressure and boiling point of the refrigerant as shown in Fig. 9. It acts as a expansion device in the experimental setup.

2.9 Condenser and evaporator concentric tubes

In the Fig.10 shown the above pipe namely condenser having two concentric tube length 1.2 m and size of inner tube diameter is 3/8", outer tube diameter is 5/8'. The below pipe namely evaporator also having two concentric tube length 0.8 m and size of inner tube 3/8", outer tube 5/8'. Both the pipe are insulated with three layer of insulation first inner insulation with aluminum foil in the inner side then PVC black tape then KIMI foam.

Figure 10: Condenser and evaporator

2.10 Relay

Relay is used to start the compressor as shown in Fig.11

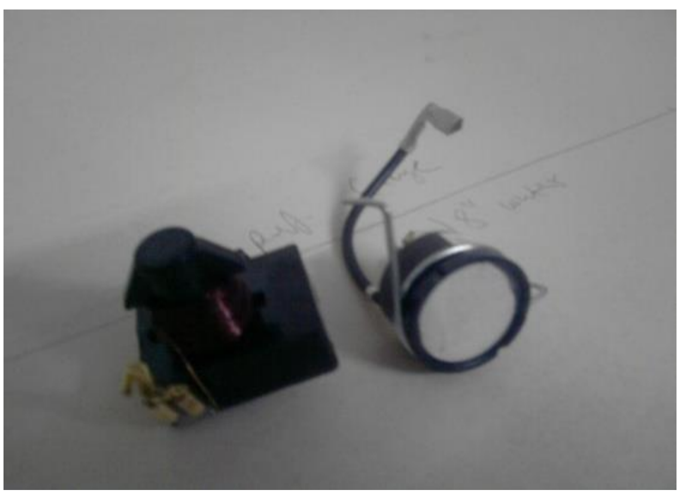

Figure 11: Relay in VCRS

2.11 Development of Experimental test Rig

Following procedure for the development of experimental test rig.

2.12 Procedural Steps of Experimental Setup

- *Selection of Refrigerant*
- *Selection of equipment.*
- *Purchase of components & testing equipments*
- *Making of testing table*

2.13 Primary refrigeration circuit (i.e. mechanical refrigeration cycle)

- Compressor model no.KCE419HAG (i.e. R-134a).
- The condenser & evaporator concentric tube type is fabricated as per the required size.
- Flow of refrigerant in both condenser & evaporator is counter flow.
- In the evaporator Refrigerant flows inside the inner tube and water surrounding the inner tube and the condenser water flows inside the inner tube and refrigerant surrounding the inner tube.
- To prevent the heat loss the condenser and evaporator tubes are insulated.
- The pressure gauge and thermometer are mounted on the compressor discharge line which is connected to the condenser.
- Liquid indicator, drier, pressure gauge, thermometer, Rota meter are mounted on the liquid line after the condenser outlet.
- Condenser liquid line is connected from the condenser liquid line through capillary and pressure gauge and thermometer is mounted before the evaporator line after the capillary tube.
- Evaporator is connected to the suction of the compressor and pressure gauge, thermometer are mounted on the suction of compressor line.

2.14 Secondary refrigeration circuit (i.e. water circulation cycle)

- In the condenser water flows inside the inner tube and refrigerant surrounding the inner tube.
- Fresh water is taken for circulation in the inside tubes.

2.15 Electrical circuit

- The electrical component such as compressor, motor, m.c.b., volt meter, ammeter & wattmeter are connected in the experimental setup.

2.16 Leak testing

- The leakage is to check with dry Nitrogen, if leakage is found it is to be rectified after rectifying again leakage is to be checked and it is to be hold for 4 hours in the same and N2 should not be charge higher than its operating pressure.

2.17 Evacuating the refrigeration system

- After leak testing evacuate the refrigeration system with two stage rotary vacuum pump for minimum 8hr's up to 150 micron.
- After the completing the leakage the refrigerant system evacuated with 2 stage rotatory compressor vacuum pump for min 8 hrs.

2.18 Charge of refrigerant in the system

- Charge the refrigerant by weight & measure all the parameters like pressure, temperature & mass flow rate.

2.19 Testing of the refrigeration system

- Once the power supply is given the voltage, wattage & frequency are noted down.
- The pressure & temperature readings are noted after starting the compressor.
- The water supply to the condenser and the evaporator tubes are given and the flow rate is set as the given requirement.
- The refrigerant is charged as per the requirement of the experimental setup and all the parameter like pressure temperature and mass flow rate measure it.
- After reaching the steady state condition the reading are noted down and C.O.P. of the system is determine.
- The experimental C.O.P. is compared with the C.O.P. is determined from program.

3. Thermodynamic Model Formulation

In this paper to evaluate the performance parameter of VCRS using eco-friendly refrigerants following model and formulation for each component of VCRS is used as shown in following Fig.12

In this model we have taken only five variables with geometric parameter of the complete system (assumed) as input variable for the VCRS. These variables, together with the thermo physical properties of the refrigerants, and geometric characteristics of the VCRS (vapour compression refrigeration system) are used to obtain the evaporation and condensing pressure and brine, water outlet temp. Power consumption, energy efficiency and exergy efficiency of VCRS. Thus we will be able to evaluate the performance of VCRS by changing the operating parameter on the system performance so that we can optimize the performance of the VCRS.

The Schematic structure of the model proposed for this investigation is presented in Fig. 12 where it can be seen that the model input variable are brine inlet temperature in the evaporator, water inlet temperature in the condenser, compressor speed, types of refrigerant, mass flow rate of brine, water and geometric parameter of the heat exchanger (Evaporator and Condenser) and used to evaluate the performance of VCRS without nanofluid. The proposed model is basically a VCRS based chiller machine in which two concentric tube (copper) heat exchanger is used for evaporating and condensing operation

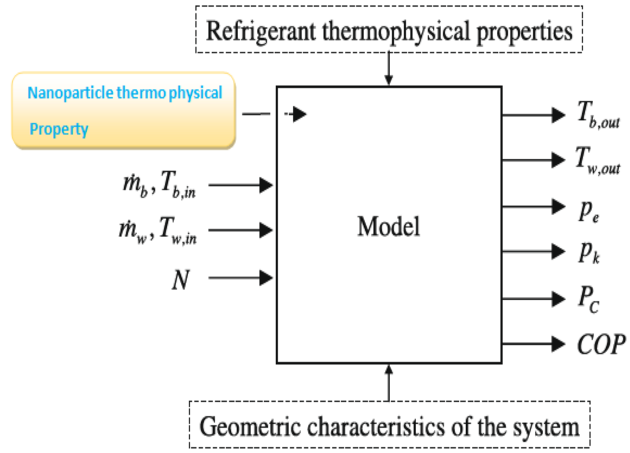

Figure 12: Vapour compression refrigeration system Model

In this model refrigerant is supposed to flow in inner side of the evaporator tube and brine in annuli side as shown in fig 13 and water flow in inner side of the condenser tube and refrigerant in the annuli side as shown in fig 13. The input parameter taken initially to compute thermal performance of VCRS using ecofriendly refrigerants is given below.

Table 2: Input variable used in VCRS.

Input variable	
Type of Refrigerant	R$='R134a',R407c,R404A
RPM of compressor	N=2900 {rpm}
Brine inlet Temperature	$T_{\_brine\_in}$ =25+273.15 {K}
Water inlet Temperature	$T_{\_water\_in}$=25+273.15 {K}
Mass flow rate of brine	$m_{\_ brine}$ =0.006 {kg/s}
Mass flow rate of water	$m_{\_ water}$ =0.006 {kg/s}
Air Temperature	T_air=30+273.15 {K}
Inlet pressure of brine	P_{brine} _in=2{bar}
Inlet pressure of water	P_{water} _in=2{bar}
Air Pressure	P_air=1.01325 {bar}
Acceleration due to gravity	g=9.81{m/s2}
Ambient Temperature	$T_{\_ambient}$=298{K}
Inside dia. of brine tube in evaporator	D_bie=0.013875 {m}
Inside dia. of refrigerant tube in evaporator	D_rie=0.007525 {m}
Outside dia. of refrigerant tube in evaporator	D_roe=0.009525 {m}

Using these inputs and the main characteristics of the compressor and heat exchangers, the model compute the operating pressures (without considering pressure drops), secondary fluids output variables and the energy performance. The property of nanorefrigerant/refrigerant and the thermo-physical properties of secondary fluids are evaluated by using Engineering equation solver *(EES).*

Table 3: Geometric Parameter used in VCRS.

Geometric Parameter	
Inside dia. of brine tube in evaporator	D_bie=0.013875 {m}
Inside dia. of refrigerant tube in evaporator	D_rie=0.007525 {m}
Outside dia. of refrigerant tube in evaporator	D_roe=0.009525 {m}
Length of tube in evaporator	L_e=0.76 {m}
Surface area of evaporator	S_e=3.14*(D_roe)*L_e {m^2}
Geometric compressor volume	V_g =5.79E-06 {m3}
Inside dia. of refrigerant tube in condenser 3-4	D_rik=0.013875 {m}
Inside radius of refrigerant tube in condenser 3-4	r_rik=D_rik/2 {m}
Inside dia. of water tube in condenser 3-4	D_wik=0.007525 {m}
outside dia. of water tube in condenser 3-4	{D_wok=0.009525 {m}}
outside radius of water tube in condenser 3-4	r_wok=D_wok/2 {m}
Length of tube in condenser 3-4	L_cond=1.05 {m}
outside dia. of water tube in condenser 3-4	D_wok=0.00508 {m}
Surface area of condenser 3-4	S_k=3.14*(D_wok)*L_k {m^2}
Inside dia. of refrigerant tube in condenser 2-3	D_ri23=0.007525 {m}
Inside radius of refrigerant tube in condenser 2-3	r_ri23=D_ri23/2 {m}
Outside dia. of refrigerant tube in condenser 2-3	D_ro23=0.009525 {m}
Outside radius of refrigerant tube in condenser 2-3	r_ro23=D_ro23/2 {m}
Length of tube in condenser 2-3	L_23=0.6 {m}
Surface area of condenser 2-3	S_23=3.14*D_ro23*L_23 {m^2}
Inside dia. of Capillary Tube	D_cap=0.0006 {m}
Fouling in the tubes	R_TFO=0.000086 {m^2 K/W}

3.1 Vapor compression system modeling

The model contain of a set of below equations based on physical laws showing the main parts of the system, as shown Properly (Schematically) in Fig. 13.

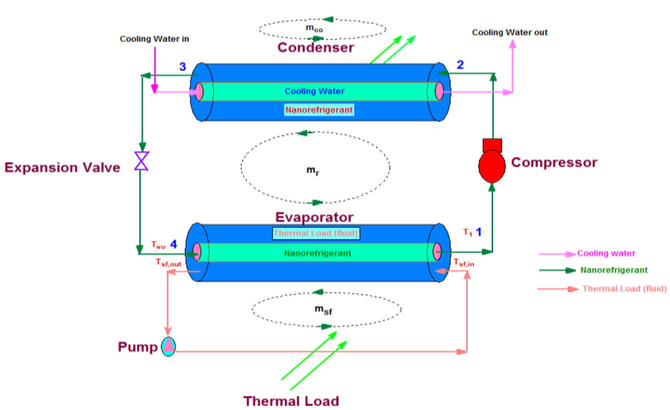

Figure 13: Schematic diagram of vapour compression refrigeration system model

The refrigerant states are numbered in Fig. 14. To compute refrigerant mass flow rate (m_r) Eq. (1) is used, where the compressor volumetric efficiency (η_v) is shown as a function of operating pressure and compressor speed (N) as shown in Eq. (2). For simplicity, (ρ_1) is the nano refrigerant/refrigerant density taken is the one corresponding to the saturated condition.

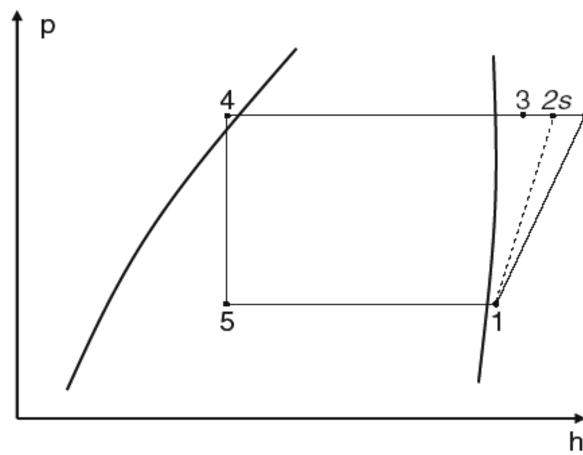

Figure 14: T-h diagram of Vapor compression cycle

Vapor at the evaporating pressure and V_g is the geometric compressor volume.

$$mr= \eta_v \cdot \rho_1 \cdot v_g \cdot N \qquad (1)$$

$$\eta_v = 1 + C - C\left[\frac{P_k}{P_e}\right]^{\left[\frac{1}{n_1}\right]} \qquad (2)$$

$$C = \frac{V_{suction}}{V_{discharge}}$$

Where,
P_k &P_e is the condenser pressure and evaporator pressure.
$V_{Suction}$ & $D_{ischarge}$ is the suction volume and discharge volume of compressor.
n_1 is the index of expansion.

3.1.1 Evaporator Formulation

The evaporator is computed with the help of energy balance equation of heat exchanger i.e. Eq. (3)

$$m_r \cdot (h_1 - h_5) = m_b \cdot Cp_b \cdot (Tb_{in} - Tb_{out}) \qquad (3)$$

Where, $h_1$is enthalpy at state one and h_5 is at state 5 of refrigerant, see Fig. 15. Cp_b is the Specific heat of brine. $Tb_{in}$and $Tb_{out}$is the inlet and outlet Temperature of the brine.

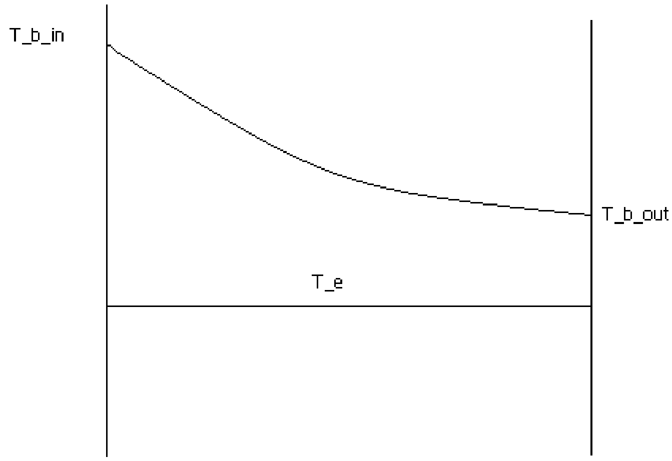

Figure 15: Temperature variation in evaporator

The logarithmic mean temperature difference and global heat transfer coefficient, with LMTD correction factor = 1

$$m_r \cdot (h_1 - h_5) = U_e \cdot S_e \cdot \left[\frac{Tb_{in} - T_e - (Tb_{out} - T_e)}{\ln\left(\frac{Tb_{in} - T_e}{Tb_{out} - T_e}\right)} \right] \quad (4)$$

Here, U_e is the global evaporator heat transfer coefficient and S_e is the evaporator surface area. T_e is the evaporator Temperature. The evaporator global heat transfer coefficient *(Ue)* computed using Eq. *(5)* as below.

$$U_e = \frac{1}{r_o} \cdot \left[\frac{1}{\alpha_b \cdot r_o} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{K_M} + \frac{1}{\alpha_{lv} \cdot r_i} + R_{TFO} \right]^{-1} \quad (5)$$

Where,

K_M is metal (copper) thermal conductivity, r_o & r_i is the inside and outside radius of the copper tube, α_{lv} is the heat transfer coefficient for the nanorefrigerent/refrigerant.
$R_{T\,Fo}$is fouling resistance in heat exchanger tube and its value is taken = 0.000086 m^2.
The coefficient of heat transfer for the brine (α_b) is computed using Zukauskas' correlation [27].

$$\alpha_b = \frac{K_b}{D_o} \cdot C_1 \cdot Re_b^{\,m_1} \cdot Pr_b^{\,0.36} \cdot \left[\frac{\mu_b}{\mu_{bM}} \right]^{0.25} \quad (6)$$

Where,

K_b is the thermal conductivity of brine.
D_O is the outer diameter of copper tube.
Re_b is the Reynold's No. of brine.
Pr_b is the Prandle No. of brine.
C_1 and m_1 is a correlation coefficient and value taken 0.683 and 0.466 respectively.
μ_b viscosity of brine.
μ_{bM} viscosity of brine at metal temperature, and the refrigerant heat transfer coefficient (α_{lv}) is modeled With the Chen's correlation [28].

$$\alpha_{lv} = sf \cdot \alpha_{nb} + F \cdot \alpha_{conv} \quad (7)$$

Where

Where,
α_{nb} is the nucleated boiling coefficient taken by Stephan and Abdelsalam [29].
α_{conv} is the convective heat transfer coefficient , obtained using the Dittus–Boelter correlation[30].
sf is the suppression factor.
F is Enhancement factor.
The values of these parameters are obtained using the following equations.

$$\alpha_{nb} = 207 \cdot \frac{K_r}{BD} \cdot \left[\frac{q'' \cdot BD}{K_r \cdot T_e} \right]^{0.674} \cdot \left[\frac{\rho_G}{\rho_{l,r}} \right]^{0.581} \cdot Pr_r^{\,0.533} \quad (8)$$

$$BD = 0.51 \cdot \left[\frac{2 \cdot \sigma}{g_g \cdot (\rho_{l,r} - \rho_G)} \right]^{0.5}$$

Where,
BD is the bubble departure diameter computed from above equation, σ is thesurface tension of refrigerant.
Kr is the thermal conductivity of refrigerant in liquid phase in evaporator.
q" is the heat flux (W/m^2) supplied in the evaporator, T_eva is the evaporator temperature
Pr is the prandtl number of the refrigerant in the evaporator
ρ_g&ρ_{Lr} isthe density of refrigerant in gas and liquid phase in evaporator.

$$\alpha_{conv} = 0.023 \cdot Re^{\,0.8} \cdot Pr^{\,0.4} \cdot \frac{K_L}{D_i} \quad (9)$$

$$sf = 0.927 \cdot \left[\left(\frac{1 - x}{x} \right)^{0.8} \cdot \left(\frac{\rho_G}{\rho_{l,r}} \right)^{0.5} \right]^{0.319}$$

$$F = 53.64 \cdot \left[\frac{q''}{G_r \cdot h_{fg}} \right]^{0.314} \cdot X_{tt}^{\,-0.839}$$

$G_r = \rho_r - V_r$

Where,
D_i is the tube inner diameter.
X_{tt} is the Martinelli's Parameter.
X or X_v is the quality of refrigerant.
h_{fg} is the latent heat of evaporation
G_r is mass flux of refrigerant (Kg/m^2-s)
V_r is the velocity of refrigerant

$$X_{tt} = \left[\frac{1 - X_v}{X_v}\right]^{0.9} \cdot \left[\frac{\rho_v}{\rho_L}\right]^{0.5} \cdot \left[\frac{\mu_L}{\mu_v}\right]^{0.1}$$

3.2 Compressor Formulation

The compressor performance is computed from the isentropic efficiency *(η_{is})* and the pressures (Low and High) in the compressor. So that , with the help of refrigerant state at the evaporator outlet and the refrigerant state at the compressor discharge is obtained using Eq.*(10)* from the isentropic compression process work and the isentropic efficiency of the compression,

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_{is}} \tag{10}$$

η_{is} = 0.156323 + 0.0000912. N + 0.004302. P_k + 0.09151. P_e (11)

Where, η_{is} is the compression isentropic efficiency and it can be obtained from gathered empirical data, as a function of operating pressures, yielding Eq. *(11):* To find out compressor power consumption, the model if supposed to use of a global electromechanical efficiency eq. (12) as a function of *N*,

$$\eta_g = 0.00002805 \cdot N^2 + 0.02593961 \cdot N + 6.4965 \tag{12}$$

As in Eq. (2) the efficiencies obtained by Eq. (11) and (12) Show a significant relationship between variables with a satisfactory level of 99%. The correlations used to obtain isentropic and electromechanical efficiency of the compressor in the system, and similar relations must be obtained from experimental results for second compressor. The heat transfer from the refrigerant in the compressor discharge line to the inlet of the condenser has been modeled, due considering the length of that line in the chiller facility, using eq. (13:

$$\dot{m}_r \cdot (h_2 - h_3) = U_{23} \cdot S_{23} \cdot \left[\frac{T_2 - T_{air} - (T_3 - T_{air})}{\ln\left(\frac{T_2 - T_{air}}{T_3 - T_{air}}\right)}\right] \tag{13}$$

U_{23} is the universal heat transfer coefficient used in Eq. (13) is obtained by:

$$U_{23} = \frac{1}{r_o} \cdot \left[\frac{1}{\alpha_i \cdot r_i} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{K_M} + \frac{1}{\alpha_o \cdot r_o}\right]^{-1} \tag{14}$$

With the help of a modified version of the Gnielinski's correlation (14a) for the refrigerant flowing inside the tubes, α_i. If Re < 10000 then use,

$$\alpha_i = \frac{K_v}{2 \cdot r_i} \cdot \left[\frac{\frac{Fric}{8} \cdot (Re - 1000) \cdot Pr}{1 + 12.7 \cdot \left(\frac{Fric}{8}\right)^{0.5} \cdot \left(Pr^{(2/3)} - 1\right)}\right] \cdot \left[\frac{\mu_v}{\mu_{vM}}\right]^{0.11} \tag{14a}$$

Or, If Re >10000 then use,

$$\alpha_i = \left[\frac{K_v}{2 \cdot r_i}\right]^{0.027} \cdot Re^{(4/5)} \cdot Pr^{(1/3)} \cdot \left[\frac{\mu_v}{\mu_{vM}}\right]^{0.14} \tag{14b}$$

Where,

$$Fric = \frac{1}{(0.79 \cdot \ln(Re) - 1.64)^2}$$

and the heat transfer coefficient (natural-convection) is obtained by,

$$\alpha_o = \frac{K_{air}}{D_o} \cdot \left(Nul^{10} + Nutt^{10}\right)^{\left[\frac{1}{10}\right]} \tag{15}$$

Where,

$$Nul = 2 \cdot \frac{fc}{\ln\left[1 + \frac{2 \cdot fc}{NuT}\right]}$$

Nutt = 0.103. Ra $^{(1/3)}$

Being

N_{uT}= 0.7772 . 0.103. Ra$^{0.25}$

F = 1 - $\frac{0.13}{NuT^{0.16}}$

$$Ra = g \cdot \frac{1}{T_{air}} \cdot (T_M - T_{air}) \cdot \frac{(2 \cdot r_o)^3}{\dfrac{\mu_{air} \cdot K_{air}}{\rho_{air} \cdot \rho_{air} \cdot Cp_{air}}} \tag{16}$$

3.3 Condenser Formulation

The condenser behavior is modeled by considering heat exchanger into two zone first is *the superheated vapor zone* and second is *the condensing zone*, here it is assume that no sub-cooling at the outlet of the condenser, as it has been consider in the assumptions. The overall heat exchanger is computed with two energy balance equation one using the heat flow in secondary fluid and another is the heat flow in the primary circuit refrigerant with LMTD correction factor considering equal to 1 for simplifying the equation. As below,

$$m_r \cdot (h_3 - h_{Vsat}) = m_w \cdot Cp_w \cdot (Tw_{out} - Tw_m) \tag{17}$$

$$m_r \cdot (h_3 - h_{Vsat}) = U_v \cdot S_v \cdot \left[\frac{T_k - Tw_m - (T_3 - Tw_{out})}{\ln\left(\dfrac{T_k - Tw_m}{T_3 - Tw_{out}}\right)} \right] \tag{18}$$

Where U_k is the average heat transfer coefficient obtained by eq. (19),

$$U_k = \frac{U_v \cdot S_v + U_{vl} \cdot (S_k - S_v)}{S_k} \tag{19}$$

being,

S_K is the heat exchanger overall heat transfer area.
S_V is the superheated vapor zone theoretical heat transfer area computed from eq. (20).

$$S_{vL} = S_k - S_v \tag{20}$$

$$m_r \cdot (h_{Vsat} - h_{Lsat}) = m_w \cdot Cp_w \cdot (Tw_m - Tw_{in}) \tag{21}$$

$$m_r \cdot (h_{Vsat} - h_{Lsat}) = U_{vl} \cdot S_{vL} \cdot \left[\frac{T_k - Tw_m - (T_k - Tw_{in})}{\ln\left(\dfrac{T_k - Tw_m}{T_k - Tw_{in}}\right)} \right] \tag{22}$$

For the calculation of the partial coefficient of heat transfer, U_V and U_{VL}, the convective heat transfer coefficient in the water side is obtained using Eq. (14). For the calculation of the convective heat transfer coefficient for the refrigerant one can distinguish between the convective heat transfer coefficient can be obtain by eq. (23) in the superheated vapor zone,

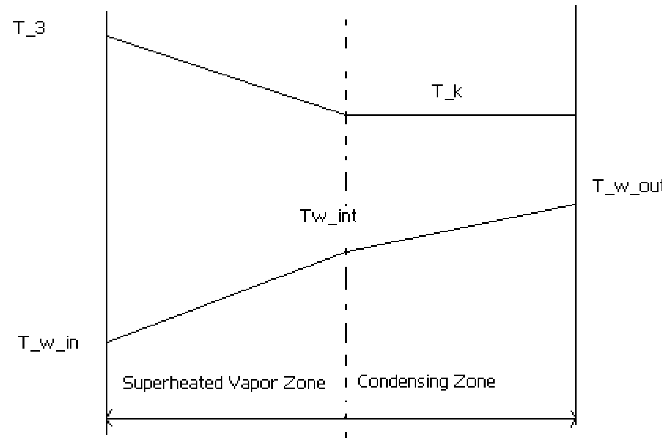

Figure 16: Temperature variation in condenser

$$\alpha_v = \frac{K_v}{D_o} \cdot C_1 \cdot Re^{m_1} \cdot Pr^{0.36} \cdot \left[\frac{\mu_v}{\mu_{vM}}\right]^{0.25} \tag{23}$$

Where, C_1 and m_1 depend on the value of Reynolds number, and the convective heat transfer coefficient in the condensing zone, obtained using eq.(24),

$$[C_1, m_1] = \begin{cases} [0.9, 0.4] & Re \le 100 \\ [0.683, 0.466] & 100 < Re \le 1000 \\ [0.4, 0.6] & 1000 < Re \le 200000 \\ [0.022, 0.84] & Re > 200000 \end{cases}$$

$$\alpha_k = 0.729 \cdot \left[\frac{g \cdot \rho_L \cdot (\rho_L - \rho_v) \cdot \lambda_{k,mod} \cdot K_l^3}{\mu_l \cdot T_{abs} \cdot 2 \cdot r_o} \right]^{0.25} \tag{24}$$

Here λ_K mod is the modified latent heat with considering effects of thermal advection,

$$\lambda_{k,mod} = \lambda_k \cdot (1 + 0.68 \cdot j_a) \tag{25}$$

Here,

$$\lambda_k = h_3 - h_{Vsat}$$

and J_a represent the Jacobsen's number computed by eq.(26).

$$j_a = \frac{Cp_L \cdot T_{abs}}{\lambda_k} \qquad (26)$$

and

$$\dot{T}_{abs} = |T_k - T_M \qquad (27)$$

All these property are used to compute the performance of VCRS using refrigerant in the above equation from (1) to (28). Thus we can find out the performance of VCRS using refrigerant in primary circuit. In this paper, the proposed model on the performance of VCRS is computed using refrigerants. Based on the above formulation we can calculate the thermo physical property are used to calculate the thermal performance of VCRS and we can find out the performance parameter of VCRS using ecofriendly refrigerants. Based upon all the formulation given above set of linear and nonlinear equation is prepared in the EES to calculate the operating performance of VCRS. In this modeling of VCRS all the property are taken in S.I. unit , temperature in °K, energy in Joule (J) specific property mass basis and pressures in bar.

4. Results and Discussion

4.1 Solution Methodology

In this paper, we discussed about the solution methodology of the various formulation used for design components of the system, we use for solution of formulation of the components by EES in which we supply the initial conditions of the software before solving, there should be No. of equations is equal to No. of variables.

First all of write all the equation for design of the components of the system. Then we put inputs as per our model and for output we put some guess values in the software then solve the equations one after one components. First we design evaporator then compressor then condenser. For design evaporator we “comment” the other components and check the No. of equations is equal to No. of variables then solve the formulation, if there is some error while solving then we update our guess nearest values of our design.for the design of the components set all the inputs like, size of the evaporator and condenser, mass flow rate of brine and water, compressor speed, Temperature of brine and water. So, as per our objective to constant our inputs data.

4.2 Initial condition for Vapour compression refrigeration system

In our experiment, Condenser and evaporator s are selected as concentric tube type. In which condenser outer tube diameter is 5/8” and condenser inside tube diameter is 3/8”.

Similarly the evaporator outer tube diameter is 5/8” and inside tube diameter is 3/8”. The initial conditions for vapour compression refrigeration system are shown in table 4.1(a-b).

4.3 Model Validation

The validation of the model using experimental measurements of different steady states was evaluated. To this end, three sets of steady state experiments have been undertaken. Each set of experiments consists of a group of tests where the facility is working at a defined set of inputs as shown in table 4.1 (a) respectively

Table-4.1(a): Inputs of the design and experimental test rig of vapour compression refrigeration system using R134a

S. No	m_b (kg/s)	m_w (kg/s)	Tb_{in} (°C)	Tw_{in} (°C)	N (rpm)	Condenser size (m)	Evaporator size (m)
1.	0.006	0.008	25	25	2900	1.2	0.8
2.	0.007	0.008	25	25	2900	1.2	0.8
3.	0.008	0.008	25	25	2900	1.2	0.8

Table 4.1(b): Experimental output data of vapour compression refrigeration system using R134a

S. No	*T_eva(°C)*	*T_cond(°C)*	*T_brine_{out}(°C)*	*$T\_water\__{out}$(°C)*	*P_eva (bar)*	*P_Cond (bar)*	*$COP\__{Actual}$*
1.	-1.8	42.10	13.1	34.70	2.86	12.90	2.67
2.	-0.7	43.60	14.3	36.10	2.56	11.80	2.75
3.	1.1	46.30	16.4	35.20	2.80	12.64	2.84

Table 4.1 (c): Computational or predict data of vapour compression refrigeration system using R134a for m_w=0.008 Kg/sec

S. No	*T_eva(°C)*	*$T\__{Cond}$(°C)*	*$T\__{brine\_out}$(°C)*	*$T\__{water\_out}$(°C)*	*P_eva (bar)*	*$P\__{Cond}$ (bar)*	*$COP\__{Predicted}$*
0.006	-1.501	48.25	12.9	37.01	2.774	12.62	2.827
0.007	-0.78	51.32	13.19	40.82	2.847	13.63	2.978
0.008	0.277	49.17	15.29	37.69	2.96	12.91	3.131

4.4 Prediction in comparison of numerically calculated results with experimental measurements.

For the experiment we use refrigerant is R134a and experimental measured data is shown in table 4.1(b), table 4.2(a) shows the initial input values used for numerical computation in the developed model and predicted results from model are shown in table 4.2(b) and experimental measured results are presented in table 4.3 (d) respectively.

Table 4.2 (a): Inputs for the test rig and computational data of vapour compression refrigeration system using R134a

S. No	m_b (kg/s)	m_w (kg/s)	Tb_{in}(°C)	Tw_{in} (°C)	N (rpm)	Condenser size (m)	$Eff\__{Volumetric}$	Evaporator size (m)
1.	0.006	0.007	25	25	2900	1.2	0.6113	0.8
2.	0.007	0.007	25	25	2900	1.2	0.6138	0.8
3.	0.008	0.007	25	25	2900	1.2	0.6159	0.8

Table 4.2 (b): Predicted output data of vapour compression refrigeration system using R134a , for m_w=0.007 Kg/sec

m_b (kg/s)	$T\__{eva}$ (°C)	$T\__{cond}$ (°C)	$T\__{brineout}$ (°C)	$T\__{water\_out}$ (°C)	P_eva (bar)	P_Cond (bar)	$Eff\__{Isentropic}$	$COP\__{Predicted}$
0.006	-1.199	49.6	13.03	38.66	2.805	13.03	0.7335	2.906
0.007	-0.2103	50.0	14.35	39.07	2.908	13.21	0.7437	2.985
0.008	0.5925	50.6	15.4	39.42	2.993	13.36	0.7522	3.05

The numerical computation was carried out using developed model for varying brine mass flow rate in the secondary circuit of evaporator from 0.006(kg/sec) to 0.008 (kg/sec) and fixing water flow rate in the secondary circuit of condenser is 0.006 (kg/sec) as shown in table 4.3(a) and results are shown in table 4.1 (d) to 4.3(b) respectively.

Table 4.3(a): Inputs for the test rig and computational.dataof vapour compression refrigeration system using R134a

S. No	m_b (kg/s)	m_w (kg/s)	Tb_{in}(°C)	Tw_{in} (°C)	N (rpm)	Condenser size (m)	Evaporator size (m)
1.	0.006	0.006	25	25	2900	1.2	0.8
2.	0.007	0.006	25	25	2900	1.2	0.8
3.	0.008	0.006	25	25	2900	1.2	0.8

Table 4.3(b): Predicted output data of vapour compression refrigeration system using R134a for m_w=0.006 Kg/sec

m_b (kg/s)	T_eva (°C)	T_cond (°C)	T_brine_{out} (°C)	$T\_water\__{out}$ (°C)	P_eva (bar)	P_Cond (bar)	$COP\__{Actual}$
0.006	-0.7882	51.3	13.19	40.82	2.847	13.61	2.823
0.007	0.2105	51.4	14.51	41.3	2.952	13.82	2.895
0.008	1.021	52.4	15.55	41.69	3.04	13.99	2.956

Table 4.3 (c): Computational or predict data of vapour compression refrigeration system using R134a for m_w=0.008 Kg/sec

M_Brine Kg/sec	T_eva(°C)	$T\__{Cond}$(°C)	$T\__{brine\_out}$(°C)	$T\__{water\_out}$(°C)	P_eva (bar)	$P\__{Cond}$ (bar)	$COP\__{Predicted}$
0.006	-1.501	48.25	12.9	37.01	2.774	12.62	2.827
0.007	-0.78	51.32	13.19	40.82	2.847	13.63	2.978
0.008	0.277	49.17	15.29	37.69	2.96	12.91	3.131

Table 4.3 (d) Experimental data of vapour compression refrigeration system using R134a for mw=0.008 Kg/sec

$m\__{Brine}$ Kg/sec	T_eva(°C)	T_cond(°C)	$T\_brine\__{out}$(°C)	$T\_water\__{out}$(°C)	P_eva (bar)	P_Cond (bar)	$COP\__{Actual}$
0.006	-1.8	42.10	13.1	34.70	2.86	12.90	2.67
0.007	-0.7	43.60	14.3	36.10	2.56	11.80	2.75
0.008	1.1	46.30	16.4	35.20	2.80	12.64	2.84

Table 4.4 (a): Validation of performance data from Experimental and predicted from model using R134a for mw=0.008 Kg/sec

$m\__{Brine}$ Kg/sec	$T\__{eva}$(°C) Exp	$T\__{eva}$ (°C) Predicted	$T\__{Cond}$(°C) Exp	$T\__{Cond}$(°C) Predicted	$T\__{water\_out}$(°C) Exp	$T\__{water\_out}$(°C) predict	$COP\__{Actual}$ Exp	$COP\__{Predicted}$
0.006	-1.8	-1.501	42.10	48.25	34.70	37.01	2.67	2.978
0.007	-0.7	0.277	43.60	49.17	36.10	37.69	2.75	3.131
0.008	1.1	-0.78	46.30	51.32	35.20	40.82	2.84	2.827

Table 4.4 (b): Validation of performance data from Experimental and predicted from model using R134a

$m_{\_Brine}$ Kg/sec	$T_{\_brine\_out}$ (°C) Exp	$T_{\_brine\_out}$ (°C) predict	P_eva (bar) Exp	P_eva (bar) predict	$P_{\_Cond}$ (bar) Exp	$P_{\_Cond}$ (bar) predict	$T_{brine\ \_in}$ (°C) Exp	$T_{\_Water\_in}$ (°C) Exp
0.006	13.1	12.9	2.86	2.774	12.90	12.62	27	27
0.007	14.3	15.29	2.56	2.96	11.80	12.91	27	27
0.008	16.4	13.19	2.80	2.847	12.64	13.63	27	27

To validate predicted values from developed thermal model with experimental results as shown in Table 4.4(a-b). It is seen that our developed model predicts experimental behavior with the minor variation of around 10% accuracy. The variation of first law efficiency in terms of coefficient of performance with varying brine flow rate in Kg/sec using different refrigerants is shown in Fig. 17 respectively. By changing brine mass flow rate 0.004 to 0.008 kg/s change in COP value for R134a is 14.10 %, similarly for ecofriendly refrigerant R404a is 13.94%, other ecofriendly refrigerant R407c is 14.39% and using hydrocarbon R290 is 17.06%.

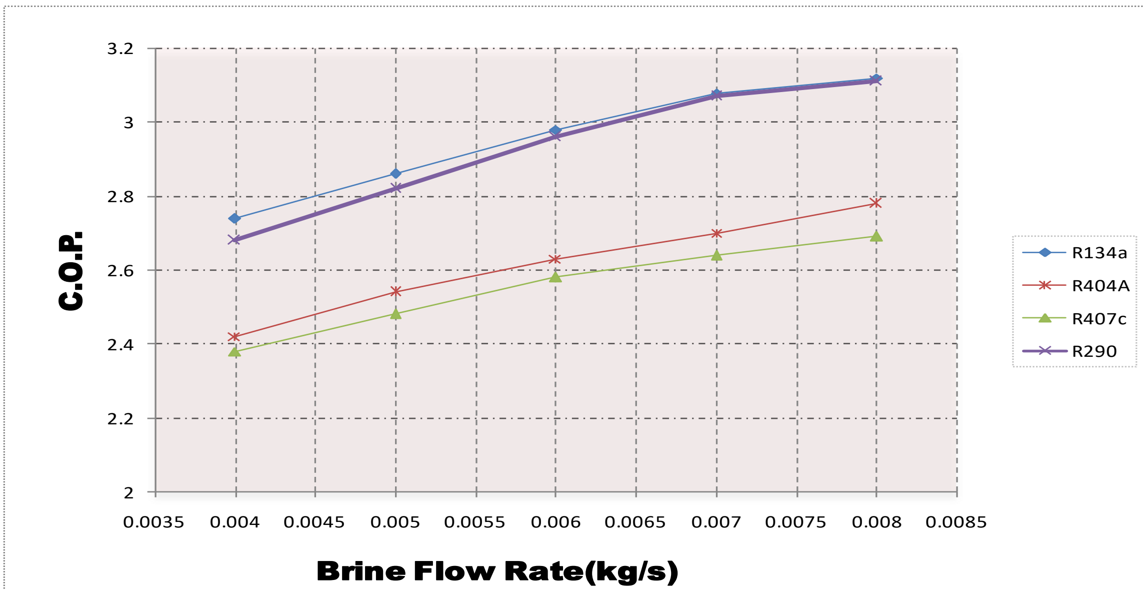

Figure 17: Variation of C.O.P with brine flow rate of VCRS using different refrigerants

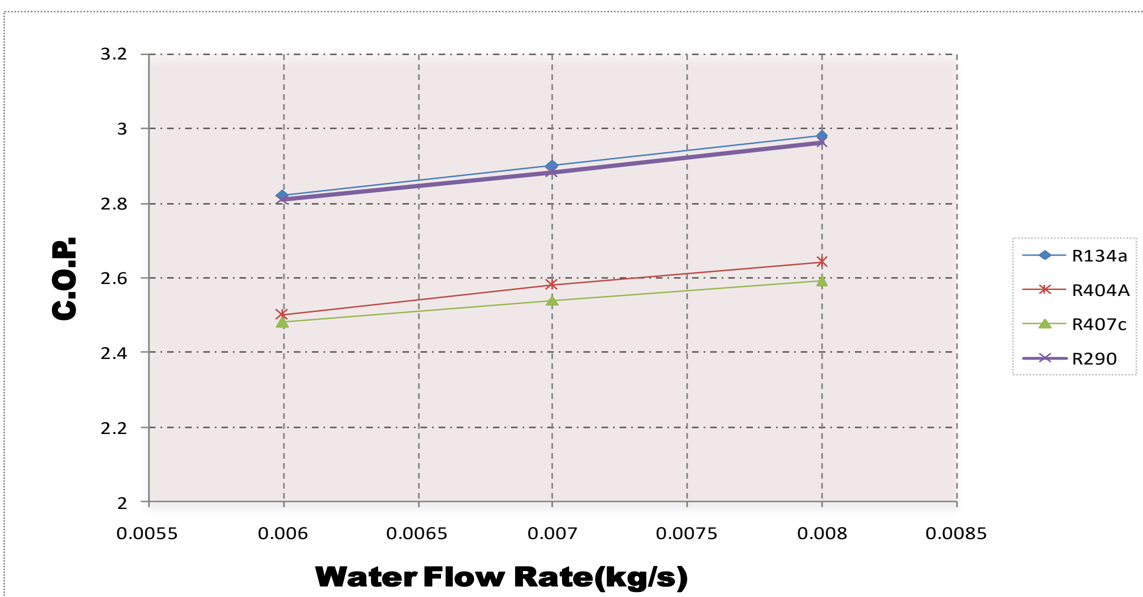

Figure 18: Variation of C.O.P with condensing water flow rate of VCRS using different refrigerants

The first law efficiency in terms of coefficient of performance with varying condenser secondary circuit water flow rate in Kg/sec using different refrigerants is shown in Fig-18 respectively. When water mass flow rate in the secondary circuit of condenser is changing from 0.006 to 0.008 kg/s (33.3%) then change in COP for R134a is 5.54%, R404a is 5.65%, R407c is 3.58% and R290 is 5%. The first law efficiency in terms of coefficient of performance with varying condensing water inlet temperature of different refrigerants is shown in Fig.19. When condensing water inlet temperature is changed from 18 to 30^{0}C (66.67%) then change in COP using R134a in Vapour compression refrigeration system is 20.27%. similarly by using ecofriendly refrigerant R404a is 16.13%, and another ecofriendly refrigerant R407c is 12.50% and using hydrocarbon R290 is 16.32%.

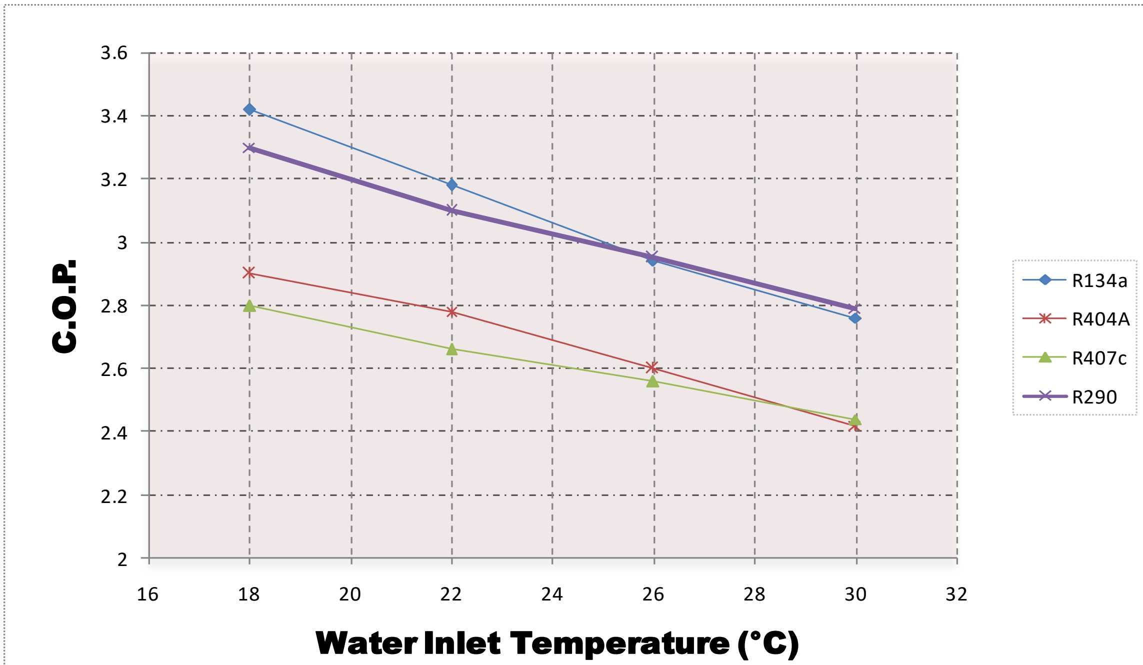

Figure 19: Variation of C.O.P with condensing water inlet temperature of VCRS using different refrigerants

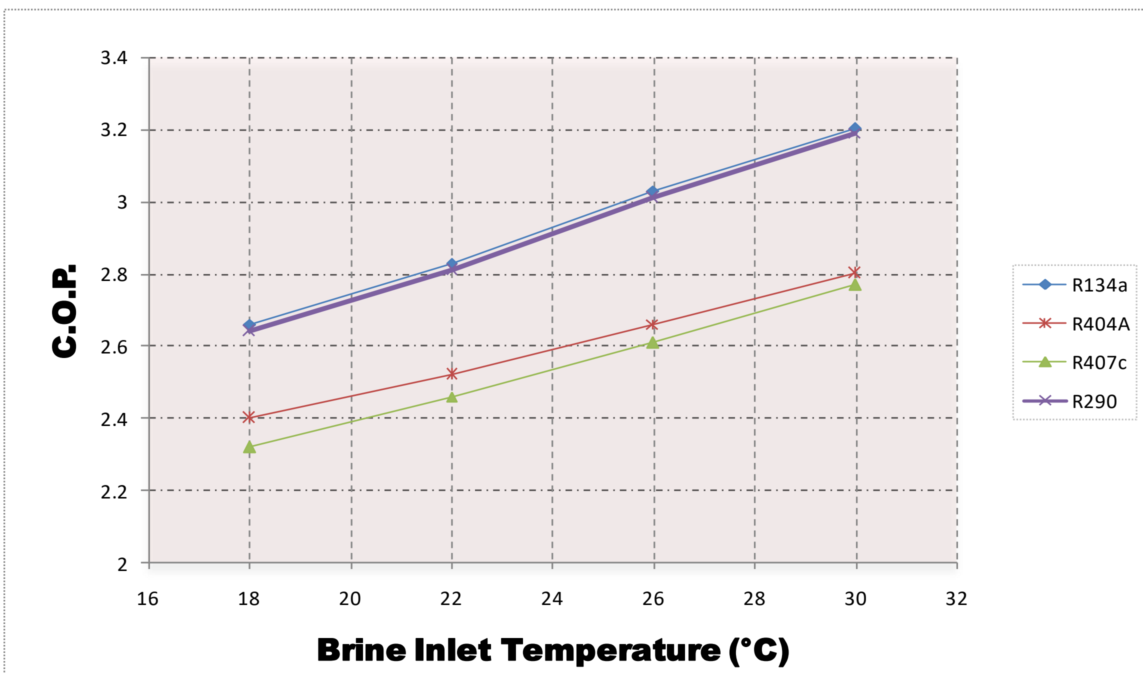

Figure 20: Variation of C.O.P with Brine inlet temperature of VCRS using different refrigerants

The first law efficiency in terms of coefficient of performance with varying Brine inlet temperature in evaporator secondary circuit in Kg/sec using different refrigerants is shown in Fig. 20. When brine inlet temperature variation from 18 to 30ºC (66.67%) then change in the first law efficiency in terms of COP using ecofriendly R134a refrigerant is 20.46%, R404a is 17.15%, R407c is 18.47% and using hydrocarbon R290 is 20.54%.

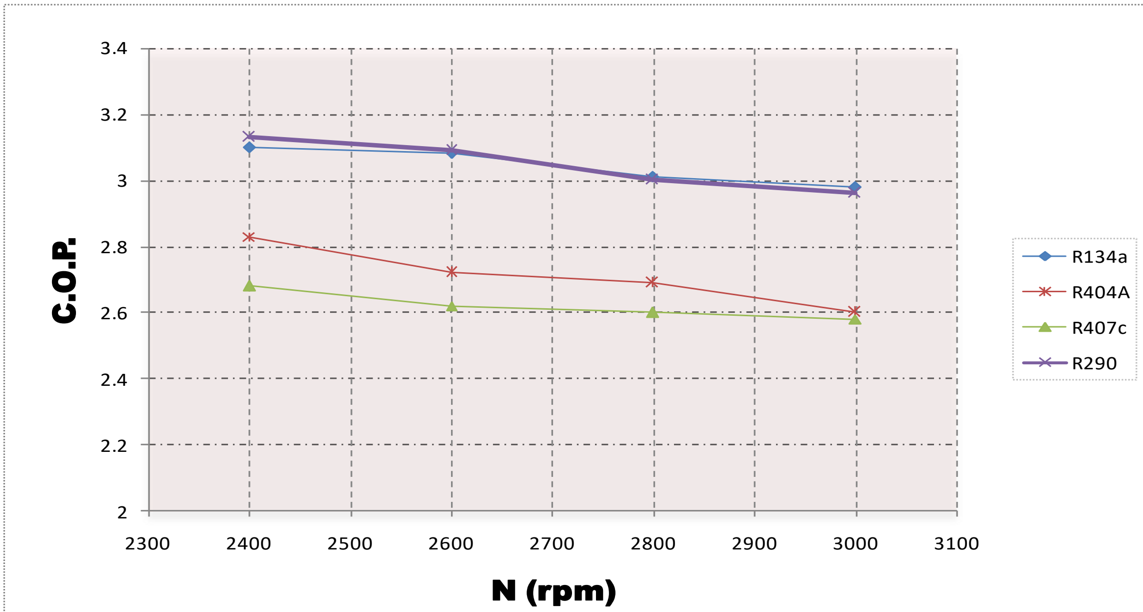

Figure 21: Variation of C.O.P with Compressor speed of VCRS using different refrigerants

The first law efficiency in terms of coefficient of performance with varying COP with the speed of the compressor N (rpm) of different refrigerants is shown in Fig.21. When compressor speed is varying from 2400 to 3000 rpm (25%) then change in the first law efficiency in terms of COP using R134a is 5.55%, using R404a is 7.38%, R407c is 5.08% and using hydrocarbon\ R290 is 6.98%.

4.5 Relative Comparison of various ecofriendly refrigerants used in the vapour compression refrigeration system

Table 4.5 showed the comparison between most commonly used four refrigerants such as R134a, R404A, and R407cand R290. It can be seen from the table that R134a have highest C.O.P. than other refrigerant for the same geometry and input parameter of the VCRS. It is because compressor work reduces about 20-30 % than other refrigerant by using R134a in VCRS. Also working pressure ratio is little lower than the other refrigerant. So that R134a is most commonly used in HVAC and automobile AC system.R407C exhibits a relatively high temperature glide compared to the other refrigerants, which have almost no glide. It also offer '0' ODP, low global warming potential. European market embraced R407C and currently offers a wide R407C AC product range. Further, a change to polyester lubricant is also required. R404A has been in the market place for more than 10 years.

Table 4.5: Physical and environmental characteristics of selected refrigerants

Properties	R134A	R404A	R407C	R290
Molecular Weight (kg / Kmol)	102	97.6	86.20	44.1
B.P. at 1.013 bar (°C)	-26.1	-51.4	-43.6	-42.2
Critical temperature (°C)	101.1	72.15	85.8	96.68
Critical pressure (bar)	40.60	37.35	46.00	42.47
ODP	0	0	0	0
GWP_{100}	1300	3260	1800	3

It is shown from table 4.5, that the value of first law efficiency in terms of coefficient of performance (C.O.P.) value of ecofriendly R134a refrigerant and hydrocarbon R290 is quite similar but ecofriendly R404A and R407C refrigerants have very less value of C.O.P than R134a and R290. Therefore the ecofriendly R134 refrigerant and hydrocarbon R290 are more efficient considering first law thermodynamic efficiency (COP). Similarly, the compressor work for different refrigerant respectively and it can be seen that compressor work is also high of R407C and R404A than the R134a and R290 so that its thermal performance in terms of COP is reduced The refrigeration effect R134a, R290, R404A and R407C respectively and it is very clear that refrigeration effect of R134a is less than the R290 but due to higher compressor work of R290 than R134a the C.O.P. value of R290 is lower than the R134.fig also show that refrigeration effect of R404A is higher than the R407C.

4.6 Characteristic performance

The characteristic performance of vapor-compression refrigeration systems using R134a, refrigerants by varying brine flow rate is shown in table 4.6 and Comparison of performance parameters for different refrigerants are shown in table 4.7 respectively.

Table 4.6: Comparison of performance parameters for different brine flow rates using $m_{water\_Flow\_Rate}$=0.008 (Kg/sec) and ecofriendly R134a

Brine flow rate (kg/sec)	0.006	0.007	0.007	0.009	0.010
COP_VCRS	2.973	3.056	3.126	3.185	3.237
EDR_Rational	0.710	0.714	0.7174	0.7204	0.7232
Exergetic Efficiency	0.290	0.286	0.2826	0.2796	0.2768
EDR	2.448	2.496	2.539	2.578	2.613
Compressor work (W)	102.1	103.1	103.9	104.5	105.1
Refrigerating effect (W)	303.6	315.1	324.7	332.9	340.0
Evaporator Heat Transfer Coefficient ($W/m^{2o}C$)	639.73	670.43	697.28	721.11	742.52
Condenser Heat Transfer Coefficient ($W/m^{2o}C$)	646.35	657.41	666.54	674.24	680.84
Condenser pressure (bar)	12.61	12.77	12.9	13.02	13.12
Evaporator pressure (bar)	2.774	2.876	2.96	3.032	3.094
Condenser Temperature (°C)	48.22	48.72	49.14	49.49	49.79
Evaporator Temperature (°C)	-1.497	- 0.5151	0.2822	0.9464	1.511
Brine outlet temperature (°C)	12.9	14.26	15.3	16.16	16.87
Water outlet temperature (°C)	37.01	37.69	37.88	37.96	38.19

Table 4.7: Comparison of performance parameters for different refrigerants

Parameters	R134A	R404A	R407C	R290
COP	2.978	2.638	2.574	2.959
Compressor work (W)	102	131.4	127.8	121.2
Refrigerating effect (W)	303.7	346.5	329	358.4
Mass flow rate (kg/s)	0.00236	0.0047	0.0027	0.00157
Condenser pressure (bar)	12.62	27	23.27	17.16
Evaporator pressure (bar)	2.774	5.788	4.208	9.148
Condenser Temperature (°C)	48.25	56.94	52.49	51.26
Evaporator Temperature (°C)	-1.501	-1.338	-2.272	-4.326
Brine outlet temperature (°C)	12.9	11.19	11.89	10.72
Water outlet temperature (°C)	37.01	39.16	38.47	39.23

Table 4.8: Comparison of performance parameters for different refrigerants

Parameters	COP	Q_Eva (W)	W_Comp (W)
R134a	2.978	303.7	102
R404a	2.638	346.5	131.4
R407c	2.574	329	127.8
R290	2.959	358.4	121.2

The Numerical computed results obtained from developed thermal model for constant brine mass flow rate in the secondary circuit of evaporator and by keeping condenser constant water flow rate (0.008 kg/sec) is shown in table 4.9.

Table 4.9: Enhancement in C.O.P using different nano refrigerant of VCRS

Refrigerant	R134a	R404A	R410a	R1234yf	R1234ze	R125
m_{brine} (kg/sec)	C.O.P	C.O.P	C.O.P	C.O.P	C.O.P	C.O.P
0.006	2.973	2.641	2.556	2.938	2.959	2.379

5. Conclusion

The research work presented in this paper, following conclusions have been drawn.

(i) The thermal performance of R134a is excellent & slightly higher than HFO refrigerants and other refrigerants. Therefore HFO refrigerants can replace R134a in coming future.
(ii) The lowest performance was observed by using R125 and R410a
(iii) The thermal performance of vapour compression refrigeration system predicted from developed thermal model matched very well with the experimental results performed on the system.

References

[1] Ying Wang, Fang Wang, Hongqun Chen, Fang Li Siyu Li [2016] Application of Environmental Friendly Material R32/R1234ze in Heat Pump Water Heaters, International Symposium on Mechanical Engineering and Material Science (ISMEMS 2016) & Advances in Engineering Research, volume 93, page- 420-425,

[2] Kapil Chopra , V. Sahni , R. S Mishra [2013] Energetic and Exergetic Based Comparison Multiple Evaporators with Compound Compression and Flash Intercooler with Individual or Multiple Throttle Valves, International Journal of Advance Research and Innovation, ISSN 2347 – 3258, Volume 1, Issue 1 (2013) 73-81.

[3] Kapil Chopra , V. Sahni , R. S Mishra [2014] Methods for Improving First and Second Law Efficiencies of Vapour Compression Refrigeration Systems Using Flash-Intercooler with Ecofriendly Refrigerants International Journal of Advance Research and Innovation, ISSN 2347 – 3258, Volume 2, Issue 1, 50-64.

[4] S.Kumar, M.Prevost, R.Bugarel-[1989] Exergy analysis of a vapour compression refrigeration system. Heat Recovery Systems & CHP.1989;9:151-157.

[5] B.O. Bolaji, M.A. Akintunde, T.O. Falade.[2011] Comparative analysis of performance of three ozone-friends HFC refrigerants in a vapor compression refrigerator. Int J Sustainable Energy & Environment.2011; 2:61-64.

[6] Recep Yumrutas, Mehmet Kunduz, Mehmet Kanoglu-Exergy analysis of vapor compression refrigeration systems. Exergy, An International Journal.2002; 2:266-272.

[7] Mark W. Spatz, Samuel F. Yana Motta. An evaluation of options for replacing HCFC-22 in medium temperature refrigeration systems. Int J Refrigeration.2004; 27:475-483.

[8] Mahmood Mastani Joybari,et.al.[2013] Exergy analysis and optimization of R600a as a replacement of R134a in a domestic refrigerator system. International Journal of refrigeration.2013; 36:1233-1242.

[9] V. Siva Reddy, N.L Panwar, S.C Kaushik-Exergy analysis of a vapour compression refrigeration system with R134a, R143a, R152a, R404A, R407C, R410A, R502 and R507A.Clean Techn Environ Policy.2012;14:47-53.

[10] Radhey Shyam Mishra(2018)Performance analysis of vapour compression refrigeration systems using eighteen ecofriendly and other CFC refrigerant, International Journal of Research in Engineering and Innovation , ISSN (Online): 2456-6934, vol 2, issue 4 , page349-359.

[11] J. N. Esbri, J.M.M. Miranda, A.M. Babiloni, A.B. Cervera and J.M.B. Flores, “Experimental analysis of R1234yf as a drop-in replacement for R134a in a vapor compression system,” International Journal of Refrigeration 36, 870-880, 2012.

[12] Reaser P.,et.al [2010] “Refrigerant R1234yf Performance Comparison Investigation,” International Refrigeration and Air Conditioning Conference 1085, 20.

[13] R.S. Mishra(2018),Thermodynamic analysis of two stage cascade refrigeration systems using R1234ZE in high temperature circuit and R1234YF in low temperature circuit , International Journal of Research in Engineering and Innovation , ISSN (Online): 2456-6934,Vol-2, Issue 4 page-325-433.

[14] R.S. Mishra(2018) Thermodynamic analysis of two stage cascade refrigeration systems using R-1234ZE in high temperature circuit and R1234YF in low temperature circuit, International Journal of Research in Engineering and Innovation , ISSN (Online): 2456-6934, vol 2, issue 4 325-339.

[15] R S Mishra [2013] Thermodynamic Performance Evaluation of Multi Evaporators Single Compressor and Single Expansion Valve and Liquid Vapour Heat Exchanger in Vapour Compression Refrigeration Systems using Thirteen Ecofriendly Refrigerants for Reducing Global Warming and Ozone Depletion, International Journal of Advance Research and Innovation, ISSN 2347 – 3258, Volume 1, Issue 3 (2013) 220-228.