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Performance and exergy analysis of vapour compression refrigeration system using various alternative of R134a.

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Abstract - *An analysis of energy and exergy on a traditional vapour compression refrigeration system using R152a, R290, R600, R600a, R123 and R717 was done theoretically for different typical ratios and their result was compared with standard refrigerant R134a. The result deduced that these alternative refrigerant R600, R600a, R717 and R152a had higher COP and efficiency (exergetic) than R134a for evaporative temperature which range from 248 K to 283 K and condensation temperature 318 K with superheating 10 K and subcooling 5 K. R600 was found to be a suitable replacement among others. Other parameter namely refrigerant type, degree of subcooling and superheating on exergetic efficiency, COP, RE, VRC, PTR, total exergy destruction were also investigated for different evaporative temperature.*

Key Words: *Alternative refrigerants, exergy analysis, Refrigeration, HC, HFC, HCFC, GWP, ODP.*

List of symbols

E_{Xw}	useful work done on/by system
Ψ_i	Exergy at inlet
Ψ_o	Exergy at outlet
η_{II}	Second law efficiency
Q_k	Heat transfer rate
$\dot{m}$	mass flow rate
To	Ambient Temperature
T_k	temperature of the heat source/sink
F	factor of safety

List of subscripts

comp	Compressor
con	Condenser
dest	Destruction
eva	evaporator
isen	isentropic

list of abbreviations

VRC	Volumetric refrigeration capacity
PTR	Power per ton of refrigeration
RE	Refrigeration Effect
GWP	global warming potential
ODP	ozone depletion potential

1.INTRODUCTION

Now a days, the ODP and GWP have emerged as the most critical criteria leading to the development and production of new refrigerant apart from traditional refrigerant CFC's and HCFC's owing to the fact that both of them have a serious impact to the ozone layer depletion and GWP. In spite of their harsh effects on Ozone layer, they were being used intensively for a decade or so. HFC refrigerants have certain specific characteristics for the likes of non-flammable, stability and similar vapour pressure. The increased global warming and ozone layer depletion leads to the investigation of ecofriendly refrigerants than HFC for the protection of environment such as HC refrigerants of propane, butane, isobutene, n-butane or hydrocarbons as working fluids in refrigeration system. Though HC refrigerants are highly flammable, they not only have zero ODP and zero GWP but also highly miscible with oils. They are used in a variety of applications as for as leakage does not occur. Many investigations have been conducted in the research into substitutes for CFC12 and CFC22. Wongwises et al. [1] presented an experimental study on the application of hydrocarbon mixtures to replace HFC134a in automotive air conditioners. The hydrocarbons investigated are propane (R290), butane (R600), and isobutene (R600a). The measured data are obtained from an automotive air-conditioning test facility utilizing HFC134a as the refrigerant. Wongwises and Chimres [2] presented an experimental study on the application of a mixture of propane, butane, and isobutene to replace HFC134a in a domestic refrigerator. The results showed that a 60%/40% propane/butane mixture was the most appropriate alternative refrigerant. Hammad and Alsaad [3] investigated the performance of a domestic refrigerator using LPG (24.4% propane, 56.4% butane, and 17.2% isobutane), which is available locally in many countries, is cheap, and possesses an environmentally friendly nature with no ozone depletion potential (ODP), as an alternative refrigerant to CFC12. Jung et al. [4] used a propane/isobutane (R290/ R600a) mixture to determine their performance for domestic refrigerators. According to their thermodynamic cycle analysis, the propane/isobutane blend in the composition range from 0.2 to 0.6 mass fraction of propane yields an increase in the coefficient of performance (COP) of up to 2.3% compared to CFC12. Granryd [5] mentioned the possibilities and problems of using hydrocarbons as working fluids in refrigeration equipment. In spite of their flammability specification, it is shown in his paper that alternative refrigerants can be obtained by means of hydrocarbons for energy efficient and environmentally friendly refrigerating equipment and heat pumps. Park et al. [6] tested two pure hydrocarbons and seven mixtures composed of propylene, propane, HFC152a, and dimethyl ether as an alternative to HCFC22 in residential air-conditioners and heat pumps. Their experimental results show that the coefficient of performance (COP) of these mixtures is up to 5.7% higher than that of HCFC22. In this paper pure refrigerants such as R134a, R152a, R290, R600, R600a, R123 and R717 were used to check for the better performance. The parameters such as refrigerant type, degree of subcooling and superheating on the exergetic efficiency, COP, RE, VRC, PTR, total exergy destruction were investigated taking temperature range from 248 K to 283 K and condensation temperature of 318 K with superheating of 10 K and subcooling of 5 K.

Table 1: Some safety and environmental data of selected refrigerants

Refrigerant	**Name:**	**Class**	**Chemical Formula**	**O.D.P.**	**G.W.P.: 100 years**	**Safety Classification**
R134a	Tetrafluoroethane	HFC	$CF_3.CH_2F$	0	1300	A1
R152a	1,1-difluoroethane	HFC	$C_2H_4F_2$	0	140	A2
R290	Propane	HC	$C_3\,H_8$	0	20	A3
R600	Butane	HC	$C_4\,H_{10}$	0	20	A3
R600a	Isobutane	HC	$C_4\,H_{10}$	0	20	A3
R123	Dichlorotrifluoroethane	HCFC	$C.HCl_2CF_3$	0	76	A1
R717	Ammonia	NH_3	NH_3	0	0	B2

2. ANALYSIS

The previous studies were based on analyzing performance of refrigeration system using first law of thermodynamic. But in our analysis we had used both first law of thermodynamic and second law of thermodynamic, allows an improved comprehension of thermodynamic processes by quantifying the effect of irreversibility occurring in the system along with its location.

2.1 Assumptions

To access the performance of selected refrigerants in vapour compression refrigeration cycle, following assumptions were made:

Isentropic efficiency of compressor (η_{isen}) =0.85
Motor Efficiency ($\eta_{motor)}$ =0.9
Factor of safety (F) =0.8
Mass flow rate of refrigerant (m) =1kg/s
Surrounding temperature (T_0) =303K.
Evaporator Temp = 248 to 283K
Condenser Temp = 298 to 323K
Degree of Super heat = 10K
Degree of Sub cooling = 5K
Pressure losses in pipelines are neglected.
Steady state operations are considered in all components.

2.2 Energy Analysis of VCRS

The following formulae were used for energy analysis of VCRS:

- The pressure ratio of the cycle can be seen below as follows
  Pressure ratio=Pcon/ Peva

- Isentropic compression work of the compressor (Wcomp) is expressed as follows:
  Wcomp = h_3-h_2

- The refrigerating effect (RE), in other words, the heat transfer rate of the evaporator (Qevap), is calculated as follows:
  RE = Qeva= h_1-h_6

- Power per ton of refrigeration is calculated as follows:
  Power per ton of refrigeration (P/TR) = 3.5*Wcomp/RE

- Volumetric refrigeration capacity is calculated as follows:
  VRC = cooling capacity/specific volume at compressor inlet (v_2)

2.3 Exergetic Analysis of VCRS

To analyse the possible realistic performance, a detailed exergy analysis of a vapour compression refrigeration system has been carried out by ignoring the kinetic and potential energy change. For steady state flow, the exergy balance for a thermal system can be estimated by using Equation. Cycle diagram of vapour compressor refrigeration is illustrated in Fig. 1, T–S and P–h cycle diagram is presented in Figs. 2 and 3, respectively.

$$EXw=\sum_{k=1}^{n}\left(1-\frac{T0}{Tk}\right)Qk+\sum_{k=1}^{n}[(\dot{m}\Psi)i-(\dot{m}\Psi)o]k$$

The exergy loss for the each component of cycle is given by:

- Exergy destruction in Compressor
  $EX_{dest,comp}=\dot{m}T_0\,(s_3-s_2)$

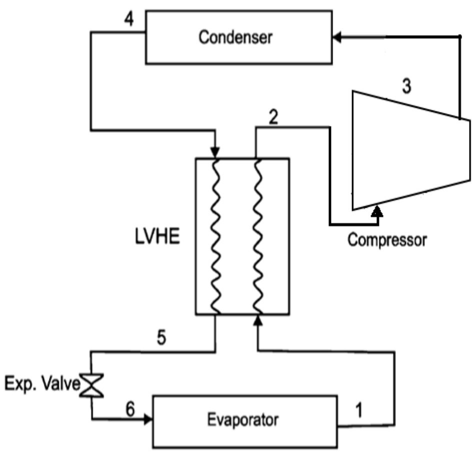

Fig -1: Vapour Compression Refrigeration Cycle

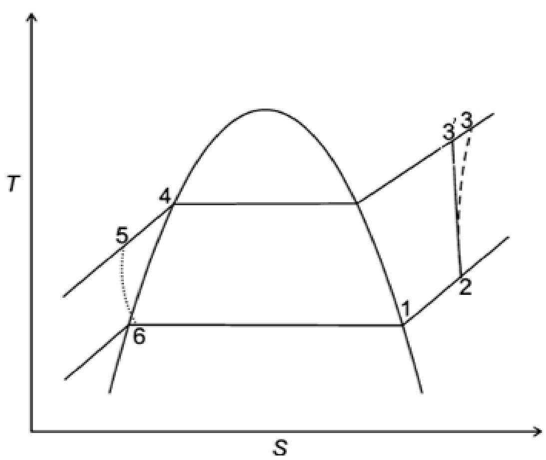

Fig -2: Temperature entropy diagram of VCRS

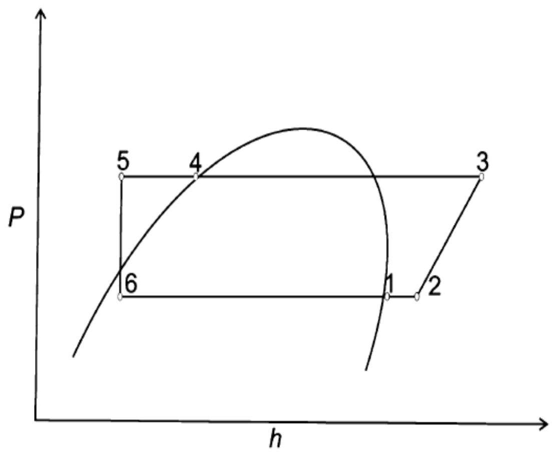

Fig -3: Pressure enthalpy diagram of VCRS

- Exergy destruction in condenser
  $EX_{dest,con}= (\dot{m}\,(h_3-h_4)-T0\,(\dot{m}\,(s_3-s_4))-(1-T_o/T_k)\,Q_k$

- Exergy destruction in Heat exchanger
  $EX_{dest,HE}= (\dot{m}\,(h_4-h_5)-\dot{m}\,(h_2-h_1))-To\,(\dot{m}\,(s_4-s_5)-\dot{m}\,(s_2-s_1)$

- Exergy destruction in throttle valve
  $EX_{dest,tv} = \dot{m}\,(h_5-h6)-To\,(\dot{m}\,(s_5-s_6)$

- Exergy destruction in Evaporator
  $EX_{dest,eva}=(\dot{m}(h_1-h_6)-T0(\dot{m}(s_1-s_6))-(1-T_o/T_k)\,Q_k$

- The cooling COP of the vapour compression refrigeration system is defined as the heat load of the evaporator per unit power input to the compressor and is expressed as
  $COP=Qe\times F\times\eta_{isen\times}\,\eta_{motor}/W_{comp}$

- Total exergy destruction
  $EX_{dest,total}= EX_{dest,comp}+EX_{dest,con}+EX_{dest,HE}+EX_{dest,tv}+EX_{dest,eva}$

- Second law efficiency

  $\eta_{II}=1-(EX_{dest,total}/W_{comp})$

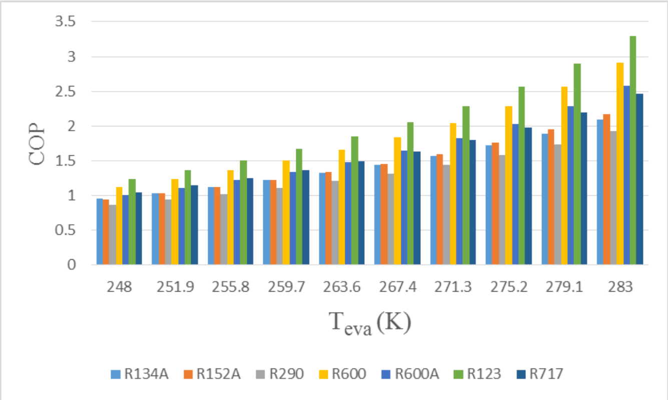

Fig -4: Evaporator temperature Vs COP

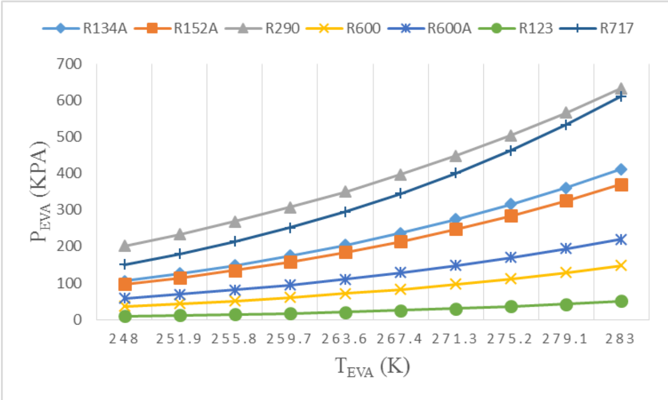

Fig -6: Evaporating Pressure Vs evaporating temperature

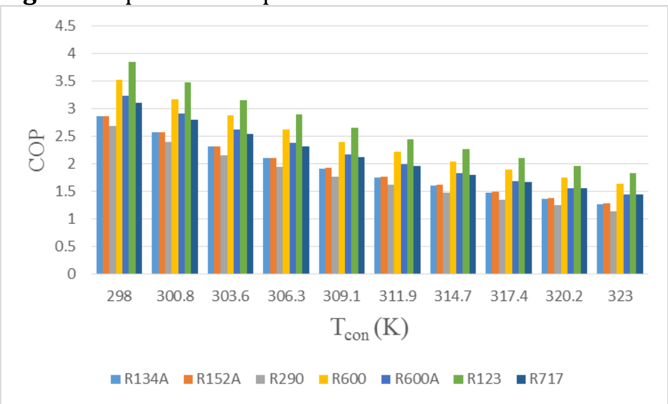

Fig -5: Condenser temperature Vs COP

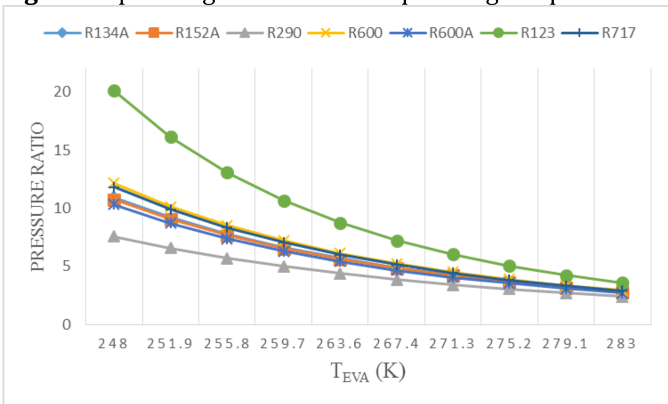

Fig -7: Pressure Ratio Vs evaporating temperature

Table -2: Operation on a standard vapour-compression cycle using R134a and various refrigerants At T_{con}=45°C and T_{eva}=-5°C with super heating 10°C and sub cooling 5°C

Refrigerant	Peva (kPa)	Pcon (kPa)	Pressure Ratio	Wcomp (KJ/kg)	RE (KJ/kg)	Power per ton refrigeration	VRC (kJ/m³)	COP	η_{II}	Total Exergy Destruction
R134a	242.1	1156	4.774	58.48	139.4	1.324	1596	1.459	47.15	30.9
R152a	218.8	1034	4.725	95.91	231.6	1.307	1545	1.478	47.71	50.15
R290	404.2	1529	3.784	120	261.6	1.448	2208	1.335	43.58	67.69
R600	84.67	433.2	5.116	92.25	281.1	1.036	618.4	1.865	59.19	37.65
R600a	130.8	598.1	4.571	89.61	243.9	1.16	843.4	1.666	53.27	41.87
R123	25.71	181	7.041	40.47	137.7	0.9277	237.7	2.083	65.05	14.15
R717	352.8	1775	5.032	394.7	1067	1.168	2922	1.654	52.75	186.5

Table-3: Some deviation values of alternative refrigerants from R134a

R134a at T_{con}=45°C and T_{eva}=-5°C with super heating 10°C and sub cooling 5°C							
Refrigerant	Pressure Ratio (%)	Wcomp (%)	RE (%)	Power per ton refrigeration(%)	VRC(%)	COP(%)	η_{II} (%)
R152a	-1.03	64	66.14	-1.28	-3.20	1.30	1.19
R290	-20.74	105.2	87.66	9.37	38.35	-8.50	-7.57
R600	7.17	57.75	101.6	-21.75	-61.25	27.83	25.54
R600a	-4.25	53.23	74.96	-12.39	-47.16	14.19	12.98
R123	47.49	-30.8	-1.22	-29.93	-85.11	42.77	37.96
R717	5.40	574.9	665.4	-11.78	83.08	13.37	11.88

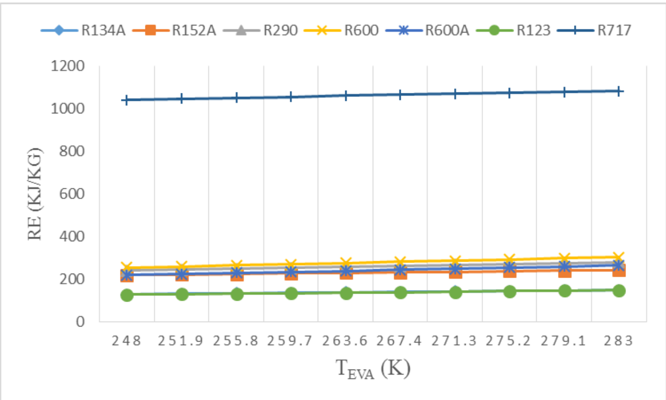

Fig -8: Refrigeration effect Vs evaporating temperature

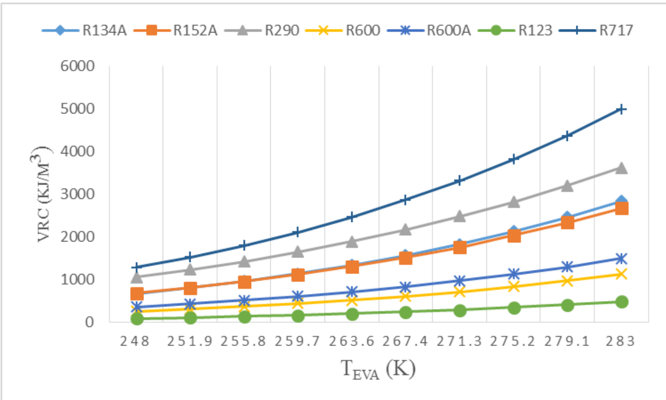

Fig -10: VRC Vs evaporating temperature

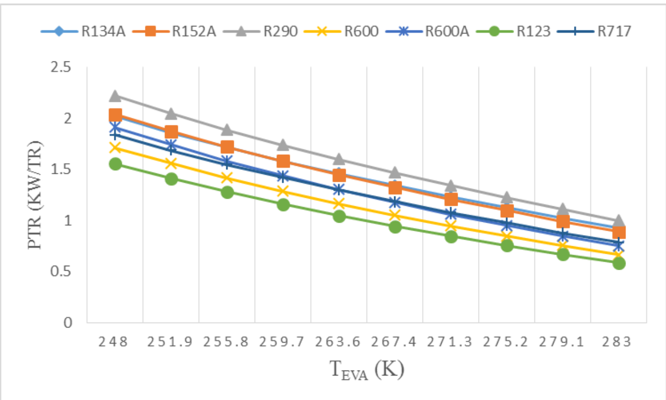

Fig -9: PTR Vs evaporating temperature

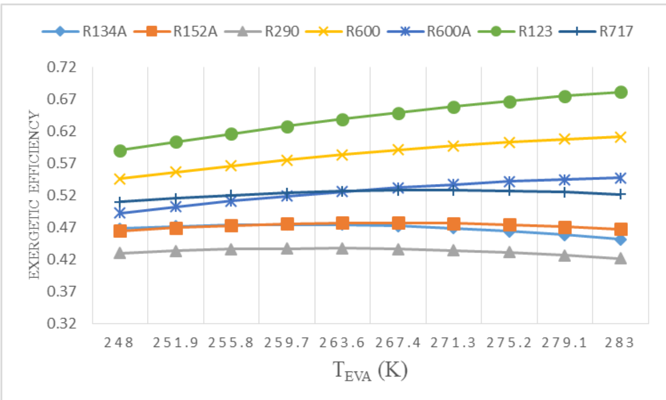

Fig -11: Exergetic efficiency Vs evaporating temperature

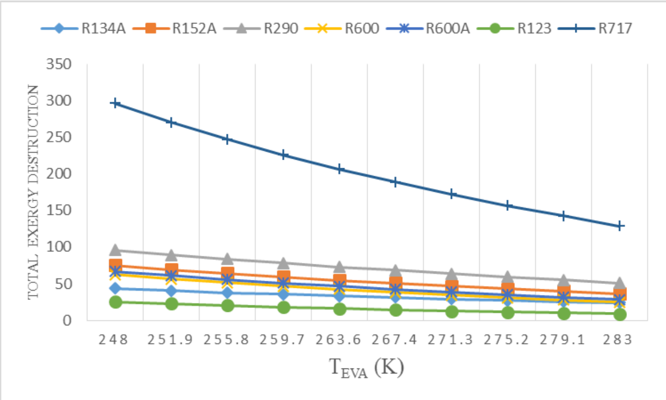

Fig -12: Total exergy destruction Vs evaporating temp.

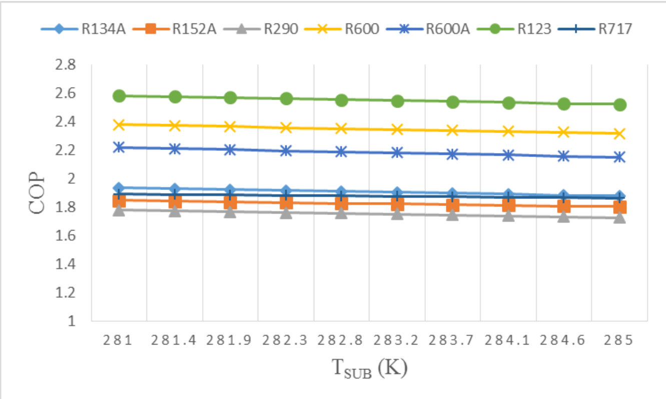

Fig -13: Subcooling temperature Vs COP

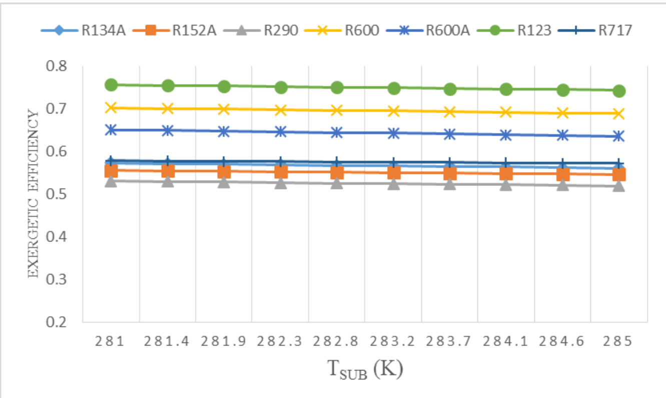

Fig -14: Subcooling Vs exergetic efficiency

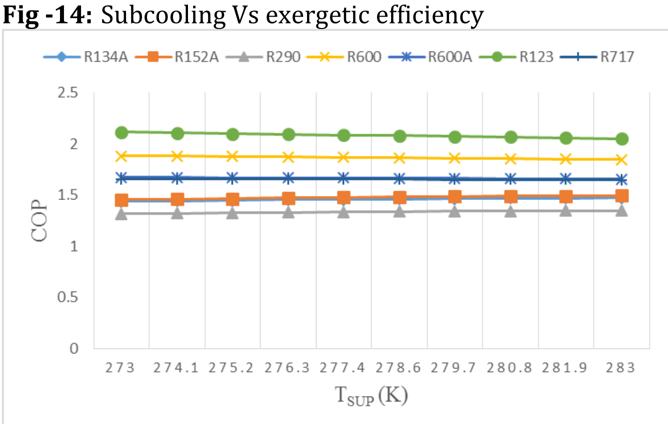

Fig -15: Superheating temperature Vs COP

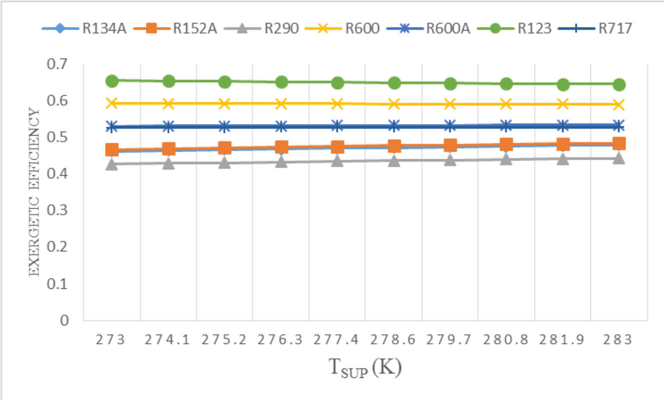

Fig -16: Superheating Vs exergetic efficiency

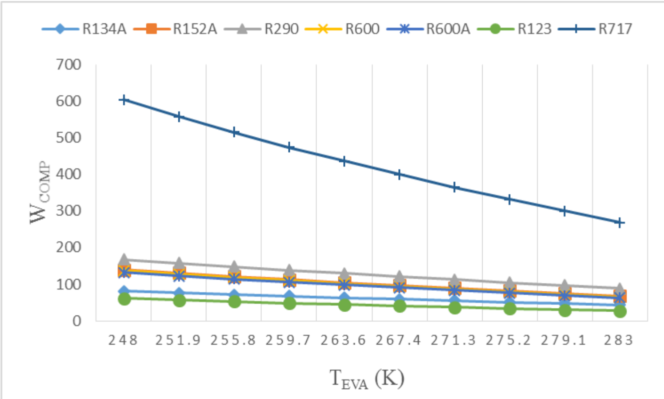

Fig -17: Compression work Vs evaporating temperature

3. RESULT AND DISCUSSIONS

The changes in evaporating pressure (Pevap) and pressure ratio with the evaporation temperature (Tevap) were shown in fig.6 and 7 for listed refrigerants. The nearest pressure ratio of refrigerant substituted for R134a belongs to R600a whose pressure ratio was 4.25% lower than that of R134a as shown in table 3 for the constant condensation and evaporation temperatures of 318 K and 268 K respectively. In addition to this R152a gives the lowest ratio as substitute for R134a according to the same table. It can be seen that the saturated vapour pressure for R600a and R600 was closer to the vapour pressure curve of the refrigerant R134a than others. Fig 8 and 17 show that the refrigerating effects (RE) increase with increasing evaporation temperature (Tevap) while the compressor power (Wcomp) decreases with increasing Tevap for the constant condensation temperature of 318 K and the evaporation temperature ranging from 248 K to 283 K.

All of the investigated refrigerants have much higher refrigerating effect and isentropic compression work than R134a in fig 8, 17 and as shown in table 3.The variation of the performance coefficients (COP) with evaporating temperatures (Tevap) is illustrated in fig 4. It is found that the coefficient of performance (COP) increases as the evaporation temperature (Tevap) increases for the constant condensation temperature of 318 K and the evaporation temperature ranging from 248 K to 283 K. The performance coefficients (COP) of the alternating refrigerants R152a, R600, R600a and R717 were found to be higher than that of R134a. The power per ton of refrigeration with evaporation (Tevap) were shown in fig 9. The variation in volumetric refrigeration capacity were illustrated in fig 10 in order to verify the advantages of cycle. The cycle performance can be improved by the sub cooling and super heating applications.

The variation of exergetic efficiency with evaporating temperature is illustrated in fig 11. It is found that the exergetic efficiency increases to the optimal temperature and after the optimal temperature it will decreases correspondingly. The exergetic efficiency of the alternating refrigerants R600, R600a, R123 and R717 are much higher than the R134a but exergetic efficiency of alternating refrigerant R152a was slightly higher than the R134a.

4. CONCLUSION

The effect of condenser temperature, evaporator temperature, sub cooling and superheating on the seven refrigerants were deduced. During the course of action, it was found that the evaporator temperatures have considerable effects on Evaporating pressure, Pressure ratio, COP, power per ton of refrigeration, volumetric refrigeration capacity, refrigeration effect, exergetic efficiency of the system. In this analysis, it was found that alternative refrigerants for R134a in order of their data which is given in the above table are R600, R600a, R717, R123, R152a and R290.

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