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EXERGY ANALYSIS OF A VAPOR COMPRESSION HEAT PUMP WORKING WITH R12 ALTERNATIVES

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ABSTRACT

In the present work, exergy analysis method is applied to estimate exergy and exergy loss of each component, total exergy loss, dimensionless exergy loss of each component, and exergetic efficiency of vapor compression heat pump for simultaneous cooling and heating applications over a wide range of operating conditions. Varied parameters include evaporation and condensation temperatures, refrigerant type and compressor speed. Five refrigerants namely; R12, R124, R134a, R152a and R290 are used as working fluids. Evaporation temperature ranging from 0 to 10°C is used to achieve the required water supply temperature for chiller systems. Condensation temperature is varied between 50 and 80°C to cover a wide range of applications such as hot water supply, drying, cleaning ... etc. Compressor speed is changed from 750 to 3000rpm. Results showed that the change in compressor speed yields the highest influence on the refrigerant mass flow rate followed by that of evaporation temperature and then by that of the condensation temperature. This leads to that varying of compressor speed can effectively control exergy output of the heat pump. Effect of the evaporation temperature on exergy loss rate and its dimensionless exergy loss of the compressor and expansion device is essential. The exergy loss rate of the expansion device is more sensitive to the condensation temperature followed by the compressor, liquid-suction heat exchanger, the condenser and the evaporator in that order. However, at higher condensation temperatures, the expansion device and the compressor yield about 47% and 35% of total exergy loss respectively. R290 demands the highest input exergy while R124 needs the lowest required exergy. From high useful exergy point of view, the preferable refrigerants are R290, R152a, R134a, R12 and R124 in that order. At evaporation temperature of 0°C, the maximum exergetic efficiency can be obtained by R152a followed by R12, R124, R134a and finally by R290. However, at evaporation temperature of 10°C, the maximum efficiency value of the VCHP for simultaneous cooling and heating applications is 0.54, 0.53, 0.49, 0.48 and 0.44 for R12, R152a, R124, R134a and R290 respectively. The condensation temperatures ranging from 55°C to 60°C yield the maximum exergetic efficiency for all refrigerants under investigation.

KEY WORDS Heat pump, R12 alternatives, exergy analysis, total exergy loss, exergetic efficiency

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NMENCLATURES

C	Clearance ratio (-)
COP	Coefficient of performance (-)
e	Specific exergy loss (kJ/kg)
E	Exergy loss rate (kW)
GWP	Global warming potential
H	Specific enthalpy (kJ/kg)
I	No of cylinders (-)
m	Mass flow rate (kg/sec)
n	Refrigerant specific heat ratio (-)
N	Compressor speed (rpm)
ODP	Ozone depletion potential
p	Pressure (bar)
P	Power (kW)
s	Specific entropy (kJ/kg K)
T	Temperature (K)
V	Compressor stroke volume (cc)
Q	Heat transfer rate (kW)

GREEK

Δ	Difference (-)
η	Efficiency (-)
ρ	Density (kg/m^3)

SUBSCRIPTS

1-4	Refer to state points in Figs. 1 and 2
∞	Dead state point
c	Cooling
com	Compressor
con	Condensation
d	Desired
eva	Evaporation
ex	Exergetic
exd	Expansion device
hex	Heat exchanger
i	Input
hp	Heat pump
sup	Superheated
l	Loss
n	Needed
o	Output
q	Heat
r	Refrigerant
sin	Sink
sou	Source
st	Stroke
vol	Volumetric
tot	Total
h	Heating

Abbreviations

R12	DichlorodiFluoromethane
R124	2 chloro-1,1,1,2 tetrafluro-ethane
R134a	1,1,1,2-Tetrafluroethane
R152a	1,1-Difluroethane
R290	Propane

INTRODUCTION

In many areas, a cooling as well as heating load exists and the demand of air conditioning is increasing. The amount of energy used to achieve such loads is a substantial portion of the total energy consumed worldwide [1]. In order to reduce this demand, energy efficient cooling and heating system must be adopted. Heat pumps are a natural choice in such areas where there are needs for simultaneous cooling and heating applications as they can improve the overall energy utilization efficiency [2]. Traditional refrigerants namely R12 and R22 are widely employed refrigerants in vapor compression refrigeration / heat pump system. The Montreal Protocol in 1987 called for a freeze in R12 at 1986 consumption level while at the Copenhagen Meeting in 1992, R22 is included in the materials to be phased-out [3] due their environmental problems such as stratospheric ozone depletion and global warming. As a result of this issue, notable research efforts have taken place to find a replacement refrigerant. The subject matter of the present work deals with R12 alternatives in the VCHP.

A number of R12 alternatives for vapor compression heat pump systems have been proposed in the literature based on thermodynamic property data. Identification of such refrigerants for the heat pumps have been reported using energy (First Law) analysis to predict cycle performance. Devotta and Gopichand [4] have comparatively assisted the theoretical performance of R134a along with R22, R134, R124 and R152a as alternatives to R12 using coefficient of performance, pressure ratio and specific compressor displacement. Their results showed that there was a significant scope for the use of R152a both from energy as well as global warming potential points of view. Performance parameters for pure isobutane (R600a), R134a and mixture of propane with isobutane, as alternative refrigerants to R12 for theoretical cycle having a –25°C evaporation temperature and + 55°C condensation temperature were compared by Agrawal [5]. His results indicated that the isobutane offers slightly lower discharge temperature and higher COP but lower volumetric capacity than R134a. Kim et al. [6] have experimentally investigated the performance and heat transfer characteristics of heat pump system working with either single component hydrocarbon refrigerants (propane, isobutane, butane and proplene) or binary mixture of propane/isobutane and propane/butane. Their results showed that R600a or the mixture of R600a with R290 is good candidate as a replacement of R12. Theoretical vapor compression cycles for R12, R22 and propane have been simulated by Charters et al. [7]. They have reported that the heating COP of three refrigerants is almost the same for low evaporation temperature. At high evaporation temperatures, R12 has heating COP slightly higher than those of R22 and propane. Mclaine-Cross and Leondardi [8] have reported that the use of R600a in heat pumps and air conditioners is attractive but requires design changes. Indeed, there is a need to minimize the energy consumption. Exergy (Second Law) analysis is an effective tool to detect where and how much of the input energy of a system is lost. The importance of exergy analysis compared to the energy (traditional or First Law) analysis is mainly due to the fact that the latter does not distinguish between the quality of energy. However, it results in assigning efficiency values greater than units for certain processes like refrigeration and heat pump applications. Moreover, exergy analysis gives information about how losses at different components are interdependent and where a given design should be modified for best performance. It should be noted that exergy analysis can only complement energy analysis (traditional technique) and can not replace it. Energy balance, when used in conjunction with mass balance and other physical principles, helps to define the desired system. Few investigators have been active in the exergy (second law) analysis of heat pumps cycles [9-11]. However, these investigations are limited to certain refrigerants and particular operating conditions.

Hence, exergy analysis of the heat pump running on R12 alternatives as working fluids for simultaneously heating and cooling applications over a wide range of operating conditions is carried out in the present work. The objectives of this are to (a) estimate the exergy loss associated with heat pump cycle and the components that comprise it, and (b) determine the optimum performance (best operating points) of the VCHP. R124, R134a, R152a and R290 as replacements to R12 are used. Table 1 lists the general data for these refrigerants. Varied parameters include evaporation and condensation temperatures and compressor speed. Mainly refrigeration systems for air conditioning are running with a water supply temperature range of 5°C up to 15°C [12]. Thus, evaporation temperature ranging from 0 to 10C is used to achieve the required water supply temperature. Condensation temperature is varied between 50 and 80°C to cover a wide range of applications such as hot water supply, drying, cleaning ... etc.. Compressor speed is changed from 750 to 3000 rpm. Effects of evaporation and condensation temperatures and compressor speed on the refrigerant mass flow rate, input exergy, useful exergy, exergetic efficiency and exergy loss and dimensionless exergy loss of each component are predicted for proposed refrigerants and then compared with those of R12. Based on this comparison, best operating conditions that achieve optimum performance of the VCHP for simultaneous cooling and heating applications are obtained.

Table 1 General data for R12 alternatives.

Abbreviation	R12	R124	R134a	R152a	R290
Formula	CCl_2F_2	$CHClFCF_3$	CH_2FCF_3	CH_3CHF_2	C_3H_8
Molecular Weight (kg/kmol)	120.91	136.48	102.03	66.05	44.10
Melting Point (°C)	-158.0	-199.0	-101.0	-117.0	-188.0
Normal Boiling Point (°C)	-29.8	-11.0	-26.5	-24.7	-42.0
Latent heat at 1.013 bar(kJ/kg)	166.0	163.0	216.0	325.0	430.0
Critical temperature (°C)	112.0	122.5	101.1	113.5	96.7
Critical pressure (bar)	41.2	36.3	40.6	44.9	42.5
Critical Density (kg/m^3)	558.0	554.0	510.0	365.0	220.0
Vapor pressure at 20°C (bar)	5.669	3.27	5.72	5.13	8.38
Gas Constant (J/kg.K)	68.764	60.9	81.49	125.88	188.5
O D P	1.0	0.022	0	0	0
GWP, 100 year	7,300	430	1,200	140	3
Toxicity	No	No	No	No	No
Flammability	No	No	No	Yes	Yes
Specific heat ratio	1.138	1.133	1.12	1.12	1.14

EXERGY CONCEPT AND DEAD STATES

The word "exergy" is the English version of the original Slovenian "egzergija" which is a combination of the Greek "ergon" (work) with Latin " ex." [13]. One of main uses of exergy concept is in an exergy balance in the thermal system analysis. The exergy balance is similar to an energy balance. It has the fundamental difference that, while the energy balance is a statement of the law of energy conservation (First Law of Thermodynamics) the exergy balance may be looked upon as statement of law of energy degradation (Second Law of Thermodynamics).

Exergy is defined as the theoretical maximum (reversible) work available if the energy source is at equilibrium with the environment. Two types of equilibria are unrestricted equilibrium and restricted equilibrium [14]. In unrestricted equilibrium, the conditions of thermal, mechanical and chemical equilibrium between the system and the environment are satisfied. Under these conditions of full thermodynamic equilibrium between system and environment, the system can not undergo any changes of state through any form of interaction with the environment. This is called the dead state. Restricted equilibrium requires the pressure and the temperature of the system and environment to be equal. Hence, under conditions of restricted equilibrium there is, in general, no chemical equilibrium between system and environment. The state of restricted equilibrium with the environment will be refereed to as the environmental state. As there is no chemical potential between the system and the environment, the environmental state is considered in the present work. The specific exergy of a fluid at a given state point in an open system is given by Bejan [15] as:

$$e = (h - h_{\infty}) - T_{\infty}\ (s - s_{\infty}) \tag{1}$$

Where h and s denote the specific enthalpy and entropy respectively and ∞ refers to the environmental state. In the present analysis, the environmental state used has been defined by $T_{\infty} = 288.15$ K, $p_{\infty} = 100$ kPa. Equation (1) neglects the kinetic and the potential energy of the fluid. The specific exergy from Eq. (1) is equal to the net work, which would be obtained or needed if the fluid were to be brought to the environmental state in a continuous reversible process. Exergy flows are, in most general case, mechanical or electrical power (P_i), exergies of fluid flows at the inlet ($m_i.e_i$) and at the outlet ($m_o.e_o$) and exergy of heat (Eq). Because of all irreversible processes consume exergy, the sum of the in-flowing exergries outweighs that of the out-flowing ones. The difference is the exergy loss rate (ΔE_l) of the process. Thus, exergy balance for a steady-state process yields:

$$\Delta E_l = \Sigma\, m_i.e_i - \Sigma\, m_o.e_o - \Sigma\, P_i + \Sigma Eq \tag{2}$$

Often, there is only one fluid flow (i.e. $m_i = m_o = m_{ref}$), one mechanical or electrical power term (P_i) and heat flow term (Q). Thus, equation (2) can be reduced to:

$$\Delta E_l = m_i.e_i - m_o.e_o - P_i + Eq \tag{3}$$

The first two terms on the RHS are the exergy of flow at the input and output, respectively. The third term is the rate of work done, which is positive if it is from the system and vice versa. The last term is the exergy of heat. Heat transfer rate is positive if it is to the system and vice versa. The exergy loss rate ΔE_l is the amount of exergy, which is converted to anergy by irreversible process and lost forever. Equation (3) neglects both potential and kinetic energies.

Exergy analysis of a vapor compression system

The vapor compression system consists essentially of the compressor with drive motor, evaporator, condenser and expansion device. These components are connected together by pipes in which a refrigerant with suitable thermodynamic properties circulates as illustrated in Fig. (1). Liquid – gas heat exchanger is incorporated to improve the cycle performance. The corresponding pressure (p) – enthalpy (h) diagram is shown in Fig.2. The refrigerant is kept at such a pressure in the evaporator such that the evaporation temperature (t_e) is below the temperature of the medium to be cooled. Because of the temperature difference, heat flows into the evaporator and refrigerant evaporates whilst absorbing heat. Then, it flows through the Liquid – gas heat exchange. The resulting super-heated vapor (state 1'), at low temperature and pressure, is drawn off by the compressor and compressed to high pressure and temperature (state 2) such that the condensing temperature (t_c) at this pressure is above the temperature of the medium to be heated. Due to the temperature difference, heat is extracted from the condenser and the entire refrigerant vapor condenses whilst discharging heat. The sub-cooled refrigerant liquid (state $3'$) is then expanded in an expansion device to the low evaporating pressure (state 4) and can thus absorb heat again in the evaporator thereby the cycle is completed.

For exergy analysis of the VCHP, a number of assumptions are made. They are: (a) no pressure drop, (b) compressor motor efficiency was equal to 1.00, (c) saturated liquid refrigerant leaving the condenser, (d) saturated vapor refrigerant leaving the evaporator, (e) degree of superheating was taken equal to 5°C (i.e. Δt_{sup}= 5°C), (f) no heat loss in both the expansion device and the compressor, and (g) constant source and sink temperatures.

The principal processes involved in the vapor compression cycles are compression, throttling and heat transfer. Exergy loss during each process has to be reduced to improve the thermodynamic efficiency of the system. Thus, each component of the VCHP shown in Figure 1 is modeled through a set of parameters, which characterize the mass, energy and exergy balance equations. In the following, exergy loss rate of each component is given. All subscripts refer to state points in Figures 1 and 2.

After the evaporation process, the refrigerant is compressed in an irreversible adiabatic process from 1 to 2 and equation (3) can be written as;

$$\Delta E_{l,com} = m_{ref}\,(e_1' - e_2) + P_i \tag{4}$$

The input power (P_i) to the adiabatic compression process is given by;

$$P_i = m_{ref}\,(h_2 - h_1') \tag{5}$$

The refrigerant mass flow rate (m_{ref}) is determined as;

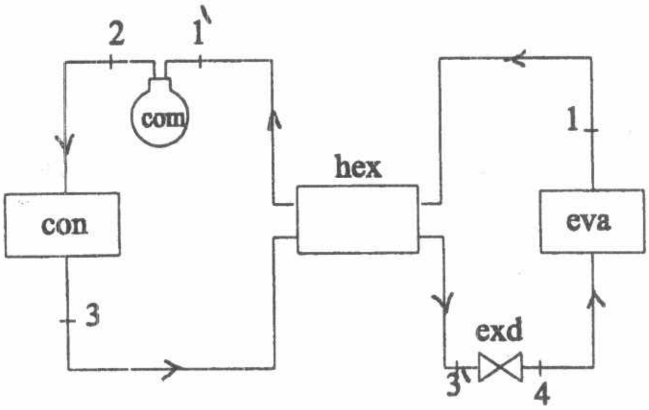

Fig. 1 Schematic diagram of a single stage VCHP.

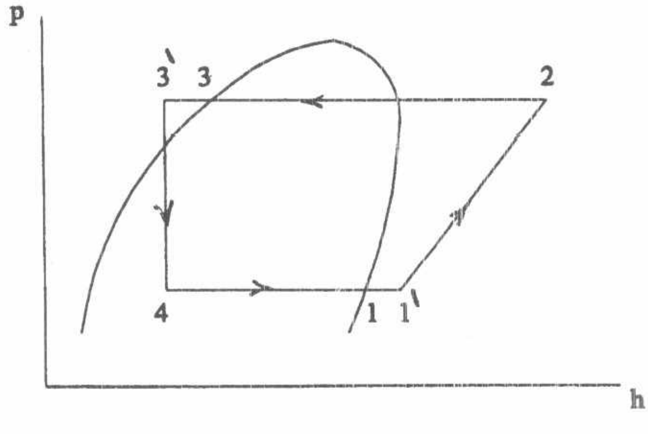

Fig. 2 Pressure-enthalpy diagram of a single stage VCHP.

$$m_{ref} = V_{st} \,.\, N \,.\, i \,.\, \rho_1^{\backslash}\, \eta_{vol} \tag{6}$$

Where V_{st} is the compressor swept volume (74.5 cc), N is the compressor speed, i is the number of cylinders (i=1), $\rho_1^{\backslash}$ is the refrigerant vapor density at the compressor inlet and η_{vol} is the compressor volumetric efficiency which is estimated as;

$$\eta_{vol} = 1 - C[\, (p_{con}/p_{eva})^{1/n} - 1)] \tag{7}$$

C is the clearance ratio (C= Vc/Vst = 0.04), p_{con} and p_{eva} are the condensation and the evaporator pressures respectively and n is the specific heat ratio of the refrigerant under investigation.
After the compression process, the refrigerant is de-superheated, and then condensed along line 2-3. The condenser exergy loss rate from 2 to 3 is expressed as;

$$\Delta E_{l,con} = m_{ref}\,(e_2 - e_3) + E_{con} \tag{8}$$

Where E_{con} is the condenser exergy given by equation (9)

$$E_{con} = -\, Q_{con}\,(1\text{-}T_{\infty}/T_{sin}) \tag{9}$$

The heat sink temperature (T_{sin}) depends on the type of system which being considered. It puts equal to the temperature at which the heat is absorbed by the fluid to be heated. It is assumed that this temperature is lower than the condensation temperature by 5°C, i.e. $T_{sin} = T_{con} - 5$. $(1\text{-}T_{\infty}/T_{sin})$ term is defined as the Carnot factor. Q_{con} is the condenser heat load that can be estimated as;

$$Q_{con} = m_{ref}\,(h_2 - h_3) \tag{10}$$

From 3 to 4, the refrigerant is throttled adiabatically thereby with no input power in Eq. (3), the exergy balance for steady flow throttling process yields

$$\Delta E_{l,exd} = m_{ref}\,(e_3 - e_4) \tag{11}$$

During evaporation process, the refrigerant absorbs heat Q_{eva} from 4 to 1 and the evaporator exergy loss can be expressed as;

$$\Delta E_{l,eva} = = m_{ref}\,(e_4 - e_1) + E_{eva} \tag{12}$$

Where E_{eva} is the evaporator exergy that can be computed as;

$$E_{eva} = Q_{eva}(1\text{-}T_{\infty}/T_{sou}) \tag{13}$$

The heat source temperature (T_{sou}) corresponds to the temperature of refrigerated substance. It is taken as 5°C higher than the evaporation temperature, i.e$T_{sou}=T_{eva}+5$. The evaporator heat load (Q_{eva}) is given as;

$$Q_{eva} = m_{ref}\,(h_1 - h_4) \tag{14}$$

The exergy loss rate in the liquid – gas heat exchanger is calculated by

$$\Delta E_{l,hx} = = m_{ref}\,[(e_1 - e_1^{\backslash}) + (e_3 - e_3^{\backslash})] \tag{15}$$

The exergy losses in the pipelines are ignored, as they are comparatively small [16]. The total exergy loss rates of the heat pump are the sum of the exergy loss in each component, i.e.

$$\Delta E_{l,tot} = \Delta E_{l,com} + \Delta E_{l,con} + \Delta E_{l,exd} + \Delta E_{l,eva} + \Delta E_{l,hx} \quad (16)$$

The exergetic efficiency is a very useful measure for the thermodynamic quality of a technical process. The general basis for the exergetic efficiency is given by

$$\eta_{ex} = E_d/E_n \quad (17)$$

Where E_d is the desired exergy of the process and E_n is the total amount of needed exergy, i.e. that required for the desired effect. Thus, the exergetic efficiency of the vapor compression heat pump for simultaneously heating and cooling applications can be expressed either by Eq. (18) or Eq.(19).

$$\eta_{ex\text{-}hp,\ c+h} = (E_{eva}+E_{con})/E_{com} \quad (18)$$

or

$$\eta_{ex\text{-}hp,\ c+h} = (E_{com} - \Delta E_{l,tot})/E_{com} \quad (19)$$

Where

$$E_{com} = P_i \quad (20)$$

In order to evaluate the performance of possible alternative refrigerants in compression system applications, knowledge of their thermodynamic properties is required. In the present work, published data by ASHRAE [17] for all the refrigerants under consideration are used.

A computer program based on Equations (1) to (20) is developed. The aim of this program is to comparatively assess the performance of the VCHP working with R124 along with R134a, R152a and R290 as alternatives to R12. Thermodynamic properties of the refrigerants are presented in subroutines. The input data for simulation are evaporation and condensation temperatures, degree of superheating (Δt_{sup}), compressor speed, refrigerant type and its specific heat ratio (n) and compressor dimensions (Vst = 74.5 cc, C=0.04, i=1). The program calculates heat load, exergy and exergy loss for each component, total exergy loss, exergetic efficiency as well as state point data such as thermodynamic properties and rates of mass flow. The results are shown in Figs. 3-11.

RESULTS AND DISCUSSION

A computer program was developed to predict the performance of the vapor compression cycle for the followings:

- Five refrigerants namely; R12, R124, R134a, R152a and R290,
- Evaporation temperature from 0 to 10°C which covers the required water supply temperature in chiller application,
- Condensation temperature lies between 50 and 80°C which is suitable for various applications such as hot water supply, drying, cleaning ... etc., and
- Compressor speed ranges from 750 to 3000 rpm.

At each state point of the system, mass flow rate, enthalpy, entropy, exergy of the working fluids are determined for different working conditions. For the condensation and evaporation temperatures of 50 and 0°C respectively, the values of enthalpy, entropy, and exergy at each state point of R134a VCHP system are given in Table 2.

Variation of refrigerant mass flow rate versus the condensation temperature for different compressor speeds and evaporation temperatures is illustrated in Fig. 3. When the condensation temperature increases, the condensation pressure increases causing to decrease the volumetric efficiency (Eq.7). This reasons the refrigerant mass flow rate to reduce at higher condensation temperature as shown in Fig. 3. Clearly, the refrigerant mass flow rate increases with the evaporation temperature and the compressor speed. With reference to Eq. (6), when the evaporation temperature increases the vapor refrigerant density ($\rho 1$) at the compressor inlet increases. This leads to increase the refrigerant mass flow rate. It can be seen that for a given set of evaporation (heat source) and condensation (heat sink) temperatures, the compressor speed has more influence on the refrigerant mass flow rate. To make full advantage of that, varying compressor speed can effectively control the heat pump output. When the heat demand falls, the compressor can be made to run at lower speed, this improves the system reliability and reduces exergy loss rate comparatively.

Specific exergy loss of each component as a function of the condensation temperature is shown in Fig. 4. As the condensation temperature increases, the specific exergy loss of the compressor, the expansion device, the liquid-suction heat exchanger increases and that of the condenser and the evaporator decreases. It can be seen that the largest specific exergy loss exhibited by the expansion device and the compressor, however, at lower and higher condensation temperature, the specific exergy loss due to compressor and expansion device respectively are significant. This is because the rate of increase in specific entropy at both compressor (s_2) and expansion device (s_4) exits is higher than that at their inlets. Clearly, specific exergy loss of the expansion device is more sensitive to the condensation temperature. It is clear that influence of evaporation temperature on the specific exergy loss at the compressor, the expansion device and the liquid-suction heat exchanger is essential. However, effect of the evaporation temperature on both the compressor and the liquid-suction heat exchanger is superior to that of the condensation temperature while the effect of the condensation temperature on the expansion device loss is superior to that of evaporation temperature. Clearly, the influence of both evaporation and condensation temperatures on specific exergy loss at the condenser and the evaporator is not essential.

Change of exergy loss rate of each component of VCHP against the condensation temperature is presented in Fig. 5. The exergy loss rate can be obtained by multiplying the refrigerant mass flow rate (Fig. 3) into the specific exergy loss (Fig. 4). As the condensation temperature increases, the exergy loss rate of the compressor, the expansion device and liquid-gas heat exchanger increases while that of the condenser and the evaporator decreases. Losses in the condenser are significant while those in the evaporator remain nearly constant. This behavior can be attributed to combined effect of both specific exergy loss and the refrigerant mass flow rate. It is clear that the expansion device exergy loss rate is more sensitive to the condensation temperature followed by the compressor, liquid-suction heat exchanger, the condenser and the evaporator in that order.

Dimensionless exergy loss of each component of the VCHP is shown in Fig. 6. The dimensionless exergy loss of each component is defined as the ratio of its exergy loss to the total exergy loss of the VCHP. At high condensation temperature (80°C),

Table 2 Vapor compression heat pump data obtained from the analysis

State point	t (°C)	p (bar)	h (kJ/kg)	s (kJ/kg K)	e (kJ/kg)
1′	05.0	0.296	403.5	1.748	-99.8
2	72.4	1.306	449.5	1.784	-64.3
3	50.0	1.306	274.5	1.247	-84.7
3′	47.8	1.306	271.0	1.237	-85.2
4	00.0	0.296	271.0	1.260	-91.7
1	00.0	0.296	400.0	1.728	-97.7

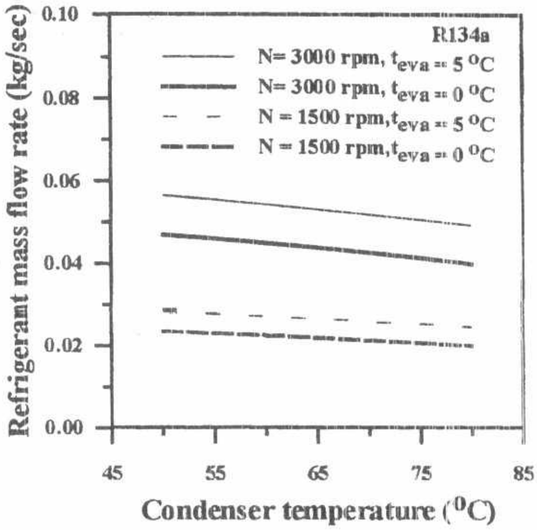

Fig.3 Effect of condensation temperature On refrigerant mass flow rate

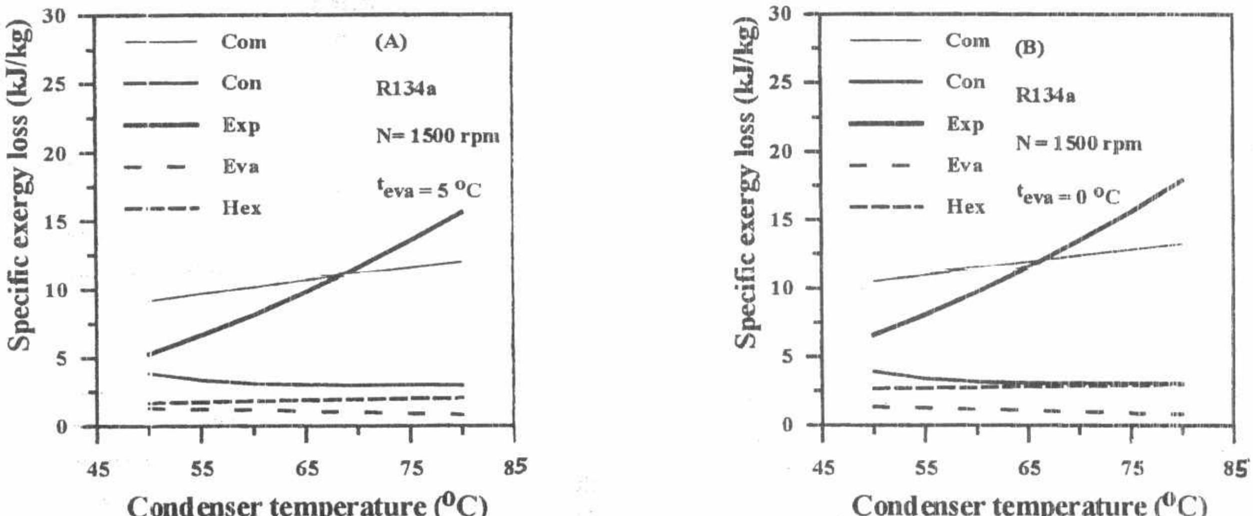

Fig. 4 Effect of condensation temperature on exergy loss rate of each component of the VCHP. A at T_{eva}=5°C and B at T_{eva}=0°C.

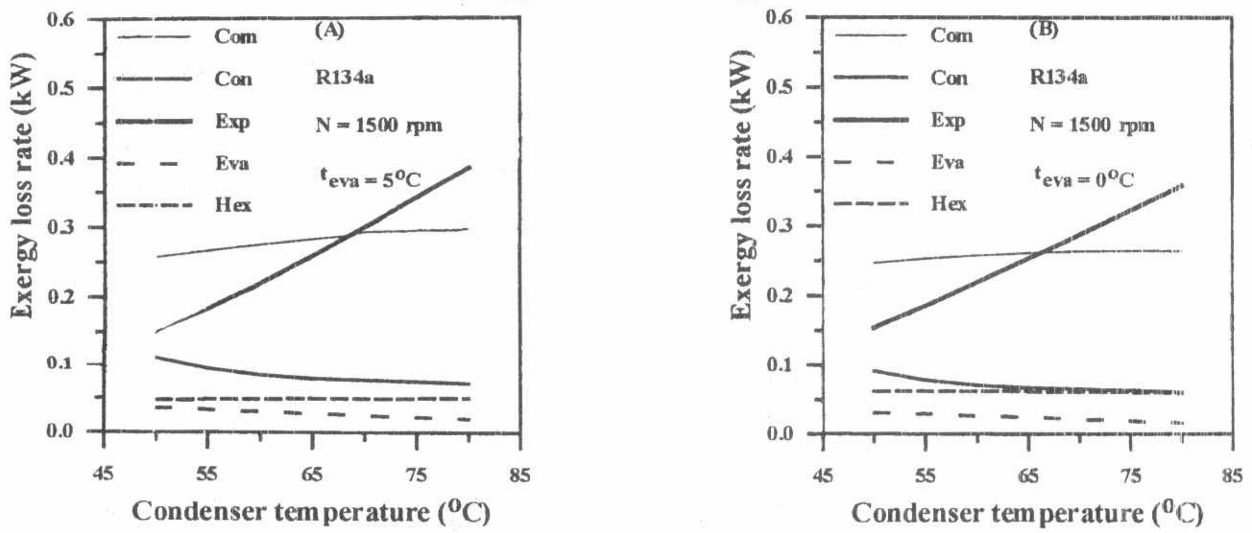

Fig.5 Effect of condensation temperature on specific exergy loss of each component of the VCHP. **A** at T_{eva}=5°C and **B** at T_{eva}=0°C.

at least 47% and 35% of the total exergy loss of the VCHP are observed in the expansion device and the compressor respectively within the operating range shown in Fig. 6. Clearly, the evaporation temperature has only little impact on the dimensionless of each component. However, average values of the dimensionless exergy loss are 40%, 37.8, for the compressor and the expansion device respectively.

Variation of exergetic efficiency and the exergy of the compressor, the condenser and the evaporator versus the condensation temperature is shown in Fig. 7. Whereas the evaporator exergy is insensitive to the condensation temperature, the exergy of the compressor and the condenser increases with the condensation temperature. The increase in the compressor exergy is mainly attributed to the increase in the compressor specific work while that in the condenser is mainly due to the increase in Carnot factor $[1-(T_{\infty}/T_{sin})]$ with the condensation temperature. It is shown earlier that as the condensation temperature increases, the refrigerant mass flow rate decreases thereby the cooling capacity decreases causing the evaporator exergy to reduce slightly as shown in Fig. 7. Clearly, the exergetic efficiency increases then decreases as condensation temperature increases. This trend is due to the combined effect of desired (condenser and evaporator) exergy and the input (compressor) exergy. It is observed that change of the compressor exergy with condensation temperature is started faster than that of the condenser and after certain temperature the trend is reversed. This figure reveals that the condensation temperature lies between 55 and 60C yield the maximum exergetic efficiency. However, the influence of the evaporation temperature on the exergetic efficiency is superior to that of the condensation temperature.

Comparison between the refrigerants

Figure 8 shows a comparison of required exergy variation for the considered refrigerants. Lowest required exergy can be obtained by R124 over the entire range of varied parameters. The next best refrigerant from low required exergy point of view is R152a. R290 demands the highest input exergy. This is attributed to high change in the specific compressor work (h2-h1) for R290 followed by R152a, R134a, R124 and R12 as given in Table 3. Also, the curve for R290 is steeper indicating that this refrigerant is more sensitive to the condensation temperature than other refrigerants.

It is clear from a comparison of total exergy loss in Fig. 9 that R124 and R290 yield the lowest and highest total exergy loss rate respectively. Also, a significant advantage of R152a compared to R12 is demonstrated. However total exergy loss of R134a is slightly higher than that of R12. This is because of variation of specific enthalpy and entropy as listed in Table 3. It is evident from Fig. 10 that high useful exergy at both evaporator and condenser can be obtained by R290, R152a, R134a, R12 and R124 in that order. The trend of useful exergy is due to combined effect of required exergy (Fig. 8) and total exergy loss (Fig. 9).

Exergetic efficiency of R12 replacements in VCHP for cooling and heating application is compared in Fig 11. Among the refrigerant under investigation, R152a yields the highest exergetic efficiency at lower evaporation temperature. High exergetic efficiency can be obtained by R12 followed by R152a, R124, R134a and finally by R290 at higher evaporation temperature. This trend is expected from the variation of required exergy and useful exergy in Fig 8 and 10 respectively. This figure reveals that the maximum exergetic efficiency can be obtained when the condensation temperature lies between 55 and 60°C.

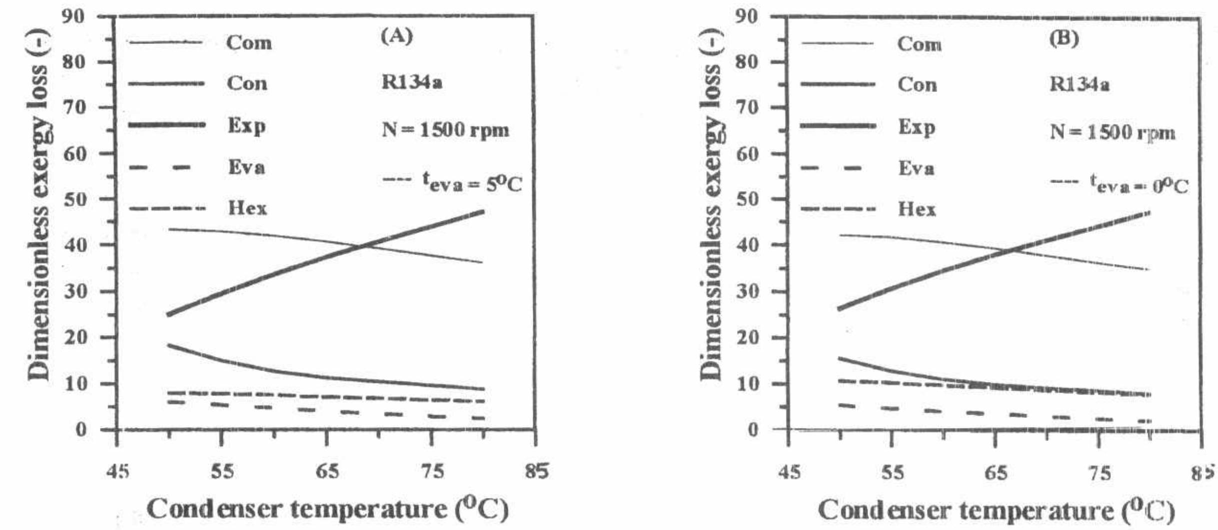

Fig.6 Variation of the dimensionless exergy loss of the main components of the VCHP.
A at T_{eva}=5°C and B at T_{eva}=0°C.

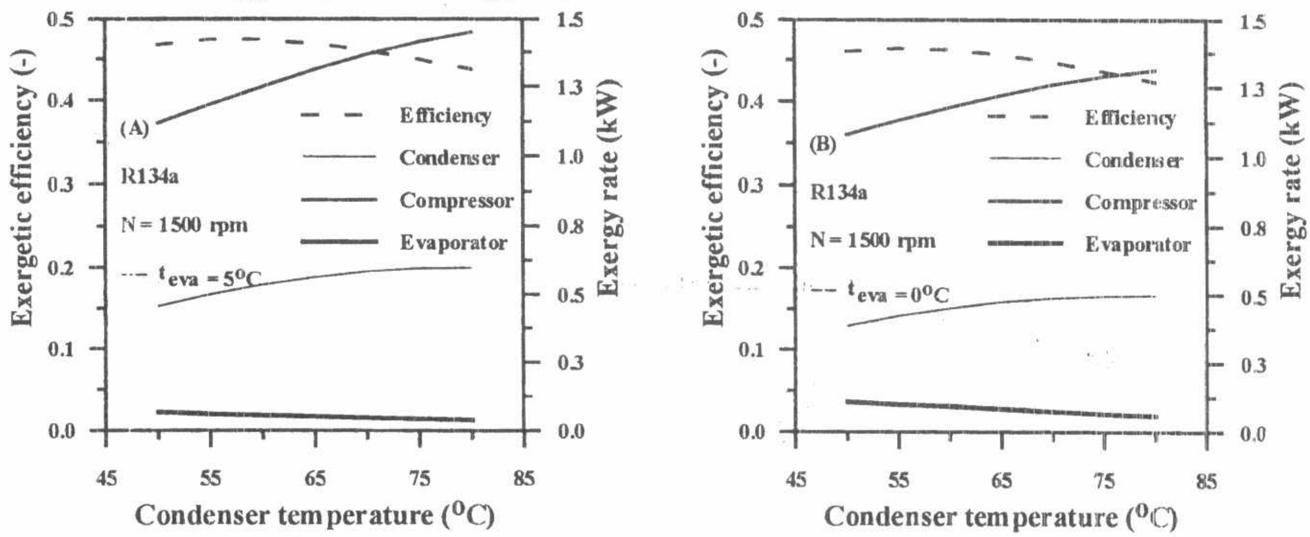

Fig.7 Variation of performance characteristics of the VCHP with the condenser temperature.
A at T_{eva}=5°C and B at T_{eva}=0°C.

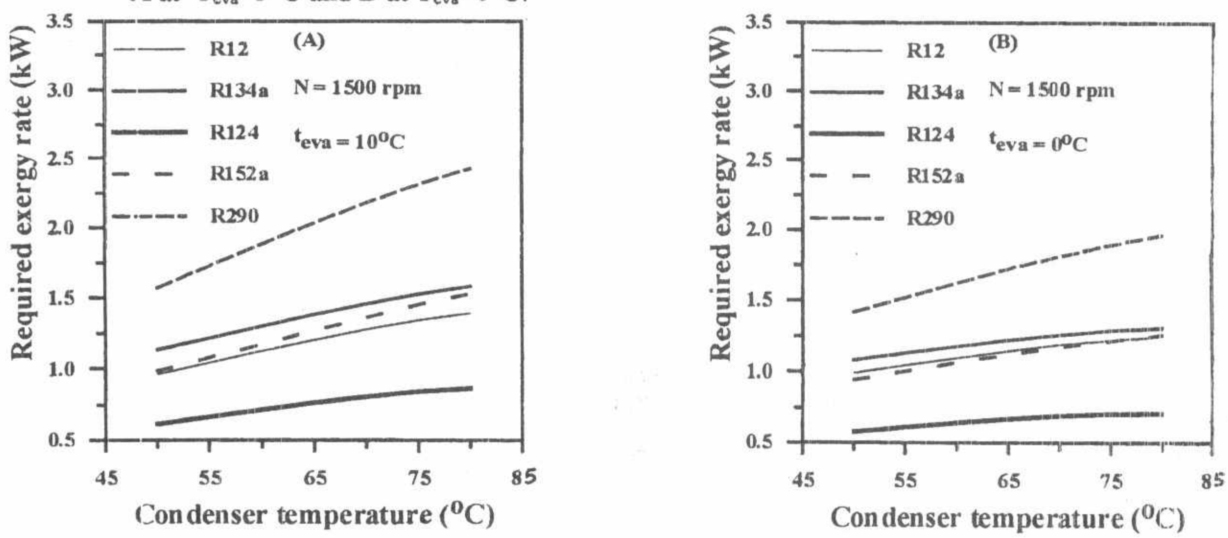

Fig.8 Comparison of required exergy variation for various refrigerants.
A at T_{eva}=10°C and B at $_{eva}$=0°C.

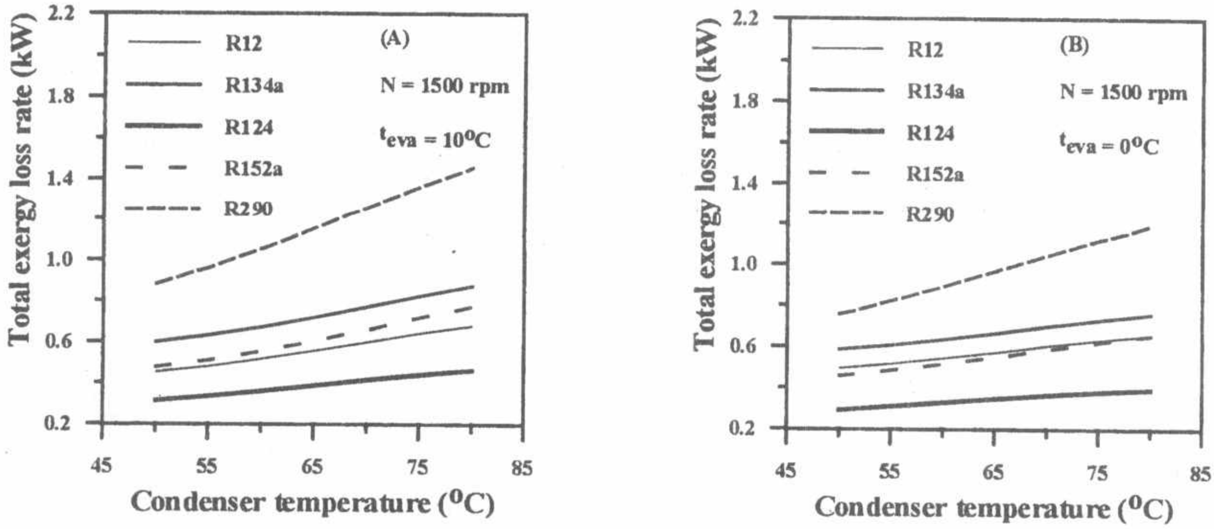

Fig.9 Comparison of total exergy loss rate variation for various refrigerants. A at T_{eva}=10°C and B at $_{eva}$=0°C.

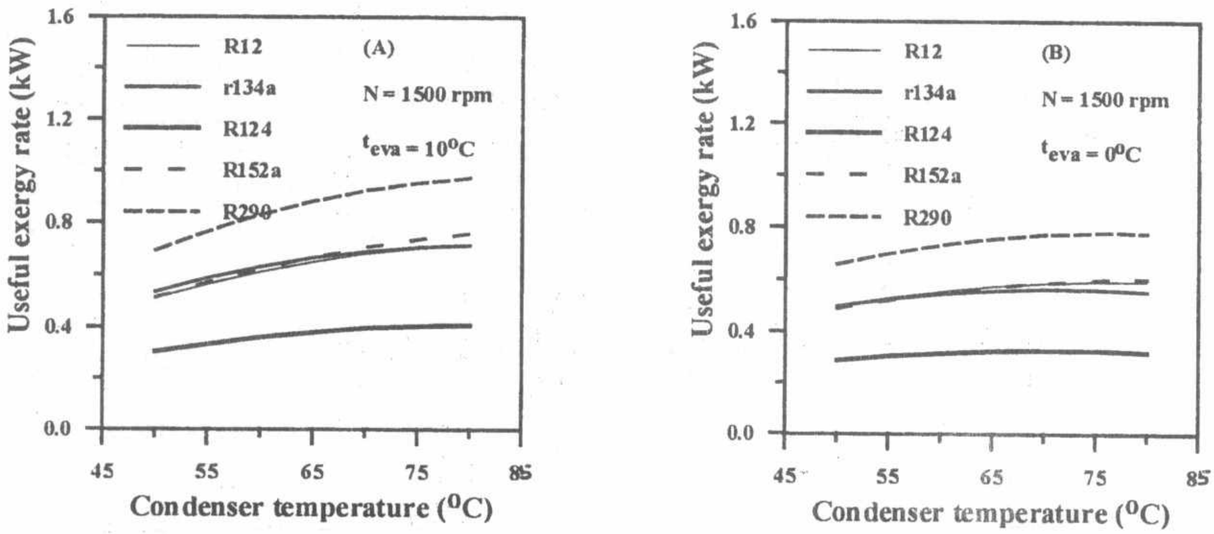

Fig.10 Comparison of useful exergy rate variation for various refrigerants. A at T_{eva}=10°C and B at $_{eva}$=0°C.

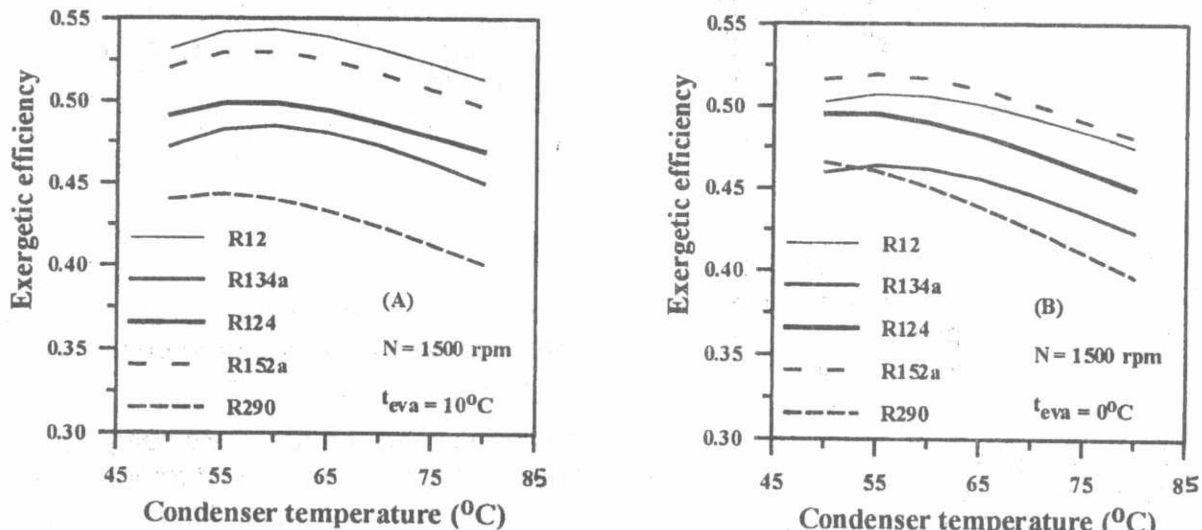

Fig.11 Comparison of useful exergy rate variation for various refrigerants. A at T_{eva}=10°C and B at $_{eva}$=0°C.

Table 3 Variation of mass flow rate and specific exergy loss of each component for various refrigerants at evaporation temperature of 0°C.

Refrigerant	t_c (°C)	m_{ref} (kg/sec)	h_2-h_1 (kJ/kg)	Δe_{com} (kJ/kg)	Δe_{con} (kJ/kg)	Δe_{exd} (kJ/kg)	Δe_{eva} (kJ/kg)	Δe_{hex} (kJ/kg)	Δe_{tot} (kJ/kg)
R134a	50	2.35E-02	46.01294	10.49239	3.861244	6.553173	1.320112	2.636959	24.86388
	55	2.30E-02	49.09424	10.96243	3.382572	8.052773	1.237018	2.694809	26.32961
	60	2.25E-02	52.34274	11.44034	3.11933	9.706318	1.152895	2.751183	28.17006
	65	2.19E-02	55.70145	11.9156	3.020645	11.51384	1.067703	2.806183	30.32397
	70	2.13E-02	59.09143	12.37579	3.022595	13.47528	0.981444	2.859776	32.71489
R12	50	2.98E-02	33.18835	7.407841	3.649853	4.279278	0.539132	0.636212	16.51232
	55	2.93E-02	35.58939	7.776226	3.282613	5.296097	0.509811	0.682114	17.54686
	60	2.88E-02	38.06754	8.137505	3.043323	6.42065	0.480087	0.726746	18.80831
	65	2.82E-02	40.57202	8.482372	2.890706	7.652939	0.449939	0.770176	20.24613
	70	2.76E-02	43.03598	8.79998	2.774302	8.992893	0.419405	0.812473	21.79906
R124	50	1.66E-02	34.62082	8.082436	7.461609	1.515015	8.17E-03	0.415489	17.48272
	55	1.63E-02	37.36923	8.604149	7.096756	2.647636	0.007600	0.509045	18.86527
	60	1.59E-02	40.18872	9.122738	6.786638	3.967506	0.007180	0.599579	20.48364
	65	1.55E-02	43.0105	9.624435	6.479821	5.474487	0.006700	0.687126	22.27257
	70	1.51E-02	45.74353	10.09238	6.112784	7.168407	0.006170	0.771961	24.1517
R152a	50	1.40E-02	67.60931	14.53601	4.945814	7.88736	2.029434	3.31097	32.70959
	55	1.37E-02	73.25317	15.39336	4.727142	9.755139	1.934479	3.393368	35.20348
	60	1.34E-02	79.29053	16.28847	4.888756	11.82029	1.838379	3.473946	38.30984
	65	1.31E-02	85.64508	17.21036	5.353796	14.08276	1.741146	3.552601	41.94066
	70	1.27E-02	92.20587	18.14557	6.023581	16.54249	1.642783	3.629333	45.98376
R290	50	1.676E-02	84.36108	19.20245	1.213268	18.26075	1.987491	4.423714	45.08768
	55	1.65E-02	91.97931	20.55837	1.715193	21.07858	1.854577	4.441944	49.64866
	60	1.63E-02	99.83099	21.90962	2.753725	23.9915	1.719341	4.452793	54.82698
	65	1.60E-02	107.7857	23.23224	4.244699	26.97964	1.581797	4.45681	60.49518
	70	1.57E-02	115.6808	24.49958	6.086774	30.02309	1.441958	4.454201	66.50561

CONCLUSIONS

Simulation model is developed to perform the exergy analysis of a vapor compression heat pump working with various refrigerants for simultaneous heating and cooling applications. Heat load, exergy and exergy losses for each component, total exergy loss, exergetic efficiency are estimated for each refrigerant.

R124 along with R134a, R152a and R290 as alternatives to R12 are compared over a wide range of varied parameters such as compressor speed and temperatures of evaporation and condensation. Results showed that exergy loss rates of the expansion device and the compressor are significant. Average values of the dimensionless exergy loss are about 40% and 38% for the compressor and expansion device respectively. Thus, in order to improve the performance, such losses should be reduced. Condensation temperature ranging from 55 to 60°C yields the maximum exergetic efficiency for all refrigerants under consideration. Results indicated that R152a is the most suitable refrigerant to replace R12 from viewpoints of energy saving and environmental problems. Unfortunately, R152a is a flammable refrigerant. Thus, R134a is a good refrigerant to substitute R12 from ozone depletion, safety and energy saving points of view.

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