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THERMODYNAMIC ANALYSIS OF A SIMPLE REFRIGERATION CYCLE USING HYDROCARBON REFRIGERANTS AS SUBSTITUTE TO R22

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ABSTRACT

In this paper, first law and second law thermodynamic analysis for hydrocarbon refrigerants are performed. Hydrocarbons, R290, R1270 and a mixture of R290/R1270 of 15/85 wt % are used as drop-in refrigerant to R22. The refrigeration system consists of a simple direct expansion refrigeration cycle with an adiabatic compressor having an isentropic efficiency of 85%. The suggested cooling load is 1 kW. The evaporator temperature is 249, 253, 257, 261, 265, 269, 273 and 277 K for a condenser temperature of 313 K, then the condenser temperature is varied between 308, 313, 318, 323, 328 and 333 K for an evaporator temperature of 263 K. Thermodynamic properties of all refrigerants are compared. Hydrocarbons showed a specific cooling capacity that is 68%-77% higher than that of R22 with 39.7%-43.5% lower refrigerant mass flow rate. R22 showed a cycle COP and a second law efficiency that are 2% and 2-4% higher than that of hydrocarbon refrigerants respectively.

Key Words: vehicular emissions , emission control regulation, combustion engine

Nomenclature

COP	Coefficient of performance
GWP	Global warming potential
h	Specific enthalpy [kJ/kg]
h_{fg}	Latent heat [kJ/kg]
HC	Hydrocarbon
I	Irreversibility [kJ/kg]
LPG	Liquefied petroleum gas

$\dot{m}$	Mass flow rate [kg/s]
Q	Specific load [kJ/kg]
ODP	Ozone depletion potential
T	Temperature [K]
s	Specific entropy [kJ/kg.K]
VRC	Volumetric refrigeration capacity (kJ/m^3)
W	Specific work [kJ/kg]
$\dot{W}$	Power consumption [kW]

Greek letters

Ψ	Availability [kJ/kg]
η	Efficiency [%]

Subscripts

0	Ambient conditions
1,2,…,6	States along the cycle, as shown in figure (5)
C	Cycle
is	Isentropic
ref	Refrigerant
cond	Condenser
comp	Compressor
evap	Evaporator
exerg	Exergetic
exp	Expansion valve

1. Introduction

Since the 19th century, natural refrigerants like water,CO_2,$NH_3$and SO_2areused. The main disadvantages of these refrigerants are their being toxic, reactive and/or flammable.At the beginning of the 20th century, Chlorofluorocarbons (CFCs), Hydrochlorofluorocarbons(HCFCs) and Hydroflourocarbons(HFC)are introduced. They are non toxic, non flammable, non reactive and provided better thermodynamic properties. In 1970's, most of these refrigerants are banned after Kyoto Protocol and Montreal Protocol [1,2] being signed due to their high Ozone Depletion Potential (ODP) and/or their high Global Warming Potential (GWP).Finding drop-in refrigerants and retrofitting old cycles with new refrigerants is the scope of many researches nowadays. Hydrocarbon(HC) refrigerants solved this main problem by having zero ODP and very low GWP as well as better thermodynamic properties, but their main disadvantage is the high flammability. This study is mainly concerned about investigating the possibility of using HC refrigerants as substitute to R22 which was decided to be phase out by the year 2030 due to its higher values of ODP and GWP. In this work first law and the second law analysis on a simple refrigeration cycle is done for R22, R290, R1270 and a mixture of R290/R1270.

2. Literature review

In this section, the previous work was surveyed; the analytical, experimentalstudies performed to assess the possibility of replacing various CFCs, HFCs by HCs.Starting with the theoretical and analytical work, Spatz and Mott [3] performed a thermodynamic analysis to compare the system performance parameters using a validated system model to replace R22 by three proposed alternatives; R404A, R410A and R290. R410A resulted in the lowest Life Cycle Climate Performance (LCCP) amongst the used refrigerants while; R290 nearly matched that of R410A with slightly better system efficiency than R22. Fatouh and El Kafafy [4] proposed to replace R134a by alternative HCs; R290, R600 and R290/R600/R600a mixtures with various mass fractions. The R290/R600/R600a mixtures containing 60% mass fraction of R290 yields to a 2.3% higher COP, 50% reduction in the refrigerant mass flow rate with nearly the same input compressor power. Arcaklioglu et al. [5] compared the COP of vapor compression cycle using different HC mixtures as alternatives of R12, R22 and R502. The results showed that the most appropriate drop in mixture for R12, R22 and R502 are R290/R600a having mass ratio of 56/44, R32/R125/R134a with mass ratio of 32.5/5/62.5, and R32/R125/R134a with mass ratio of 43/5/52 provided0.4%, 0.8% and 2% higher COP, respectively. Dalkilic and Wongwises[6] tested the theoretical performance of R290, R600a and R1270 mixtures in order to find a suitable alternative for R12 and R22. The refrigerant mixtures R290/R600a (40/60 by wt %) and R290/R1270 (20/80 by wt %) are found to be the best alternatives for R12 and R22 respectively. El Morsi and Hamza [7] concentrated their work on the optimization of a HC gas blends that can replace R12, R22 and R134a in order to maximize the cycle coefficient of performance(COP) and volumetric refrigeration capacity(VRC). The optimum substitutes of R12, R22 and R134a were found to be a mixture of R290/R600a having 89/11 wt %, R1270/R290 having 85/15 wt % and R290/R600a having 75/25 wt % respectively. El Morsi [8] used R290, R600 and(Liquefied Petroleum Gas) LPG in a simple refrigeration cycle with a Liquid Suction Heat Exchanger (LSHX) as an alternative to R134a. R600 showed the highest cycle COP, exergetic efficiency and second law efficiency while LPG showed the lowest values of the same parameters. Yan et al. [9] proposed the use of R290/R600 and R290/R600a mixtures as working fluid in an internal auto-cascade cycle (IARC) in a domestic refrigerator-freezer cycle and compared it to the performance of the conventional refrigeration cycle (CRC). The mixture of R290/R600a showed a better performance than that of R290/R600 in all aspects. The mixture of R290/R600a that consists of 56/44 wt % had the highest cycle COP. Hallak et al. [10] evaluatedthe performance of a vapor injection refrigeration system that employs various mixtures of R290/R600a. The maximum COP was obtained by the R290/R600a, 40/60 wt % mixture. Sagia and Rakapoulos [11] suggested a theoretical analysis for an office using seven binary and ternary mixtures of R32, R134a, R125 and R152a as alternatives to R22. The COP of R22 was found to be the highest among all proposed refrigerants. The ternary mixture of R32/R125/R134a consisting of 10/70/20 wt %, showed the closest value of COP compared to the other mixtures. Sarbu and Sebarchievici [12] analyzed a single stage vapor compression cycle using sixteen different refrigerants and compared the evaporator pressure, the condenser pressure, the cooling load and the cycle COP. Refrigerants R22 and R290 showed nearly the same values of the operating parameters.

Also, experimental works were conducted to demonstrate the credibility and prove the validity of the analytical studies that apply the thermodynamic analysis on refrigeration cycles with various refrigerants and refrigerant mixtures.

Akash and Said [13] used LPG as a substitute to R12in a domestic dual compartment refrigerator/freezer unit with gross capacity of 240 liter. LPG, consisting of 30/55/15 of R290/R600/R600a by wt % with different charge masses, was used in this experimental work. The evaporator and condenser temperatures were kept at -15°C and 47°C, respectively. The 80 g charge gave the best results and the cooling capacity is equal to 3-4 times that of R12.

Sekhar et al. [14] carried out their experimental work to retrofit a 165 L domestic refrigerator using CFC refrigerant with a mixture of R134a/R290/R600a. The system showed its compatibility with mineral oil and 114 g charge mass was identified as a promising alternative for R12 as the refrigerator consumed 4.8-6.4% less energy with a 3-8% higher COP when operating at -15°C. Wongwises and Chimres[15] investigated a 239 L domestic refrigerator charged with eight different pure refrigerants, binary and ternary mixtures of R290, R600, R600a to be used as an alternative to R134a. The R290/R600 (40/60 wt %) required less energy per day than R290/R600a with the same mass ratio. The ternary mixture R290/R600/R600a (40/30/30 wt %) had the least energy consumption, but none of the eight mixtures showed a close temperature-pressure characteristics to that of R134a.Mani and Selladurai[16] aimed to find a drop-in refrigerant of mixture of R290/R600a of 68/32 wt % instead of using R12 and R134a. The obtained experimental results showed that the refrigeration capacity of R290/R600a was higher than R12 by 19.9-50.1% atthe lower evaporator temperatures and higher by 21.2-28.5% at the higher evaporator temperatures. The COP of R290/R600a mixture was also higher by 3.9-25.1% than R12 atthe lower evaporator temperatures and by 11.8-17.6% atthe higher evaporator temperatures.Ching-Song et al. [17] replaced R134a in a 440 L refrigerator by a mixture of R290/R600a, 50/50 wt %. The total consumed energy saved was4.4% and the refrigerant mass was reduced by 40%. This mixture has over twice better refrigeration effect than R12 and near double the refrigeration capacity of R134a, but it required a larger compressor.Mohanraj et al. [18] conducted an experimental investigation to replace R134a in a 200 L domestic refrigerator by a HCrefrigerant mixture of R290/R600a (45.2/54.8 by wt %). The results showed that the HC refrigerant mixture consumed less energy, lower pull down time and ON time ratio, and a higher COP. Pull down time is the time required for the system to reach the set point while, ON time ratio is defined as the ratio between the time the system requires to return to the set point to the total time of an ON-OFF cycle. Rasti et al. [19] aimed to find a substitute refrigerant of R134a by a mixture of R290/R600a consisting of 56/44 wt % in a 238 L domestic refrigerator. The ON time and the total energy consumption is reduced by 13% and 5.3%, respectively. Using a charge of 55 g and yielded to a minimum energy consumption and ON time ratio while, the condenser and the capillary tube were not sufficient. Yoon et al. [20] investigated a 740 liter domestic refrigerator using R600a as a working fluid. The capacities of the freezer and the refrigerator compressor in the dual-loop cycle were reduced by 20% and 72%, respectively, while requiring an increase in the capillary tube length to compensate for the compressor size decrease. Consequently, the evaporator area of the refrigerator evaporator was increased by 77%.Yu and Teng [21] tested the performance of a domestic refrigerator running with R134a using mass ratios of R290/R600a of 65/35, 50/50 and 0/100 with different capillary tube lengths and charge masses. The energy factor (EF) of the HC mixtures was found to be higher than that using R134a by 9.1%, 12.2% and 42.3% for the mixtures 65/35, 50/50 and 0/100, respectively. Li et al. [22] compared the performance of a secondary loop system using R152a and R290 with the performance of a direct expansion system using R134a; a baseline system for an automotive air conditioning system. The refrigerant charge decreased by 28% for the R152a and by 60% for the R290. The COP for the R152a increased by 5% for highway driving and 10% for idling conditions, while the COP of R290 is increased by 8% under highway driving conditions and decreased by 15% under idling conditions.

Joudi and Al-Amir [23] performed experimental tests to find an alternative of R22 using R290, R407C and R410A at high ambient temperatures for 1 and 2 tons of refrigeration (TR) systems. The optimum charges obtained for R290, R407C and R410A were 500 g, 1100 g and 1600 g, respectively for the 1 TR unit while, the optimum charges in the 2 TR unit were found to be 900 g, 1900 g and 2800 g, respectively. Tian et al. [24] used a mixture of R32/R290 consisting of 68/32 wt % as a drop-in replacement for R410A to operate two domestic air conditioners having a capacity of 3.2 and 5.2 kW. The refrigerant mixture allowed the usage of 30-35% smaller charge of refrigerant and 14-23.7% higher cooling capacity. The COP increased by 6.4%.Antunes and Filho [25] studied the change in performance parameters of a refrigeration cycle using six different refrigerants; R438A, R404A, R410A, R32, R290 and R1270, as an alternative to R22. The most suitable refrigerant to be used in air conditioning applications to replace R22 was R1270, while R404A was the most suitable for product storage applications. HCs have the highest cycle COP with lower electric energy consumption, reduced refrigerant charge and low GWP.Cho and Park [26] compared the performance of an automotive air conditioning system using R1234yf as an alternative to R134a. The R1234yf system had 0.3-2.9% lower COP compared to the R134a system at 800-1800 rpm of the compressor, while the exergy loss of the R1234yf was 0.5-3.3% higher than that of the R134a in the same speed range. The second law efficiency of the R1234yf was found to be 3.4-4.6% lower than that of the R134a at all compressor speeds.Wang et al. [27] reviewed numerous analytical and experimental works concerning the use of secondary loop refrigeration systems. Secondary loop systems were very competitive systems that enabledthe use of HC refrigerants for systems that require a relatively large system charge. Experimental studies proved the safe operation of HC refrigerants in refrigeration and air conditioning applications. Using minichannel heat exchangers, installing exhaust fans and leakage sensors decrease the risk of using HCs.

As previously mentioned, the main concern about using hydrocarbons in refrigeration is the consequences that occur due to their leakage. Using HC refrigerant poses the risk of explosion and fire. Some researches went to evaluate the safety and the flammability hazards of HC refrigerants. Experimental work has been conducted on air conditioners and refrigerators. Zhang et al. [28, 29] considered to substitute R22 and R410A used in domestic air conditioner by R290. The test was done to measure the heat release rate and the extent of smoke emissions using a standard ignition source of 100 kW. They found that the possibility of explosion was low and will not damage the air conditioner unit. Colbourne and Suen [30] used R290 and R600a as an alternative to R22 in a split air conditioner and a domestic refrigerator. The calculated ignition frequency of the air conditioner was one event per 100 million in 10 years. The domestic refrigerator was one event per million in 10 years. Also, if the leakage occurred from the outdoor unit of the air conditioner, the concentration in the outdoor air was very small. The thermal intensity of both appliances was less than 100s which wasmuch less than the value that can cause fatal losses or significant injuries. Zhang et al. [31] considered to substitute R22 and R410A used in domestic air conditioner by R290 to determine the possibility of ignition and explosion during and after the leakage occurrence when the indoor and outdoor units contained very small charge of refrigerant. They concluded that the possibility of explosion was low and will not damage the air conditioner unit. However, if the ignition occurs during the leakage, both units will be burned.

The compatibility and the stability of HC refrigerants with the lubrication oils used in refrigeration is a major parameter that should not be disregarded while trying to select HC drop-in refrigerants instead of CFCs and HFCs. Navarro et al. [32] tested five R407C positive displacement hermetic reciprocating compressors covering different capacities, displacement volume and number of cylinders to determine the performance characteristics using R290. The compressors use Polyolester Oil (POE) oil as a lubricant. R290 showed a better thermodynamic behavior and a mean COP improvement by 9% greater than the COP when using R407C. The compatibility of POE oil was proven with R290 during the test. Ravikumarand Lal [33] tested a mixture of R134a with a HC blend of R290/R600a with the conventional mineral oil as lubricant. The commercially available HC blend of R290/R600a having 43.2/56.8 wt % was mixed with R134a in the proportion of 9/91 wt %. The test results showed that the new blend can be a promising substitute for R12 systems. Sergio et al. [34] measured the solubility of R1234yf in Polyalkylene Glycol (PAG) and Double-Capped Polyalkylene Glycol (DC-PAG) oils at temperatures ranging from 258 K to 338 K. The oil and refrigerant solubility increased with the temperature decrease. Theoretically, DC-PAG was more compatible with R1234yf. However, at temperatures higher than 293 K, with PAG, and 303 K, with DC-PAG. The measurements showed a liquid phase splitting in the high R1234yf mass fraction region.

3. Hydrocarbon as refrigerant

3.1. Environmental benefits

In order to reduce the effects of global warming caused by the usage of halogenated refrigerants, HC refrigerants were introduced as competing alternatives to these refrigerants. Values of ODP and GWP are compared in Table 1 for various types of refrigerants.

3.2. Safety and Flammability

The main disadvantage of the HCs is their high flammability. Fire occurs when HC refrigerant is mixed with air in presence of a source that initiates a spark. Using electric contactors, switches or relays should follow some specific precautions in order to prevent electric sparks. Ignition surfaces like hot surfaces should be avoided. Table 2 shows that refrigerants are classified according to their flammability into three categories; 1, 2 and 3, and are classified according totheir toxicity into two groups, A and B. A2L and B2L are refrigerants of low flammability with burning velocity less than 10 cm/s.Refrigerants R290, R1270 and R290/R1270 mixture used in this paper are classified as A3.

3.3. Lubricants and Material compatibility

Hydrocarbons are compatible with all metals used in refrigeration systems like steel, copper, aluminum, zinc, tin, lead and their alloys as well as all common elastomeric and plastic refrigeration material like O-rings, valve seats, seals and gaskets [35, 36].HC refrigerants have full chemical compatibility with almost all lubricants commonly used in refrigeration systems. Good miscibility is maintained with most lubricants under all operating conditions[36].

3.4. Thermodynamic properties

In this section, the thermodynamic properties of R22, R290, R1270 and R290/R1270 mixture 15/85 by wt %are compared. The main parameters that make HCmore competing alternatives compared to halogenated refrigerants are their higher latent heat and lower specific volume. Figures 1, 2, 3 and 4 show the variation of the different thermodynamic properties of R290, R1270 and R290/R1270 mixture compared to R22at different temperatures that could occur in the evaporator and condenser of the refrigeration cycle. The specific volume of all HCrefrigerants is lower than R22 due to their smaller molecular weight. This difference will result in a much lower mass flow rate of HC refrigerants for the same cooling capacity. R290 is 44%-51.5% less dense that R22, while the density of R1270 and R290/R1270 mixture are 52%-62% lower than that of R22.

The saturation pressure is almost equal at low temperatures. The saturation pressure of R22 is higher than that of R290 and slightly lower than that of R1270. For this reason, many researchers [6 and 7] tried to get a mixture of R290/R1270 which has a saturation pressure equivalent to R22.

The latent heat of evaporation (h_{fg}) of refrigerants is one of the most important factors that must be kept in mind while selecting the suitable refrigerant for a certain application. Also, it has a major effect on determining the mass flow rate of refrigerant through the cycle. It affects the sizing of system components; the evaporator, the condenser and the compressor. The latent heat of HCrefrigerants is 54%-56% higher than that of R22 at different evaporator and condenser temperatures. These properties made the HC refrigerants a favorable alternative in refrigeration and air conditioning applications. The effect of these properties on the overall cycle performance will be discussed in the next section of this work.

4. Cycle analysis

The cycle in this study is a simple vapor compression cycle that consists of an evaporator, a compressor, a condenser and an expansion device. Figure 5 shows a schematic diagram for this cycle. Thermodynamic analysis is performed based on the following assumptions:

1. Steady state operation in all components [37].
2. Adiabatic compressor and expansion valve [38].
3. No work is exerted in the evaporator or condenser [39].
4. No pressure drop due to friction in the evaporator and condenser [38].
5. The isentropic efficiency of the compressor is equal to 85% [38].
6. The enthalpy is constant across the expansion valve.
7. The refrigerant exits from the evaporator with 3 K of superheat [38].
8. The refrigerant exits from the condenser with zero degree of subcooling [38].

The different states along the cycle are shown in Table 3.

4.1. Thermodynamic analysis:

This section shows the mathematical model followed while applying the first law and the second law analysis on each of the cycle processes. All states in this analysis isdepictedon the pressure-enthalpy diagram shown in Figure 6.

1. Evaporator:

The specific cooling load across the evaporator, in kJ/kg, is given by Equation1.

$$Q_{evap} = h_2 - h_6 \quad (1)$$

The refrigerant mass flow rate($\dot{m}_{ref}$)through the cycle is calculated by dividing the constant evaporator cooling load, which is kept constant at 1 kW, by the change in the specific enthalpy in it using Equation 2.

$$\dot{m}_{ref} = \frac{1}{h_2 - h_6} \quad (2)$$

The irreversibility across the evaporator, per unit mass of the refrigerant, is defined by Equation 3 as:

$$I_{evap} = (\psi_6 - \psi_2) + Q_{evap}(1 - \frac{T_{evap}}{T_0}) \quad (3)$$

where $T_0$and s_0 are the temperature and the specific entropy at ambient standard conditions,T_0=293 K and p_0=1 bar.ψ is the availability of the fluid and it is calculated from Equation 4.

$$\psi = (h - h_o) - T_o(s - s_o) \quad (4)$$

2. Compressor:

The specific work input across the compression process, in kJ/kg, is given by Equation 5.

$$W_{comp} = h_3 - h_2 \quad (5)$$

The actual enthalpy at the compressor exit, h_3 is calculated from the compressor isentropic efficiency given by Equation 6.

$$\eta_{is} = \frac{h_{3is} - h_2}{h_3 - h_2} \tag{6}$$

State (3is) is the state of the refrigerant at the exit of the compressor after undergoing an isentropic compression process.The irreversibility across the compressor, per unit mass of the refrigerant, is defined by Equation 7.

$$I_{comp} = (\psi_2 - \psi_3) + W_{comp} \tag{7}$$

3. Condenser:

The amount of heat rejected in the condenser, per unit mass of the refrigerant, is calculated using Equation 8.

$$Q_{cond} = h_3 - h_5 \tag{8}$$

The irreversibility across the condenser, per unit mass of the refrigerant, is defined by Equation 9.

$$I_{cond} = (\psi_3 - \psi_5) - Q_{cond}\,(1 - \frac{T_0}{T_{cond}}) \tag{9}$$

4. The expansion device:

The process across the expansion device is assumed to be adiabatic, with no change in the kinetic or potential energies. Therefore, by applying the first law of thermodynamics across the process, the enthalpy remains constant, as shown in Equation 10.

$$h_5 = h_6 \tag{10}$$

The irreversibility across the expansion valve, per unit mass of the refrigerant, is defined by Equation 11.

$$I_{exp} = (\psi_5 - \psi_6) \tag{11}$$

The coefficient of performance of the cycle, COP is calculated from Equation 12.

$$COP = \frac{Q_{evap}}{W_{comp}} \tag{12}$$

The total cycle irreversibility is equal to the algebraic sum of the irreversibility in each process as given by Equation 13.

$$I_{total} = I_{evap} + I_{comp} + I_{cond} + I_{exp} \tag{13}$$

The exergetic efficiency (η_{exerg}) of the cycle is defined by the irreversibility at the evaporator divided by the work input to the cycle and it is calculated from Equation 14.

$$\eta_{exerg} = \frac{I_{evap}}{W_{comp}} \tag{14}$$

The coefficient of performance based on Carnot cycle assumptions (COP_{carnot}) is calculated from Equation 15.

$$COP_{carnot} = \frac{T_{evap}}{T_{cond} - T_{evap}} \tag{15}$$

The second law efficiency ($\eta_{2nd\ law}$) of the cycle is defined as the ratio between COP to COP_{carnot} and is given by Equation 16

$$\eta_{2nd\ law} = \frac{COP}{COP_{carnot}} \tag{16}$$

Where T_{evap} and T_{cond} are the saturation temperatures at the evaporator and condenser pressure, respectively.

5. Results and discussion

5.1. For condenser temperature of 313 K:

In this section, the condenser temperature is kept constant at 40°C and the evaporator temperature is varied by 249, 253, 257, 261, 265, 269, 273 and 277 K. First, the first law analysis is performed in which the $\dot{m}_{ref}$, VRC, the compressor exit temperature, the cycle COP, the exergetic and the second law efficiencies are compared for 1 kW cooling capacity.

The results of this study show that the mass flow rate of all refrigerants decreases with increasing the evaporator temperature due to the increase in h_{fg} for the same cooling capacity of 1 kW, as shown in Figure 7. Also, Figure 7 indicates that R1270 provided the lowest refrigerant flow rate. The mass flow rates of HC refrigerants are almost equal and are 39.7-43.5% lower than that of R22 which reduces the total refrigerant charge.

Another important parameter that affects the cycle performance is the VRC. VRC is the cooling load per unit volume of the refrigerant at the evaporator inlet. This parameter is higher for R22 due to its low specific volume, as shown inFigure 8.

Figure 9 shows one of the most important parameters in this discussion; the compressor exit temperature. In general, all fluids have a decreasing compressor exit temperature with the increase in the evaporator saturation temperature, as shown on the pressure-enthalpy diagram in Figure 6.

While all HC refrigerants have almost half the $\dot{m}_{ref}$ of halogenated refrigerants and double the latent heat of R22, R290 shows the least compressor exit temperature which is a key parameter in selecting the condenser size and the condenser cooling fluid, whether it is air cooled or water cooled. Also, it affects the thermophysical properties of the lubrication oil and the overall compressor life-time and performance. Cycles working at low evaporator temperatures that employ R22, the compressor head requires a special cooling process due to its high discharge temperature.

As the evaporator temperature increase, the saturation pressure increases which reduces the power input to the compressor, as shown in Figure 10. The power consumed by the system using

R22 is slightly lower than that of hydrocarbons, which indicate that hydrocarbons can be used as drop-in refrigerant of R22 using the same compressor.

The COPincreases with the increase in the evaporator temperature due to the reduction in the compressor work, as shown in Figure 11. TheCOP for all refrigerants is similar with a very slight increase in favor of R22 for the same cooling load of 1 kW.

In the previous part of this section, the first law analysis is performed for the different evaporator temperatures and the same condenser temperature. The following discussion will compare the cycles from the point of view of the second law of thermodynamics.

The irreversibility in the evaporator decreases with the decrease in the specific cooling load and the increase in the evaporator temperature. Thus, the exergetic efficiency decreases with the increase in the evaporator temperature, as shown in Figure 12. The exergetic efficiency of the refrigeration cycle using R22 is only 3% greater than that of HC refrigerants.

The second law efficiency, given by Equation 16, for all refrigerants increases with the increase in the evaporator temperature, as shown in Figure 13. The second law efficiency of the refrigeration cycle using R22 is about 2% higher than that of hydrocarbons.

Table 4 shows the numerical percentage difference between R22 and the HC refrigerants; R290, R1270 and R290/R1270 mixture at the lowest and highest evaporator temperatures for a condenser temperature of 313 K.

5.2. For evaporator temperature of 263 K:

In this section, the condenser temperature is changed from 308 to 313, 318, 323, 328 and 333 K for a constant evaporator temperature of 263 K. The first law and second law analysis are performed to observe the effect of changing the condenser temperature on the following parameters; $\dot{m}_{ref}$, VRC, the compressor exit temperature, the cycle COP, the exergetic and the second law efficiencies.

Figure 14 shows that the mass flow rate of all refrigerants decreases with increasing the condenser temperature due to the increase in h_{fg} for the same cooling capacity of 1 kW. The mass flow rate of HC refrigerants is almost equal and is 37% to 43% lower than that of R22.

The variation of the VRC is shown in Figure 15. This parameter is higher for R22 due to its low specific volume, and decreases with the increase in the condenser temperature due to the decrease in Q_{evap} and the increase in the specific volume of the liquid entering the evaporator.

For all types of refrigerants, by increasing the condenser temperature, for the same degree of sub cool and superheat, VRC decreases as the specific volume of the liquid at the evaporator inlet increases. This effect is shown as an example for R22 on the p-h diagram Figure 6 and the effect is similar for all the refrigerants used in this study.

The compressor power consumption increases with the increase in the condenser temperature and pressure. R22 and hydrocarbons have almost the same power consumption at low condenser temperatures while; R22 consumes a slightly lower power at higher condenser temperatures than hydrocarbons, as indicated in Figure 17.

Figure (18) indicates that the compressor exit temperature increases with the increase in the condenser temperatures for all types of refrigerants. Also, R22 showed the highest compressor discharge temperature which makes it a less favourable working fluid.

As the condenser temperature increase, the saturation pressure at the compressor exit increases; this results in an increase in the power input to the compressor and a decrease in the evaporator specific cooling load, as shown in Figure 17. As a result, COP of the cycle decreases with the increase in the condenser temperature. Figure 18shows the comparison between the$COP$of R22 and HC refrigerants.

The exergetic efficiency decreases with the increase in the condenser temperature as shown in Figure 19. The irreversibility in the evaporator decreases with the decrease in the specific cooling load and the increase in the condenser temperature. The exergetic efficiency of R22 is around 2% higher than that of the same refrigeration cycle employing HC refrigerants.

The second law efficiency for all refrigerants decreases with the increase in the condenser temperature due to the increase in theCOP. The second law efficiency of the refrigeration cycle using R22 is around 3% higher than that of the HC refrigerants, as shown inFigure 20.

Table 5 shows the percentage difference between R22 and the HC refrigerants; R290, R1270 and R290/R1270 mixture at the extreme condenser temperatures for a fixed evaporator temperature of 263 K.

6. Conclusion

This study shows that natural HC refrigerants are promising substitutes of the synthetic hydrofluorocarbon refrigerant, R22. They have zero ODP, very low GWP, nearly equal saturation pressures on a wide range of temperatures and double the latent heat compared to R22. The mass of hydrocarbons are 40-44% lower than that of R22 and, the second law and exergetic efficiencies are 2-4% lower than that of R22. VRC of R22 is 6.36-14.5% higher than R1270 and R290/R1270 which makes them an acceptable substitute of R22 in refrigeration cycles without the need of replacing any of the cycle components as long as it is lower in the range of 20% [7].

On the other hand, hydrocarbons pose the danger of fire and explosion due to their high flammability. However, researches proved that the use of HC refrigerants is safe when used in domestic refrigerators and air conditioners. In high capacity systems, the usage of leakage detectors and ventilation make it safe to use hydrocarbons according to the safety standards [36, 40].

References

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Tables

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Table 1: Comparison between ODP and GWP of refrigerants (ANSI/ASHRAE Standard 34-2007 [36]

Chemical Group	Refrigerant Number	Safety Group	ODP	GWP
CFC	R11	A1	1	4680
	R12	A1	1	10900
HCFC	R22	A1	0.05	1700
HFC	R134a	A1	0	1200
HFO	R1234yf	A2L	0	4
	R1234ze	A2L	0	6
HC	R290	A3	0	3
	R600	A3	0	3
	R600a	A3	0	3
	R1270	A3	0	2
Zeotropic blends	R404A	A1	0	3922
	R407A	A1	0	2107
	R410A	A1	0	2088
Azeotropic blends	R502	A1	0.33	5600
	R507A	A1	0	3985
Inorganic compounds	R717	B2	0	0
	R718	A1	0	0
	R744	A1	0	1
	R764	B1	0	0

Table 2: Safety and flammability classifications

	Safety group	
High flammability	A3	B3
Low flammability	A2	B2
	A2L	B2L
Non flammable	A1	B1

	Low toxicity	High toxicity

Table 3: Points nomenclature and states along the cycle

Point	State
1	Saturated vapor at p_{evap}
2	Actual evaporator exit where $T_2 = T_1 + \Delta T_{superheat}$
3	Actual compressor exit
3s	Isentropic compressor exit
4	Saturated liquid at p_{cond}
5	Actual condenser exit where $T_5 = T_4 + \Delta T_{subcool}$
6	Expansion valve exit

Table 4: The percentage difference of the performance parameters of R290, R1270 and R290/R1270 compared to those of R22 at various evaporator temperatures.

	R290		**R1270**		**R290/R1270**		**R22**	
T_{evap} **[K]**	**249**	**277**	**249**	**277**	**249**	**277**	**249**	**277**
% reduction in $\dot{m}_{ref}$	39.74	42.65	41.75	43.50	41.24	43.14	These values are calculated as a percentage of that of R22	
% increase in h_{fg}	54.9	54.7	54.2	54.3	54.5	54.5		
% increase in v_f	58.8	58.7	57.3	57.4	57.6	57.6		
% increase in v_g	48.4	51.6	37.7	42.5	38.8	43.3		
% decrease in VRC	27.28	28.3	6.36	12.23	9.37	14.55		

% increase in $\dot{W}_{comp}$	4.2	2	3.9	2.6	4.1	2.6		
η_{exerg} %	76	59.3	76.2	58.9	76.1	59	79.2	60.5
$\eta_{2nd\ law}$ %	59.8	68.6	60	68.2	59.9	68.2	62.3	70
COP	2.327	5.28	2.33	5.25	2.329	5.249	2.425	5.39

Table 5: The percentage difference of the performance parameters of R290, R1270 and R290/R1270 compared to those of R22 at various condenser temperatures.

	R22		**R290**		**R1270**		**R290/R1270**	
T_{cond} [K]	**308**	**333**	**308**	**333**	**308**	**333**	**308**	**333**
% reduction in $_{ref}$	41.9	37.3	43.2	39.4	42.7	38.6	These values are calculated as a percentage of that of R22	
% increase in h_{fg}	54.3	54	54.8	56	55	56.1		
% increase in v_f	58.6	58.5	57.6	57.9	57.8	58.1		
% increase in v_g	54.2	55.6	45.8	47.1	46.6	47.9		
% decrease in VRC	27.1	32.7	8.6	14.8	12.1	18.5		
% increase in $\dot{W}_{comp}$	2.3	7.9	2.56	7.74	2.61	8.12		
η_{exerg} %	84.4	42.6	84.2	42.6	84.2	42.5	86.4	46
$\eta_{2nd\ law}$ %	66.8	52.4	66.6	52.5	66.6	52.5	68.3	56.5
COP	3.9	1.97	3.89	1.97	3.89	1.96	3.99	2.12

Figures

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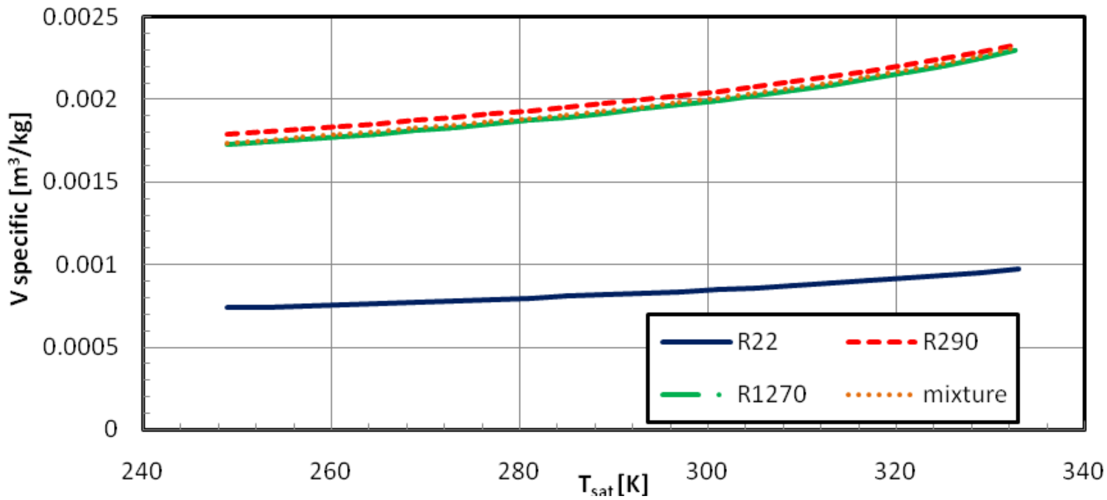


Figure 1: Specific volume of saturated liquid at different temperatures

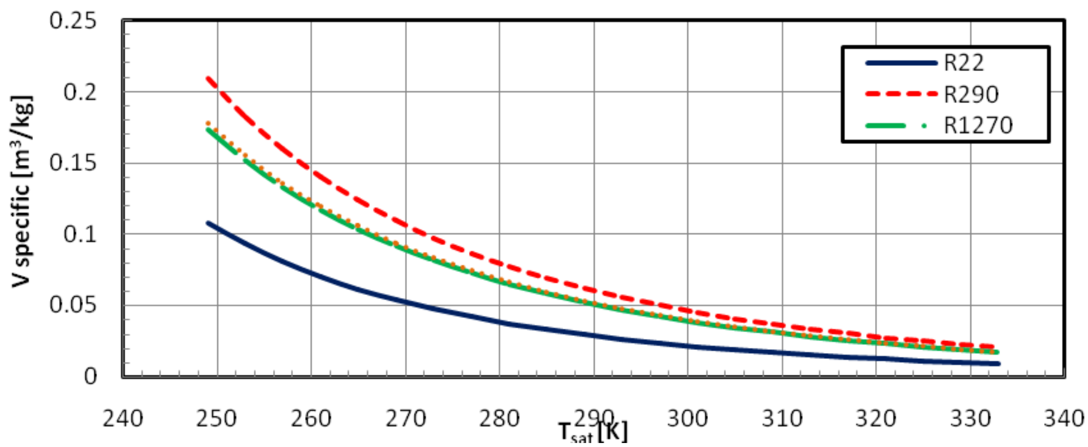


Figure 2: Specific volume of saturated vapor at different temperatures

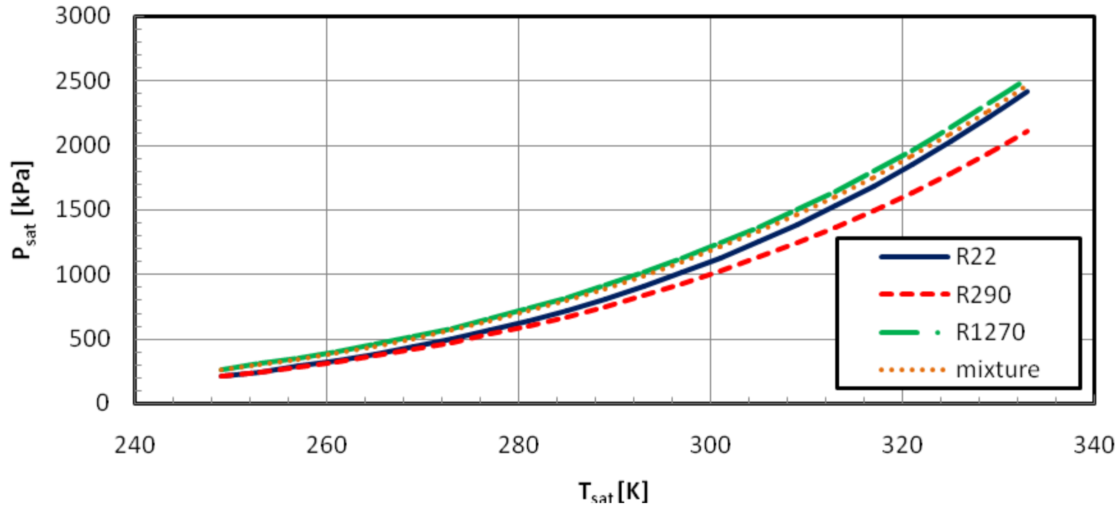
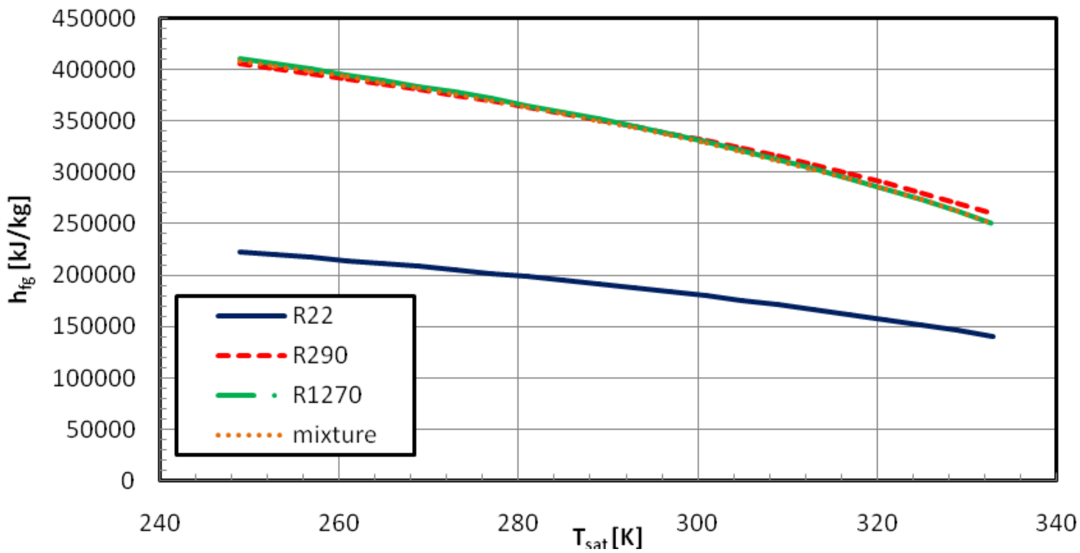


Figure 3: The variation of saturation pressure of refrigerants with the temperatures



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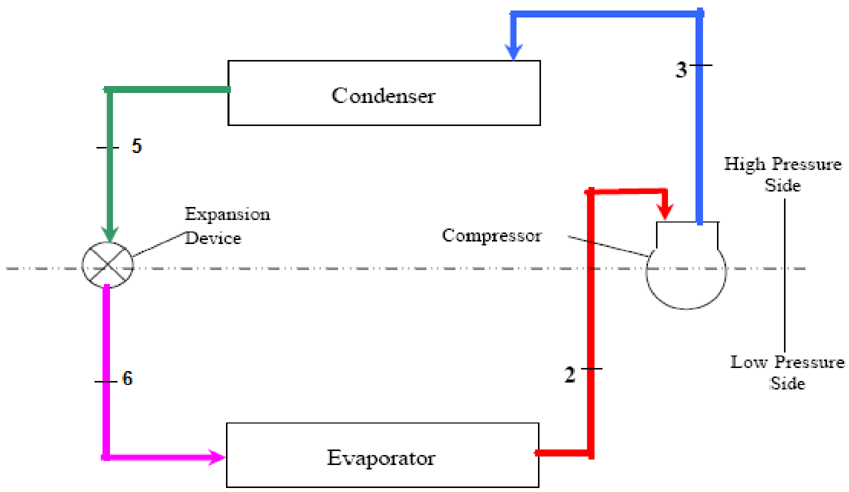
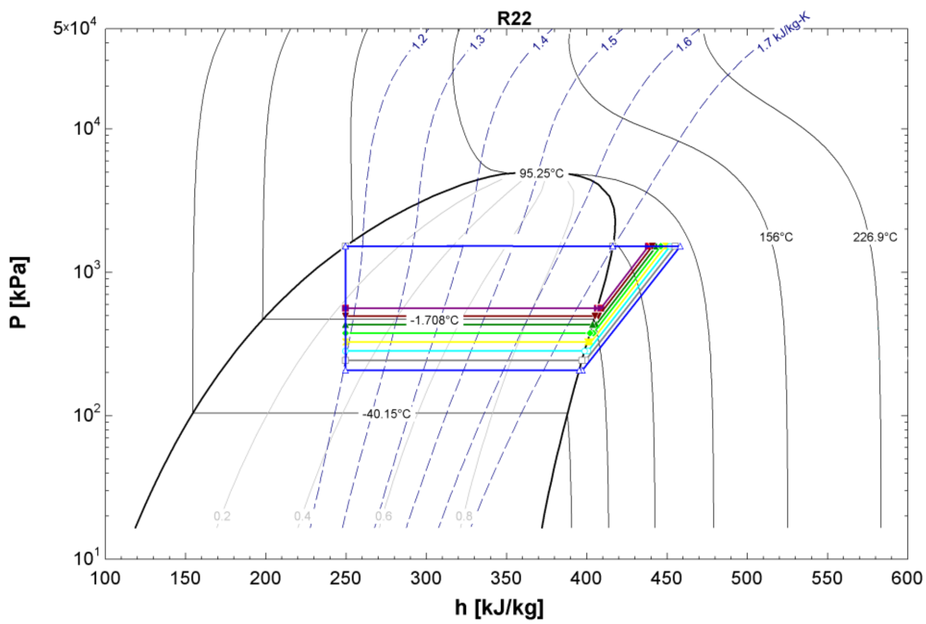


Figure 5: Schematic sketch of the cycle



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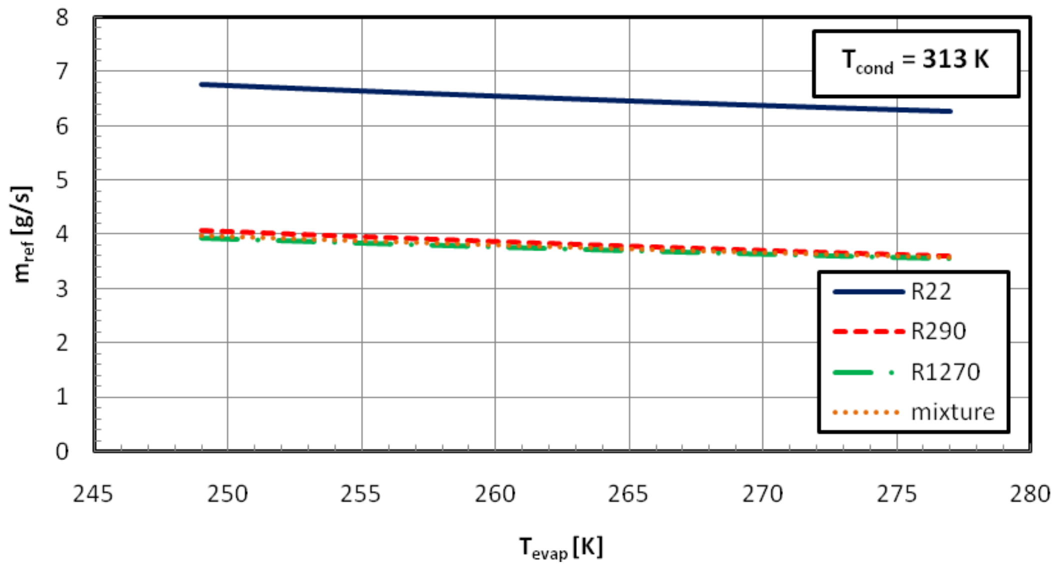

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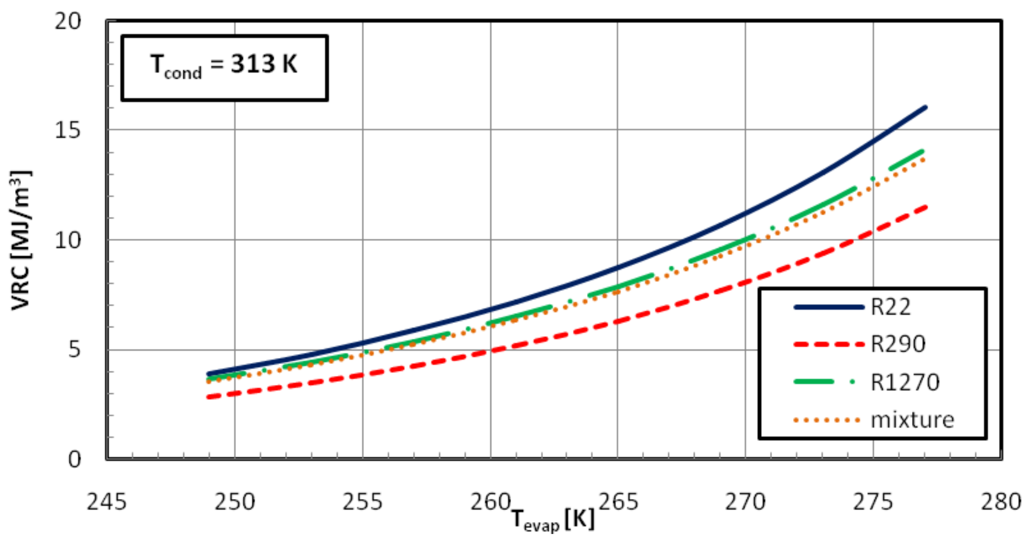


Figure 8: VRC of refrigerant of R22, R290, R1270 and R290/R1270 mixture at different evaporator temperatures

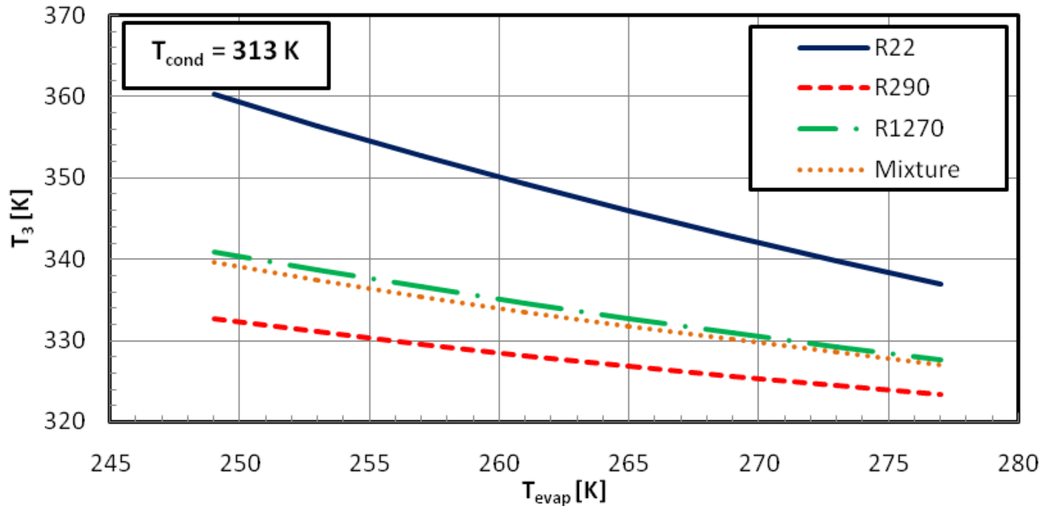
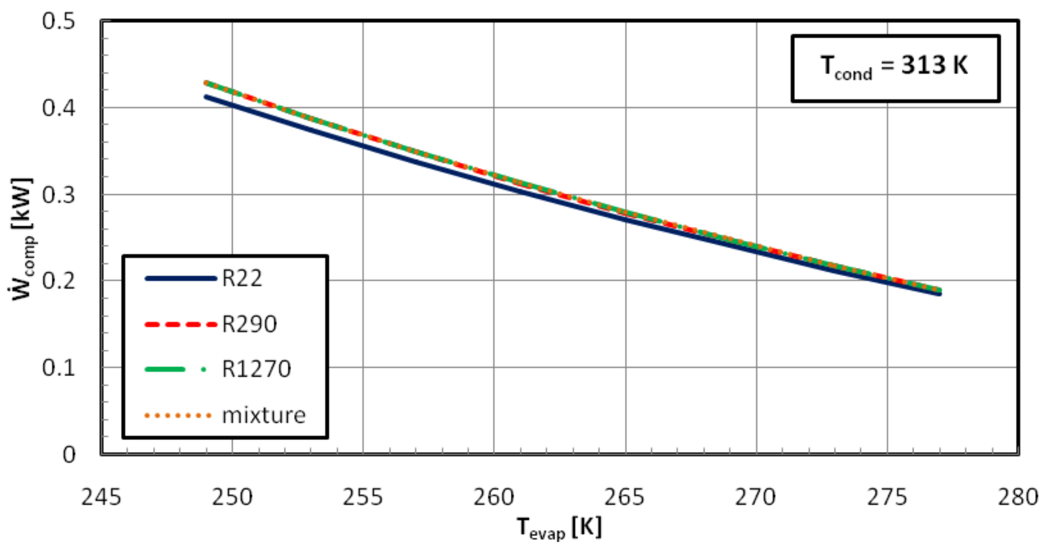


Figure 9: The compressor exit temperature of R22, R290, R1270 and R290/R1270 mixture at different evaporator temperatures



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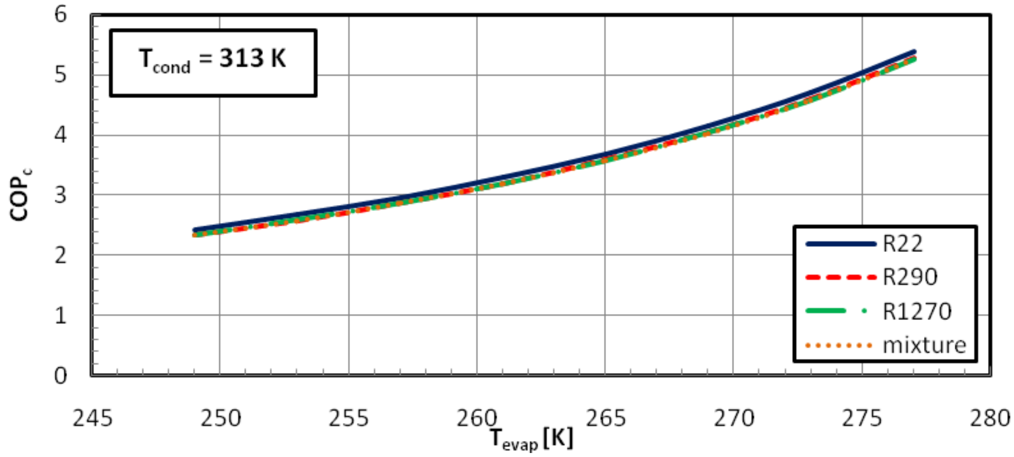

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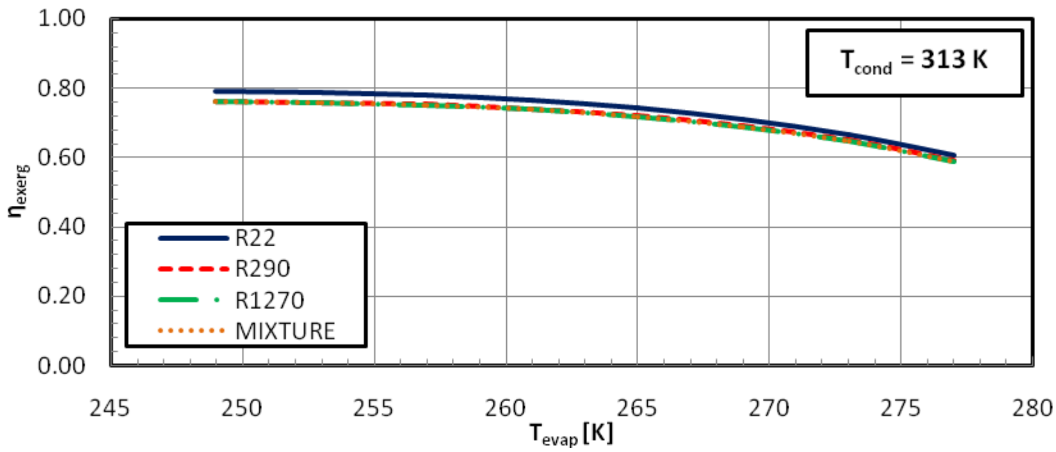

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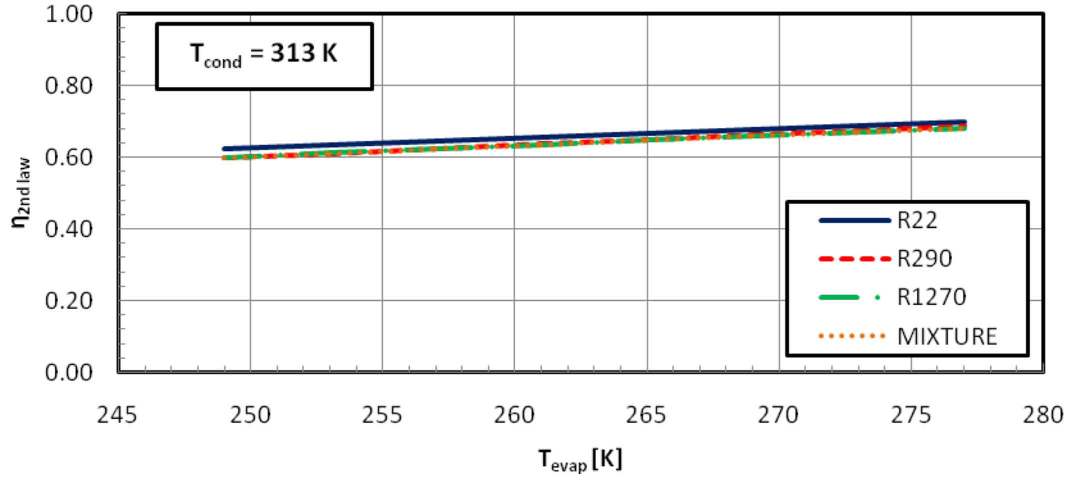
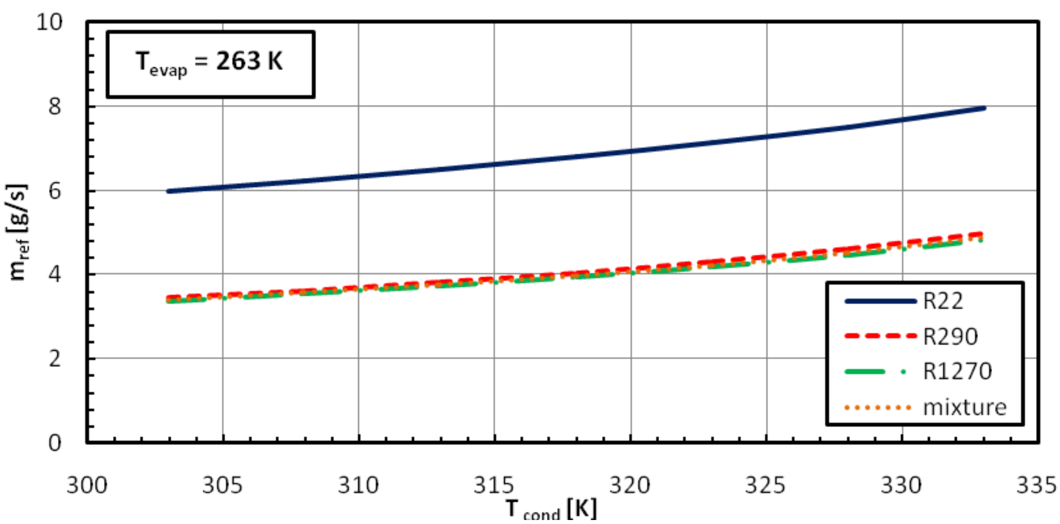


Figure 13: The second law efficiency of R22, R290, R1270 and R290/R1270 mixture at different evaporator temperatures



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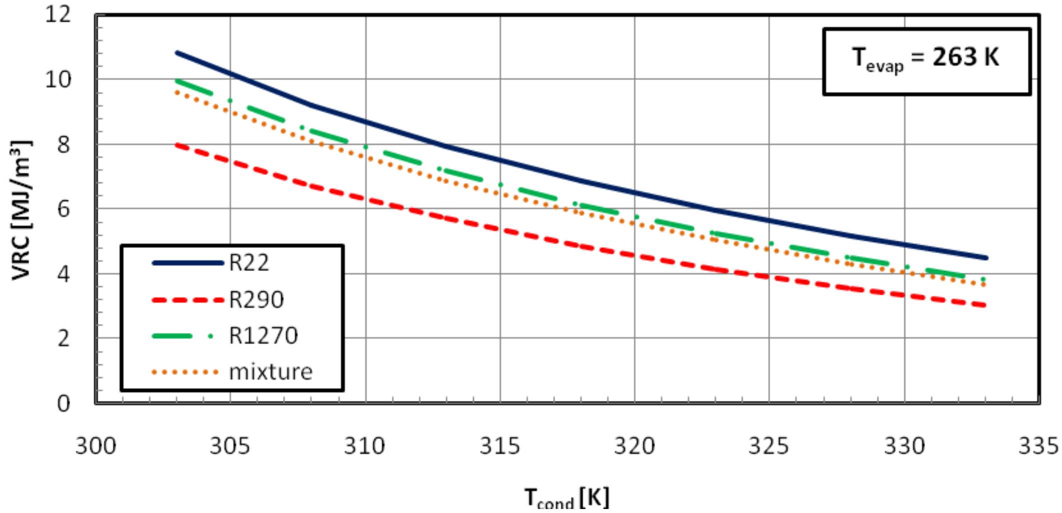


Figure 15: VRC of refrigerant of R22, R290, R1270 and R290/R1270 mixture at different condenser temperatures

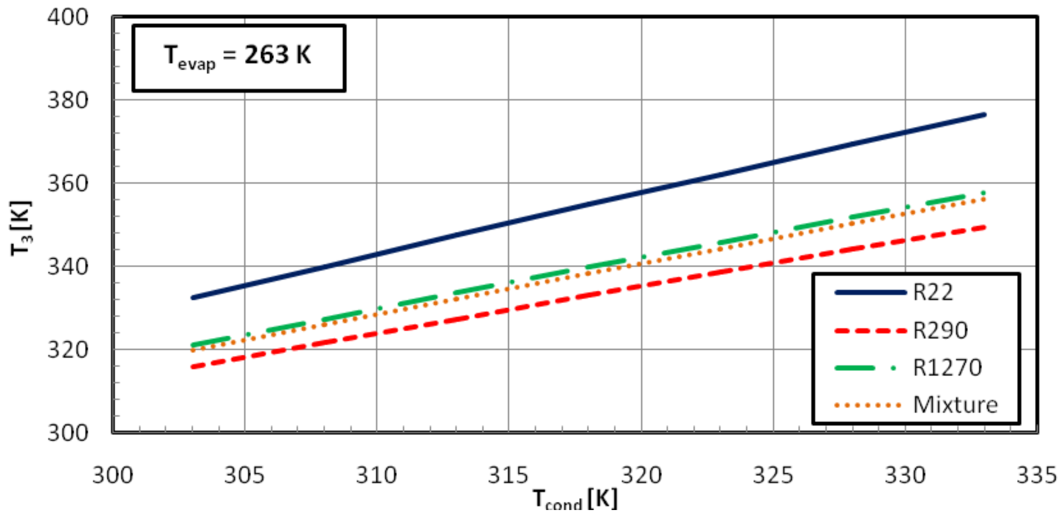


Figure 16: Compressor exit temperature of refrigerant of R22, R290, R1270 and R290/R1270 mixture at different condenser temperatures

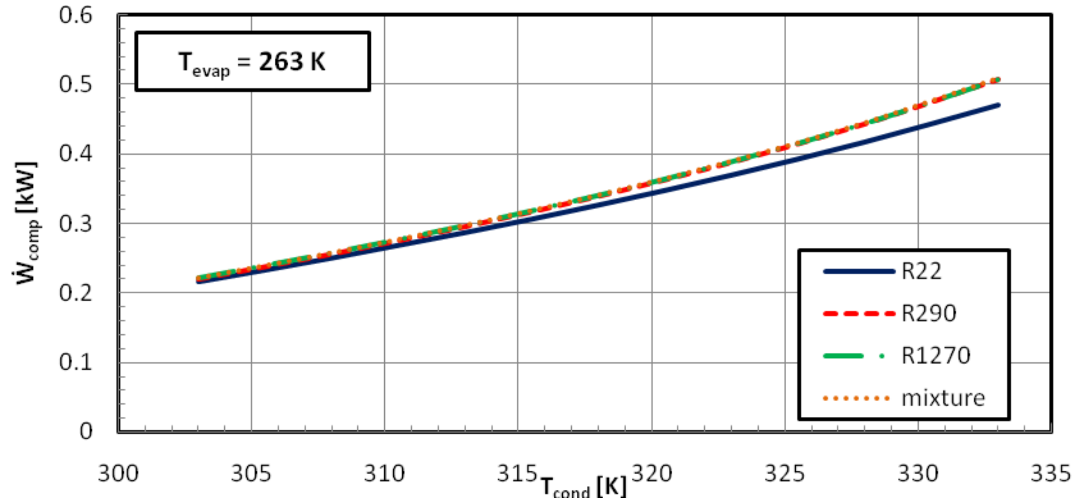


Figure 17: Compressor power consumption of R22, R290, R1270 and R290/R1270 mixture at different condenser temperatures

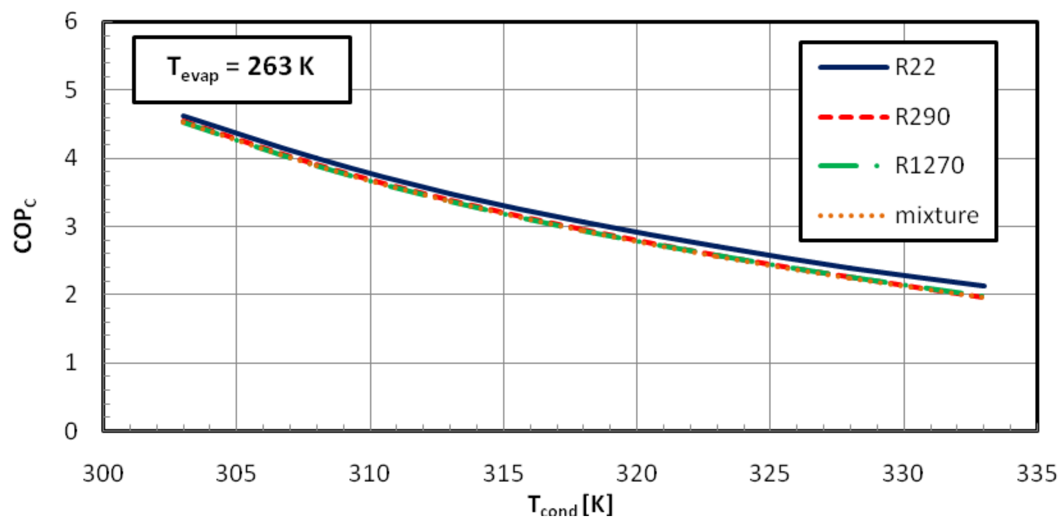
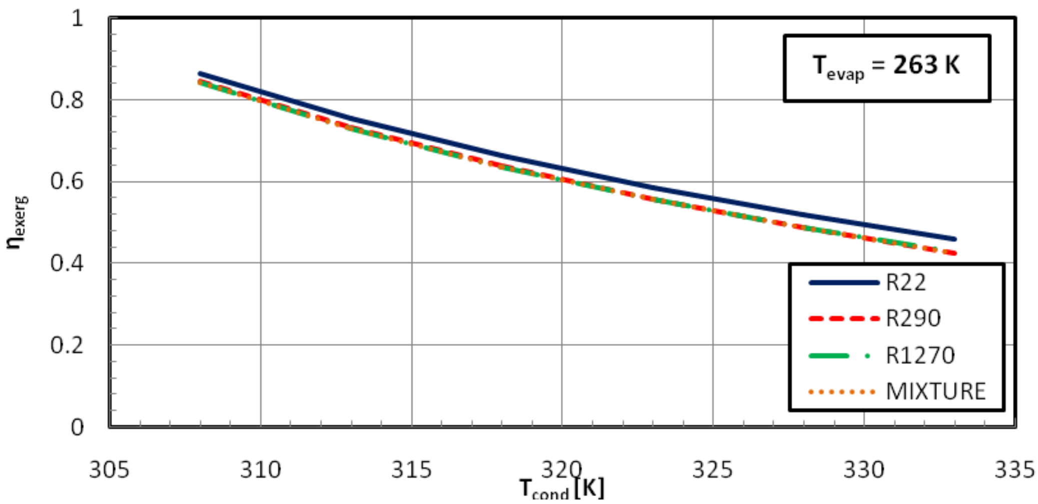


Figure 18: COPof R22, R290, R1270 and R290/R1270 mixture at different condenser temperatures



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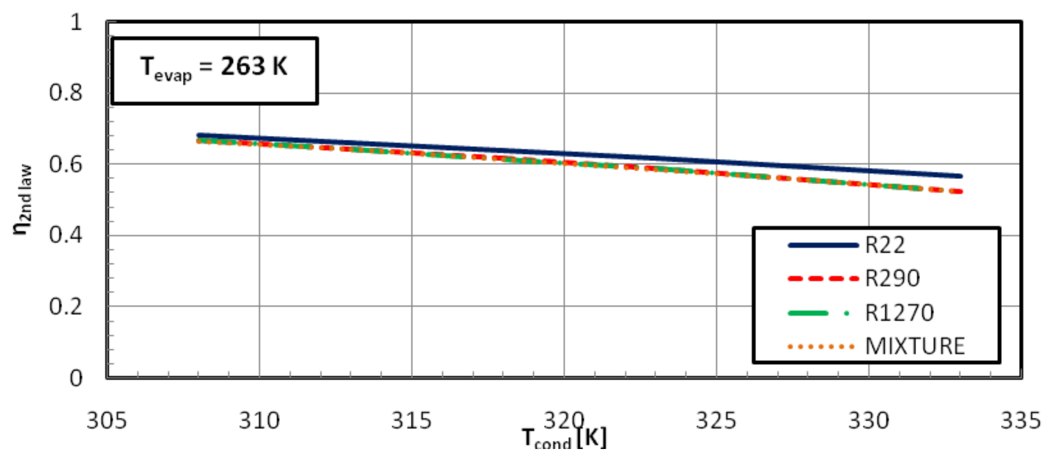

Figure 20: The second law efficiency of R22, R290, R1270 and R290/R1270 mixture at different condenser temperatures