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INFLUENCE OF VARIATION AMBIENT TEMPERATURE ON VAPOUR COMPRESSION SYSTEM PERFORMANCE WITH 12 TYPES OF FERION

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Abstract- The ambient temperature has been considered one of the important factors that directly affected on the vapour compression system performance. In this paper, a numerical study of vapour compression system performance with twelve types of ferion (R134a, R1234ze, R22, R12, R290, R1234yf, R152a, R245fa, R407C, R600a, R410A, and R507A) under different ambient temperature varied from 35 °C to 55 °C. It has been found that, the using of R245fa instead of R22 increased the coefficient of performance by 1.7 at the same ambient temperature. The ambient temperature has a little effect on compressor work reached to 2.5% increment with the increase of ambient temperature. The results show that, R290 and R600a do not condensate to saturated liquid line. The system capacity can be increased by 30% when R134a replaced by R152a. R410A and R507A do not effective at higher ambient temperature. R134a can be replaced by R1234yf and R1234ze with same performance due to the growing concerns about the relatively high global warming potential of current ferion.

Keywords: Ferion, Sub-Cooling, COP, Ambient Temperature.

1. INTRODUCTION

The air conditioning system is very significant to give relaxation and comfort to people in the room. In the selection of the best system, some criteria must be considered to ensure give best performance. Refrigerant type, the temperature outside, evaporator, compressor, and condenser should be considered. Usually, the changing of ambient temperature according to place type. Generally, the temperature close to coast areas considered high as compared with inland regions. The difference in ambient temperature has significant effect on the coefficient of performance (COP) of air conditioning cycle system [1].

Chlorofluorocarbons (CFCs) such as ferion (11, 12, 22 and 502) were banned because of the potentially of causing large problems in the environment as both significant contributor toward the ozone layer destruction and greenhouse effect after being prevalently used in the air conditioning and refrigeration systems [2-3].

The preventing was done under "Montreal Protocol" in 1987 which enforcing that the Chlorofluorocarbons (CFCs) to be fully phased out in the developed countries starting from 1996 and developing countries starting from 2010 [4]. Cheapness, high coefficient of performance (COP), application flexibility, superiority on heat transfer and thermodynamic properties, therefore, low maintenance and operation cost made these refrigerants of widespread utilization in this market. As well as, CFCs are considered as a destructive to the global environment due to their chemical content. They have both of high ozone depletion potential as well as a global warming potential. Therefore, the operating and performance system characteristics of environmentally friendly alternative refrigerants system in place of hydro-CFC and CFC have been studied on theoretical and experimental principles for refrigeration cycle systems in the recent years.

An experimental investigation of a typical small volume of food freezer has been accomplished by Torchio and Anglesio (2004) [5] to determine the refrigeration cycle. The conditions of design were respected when ($T_{ambient}$) was changed from 30 °C to 40 °C. Inside the cool rooms, the temperature had a considerable thermal gradient that be controlled in all the points to continuously achieve the correct temperature level. Bjork and Palm (2006) [6] present experimental outcomes of an on/off cycling refrigerator system at varied ambient temperature, quantity of charge and expansion device capacity. It was revealed that the ideal system charge increased at ambient temperature decreased, for an altered ambient temperature, due to a minimize redistribution process, as well as by more refrigerant that held up in the compressor at lower ($T_{ambient}$), these increasing the optimum charge. An exergy analysis study of an actual vapour compression refrigeration cycle (VCR) presented by Arora and Kaushik (2008) [7]. A computational relationship has been developed for calculating exergy destruction, coefficient of performance (COP), efficiency defects and energetic efficiency for Ferion (502, 404A and 507A).

The results showed that the efficiency defect is lowest in liquid vapour heat exchanger, and highest in condenser for the refrigerants considered. Also, ferion 507A is a better substitute to ferion 502 than ferion 404A.

Experimental study has been carried out by Mathur (2010) [8] to calculate the AC performance using "HFO-1234yf" as the working fluid. The inside cabin condition is kept constant at 50% RH and 20 °C, airflow rate in evaporator is varied between 5-9 m^3/min and velocity of condenser air face about 2 m/s with ambient temperatures kept at 25 °C, 35 °C and 45 °C, respectively. The results indicate that, the performance of system (COP) with HFO as "drop in" is higher than the base case, essentially at ambient temperature greater than 30 °C. The capacity of cooling is greater than the base case at 35 °C and 45 °C, respectively. Al-Rashed (2011) [9] presented a comparable investigation of ferion (600a isobutane, 290 propane, 134a, 22, 410A, and 32) an optimized finned tube evaporator, and analyzes the effect of evaporator on the (COP) of system. For cycle simulation containing evaporator effects, the coefficient of performance of ferion 290 was better than that of ferion 22 by up to 3.5%, while the residual refrigerants performed about within a 2% coefficient of performance band of the ferion 22 baseline for the condensing temperatures considered.

While Ust, et al., (2011) [10] found that, ferion 32 demonstrates the best coefficient of exergetic performance among the other type of refrigerants (410A, 143A, 404A and 125). An experimental study into the effects of employing a dedicated mechanical sub cooling cycle, in terms of energy, with a residential 1500 Kg simple vapor compression refrigeration system is accomplished by Qureshi, et al., (2013) [11]. Ferion R22 is employed in the main cycle whereas ferion 12 is flowing in the dedicated sub cooling cycle. Also, the using of sub cooling cycle, the cycle efficiency for second-law improved about 21%. Additionally, the results showed that this percentage enhance is "inversely" proportional to the variation of ($T_{ambient}$).

Mohd Yunus, et al., (2016) [1] analyzed coefficient of performance, temperature distribution and the energy consumption of automotive air conditioning cycle system at different ambient temperature (30 °C, 35 °C and 40 °C), engine speed and internal heat load using ferion 134a (HFC) as the refrigerant. The results indicated that the automotive air-conditioning performance, reduces when internal heat load, speed of compressor, and ambient temperature increased.

Suhermanto, et al., (2016) [12] presented an experimental results of air conditioning system of an automotive charged with two refrigerants: ferion 1234yf and ferion 134a which were analyzed to determine the effect of rising ambient temperature using exergy and energy-based approaches. results indicated that the air conditioning system of automotive with refrigerant R1234yf produce comparative performances shown by comparably less cooling capacity, COP and higher mass flow rate than that of another refrigerant (R134a). The power absorbed and mass flow rates in the compressor became higher as increasing of the ambient temperature.

The effects of ambient air temperature and refrigerant charge amount on thermodynamic condition and performance of refrigerating cycle system in the split type air conditioner have been studied by Deymi Dashtebayaz, et al., (2018) [13]. It is revealed that energy efficiency ratio decreases about 30% as the increasing of ambient temperature from 27 to 45 °C. Agarwal, et al., (2019) [14] presented a thermodynamic investigation of mechanically sub cooled vapour compression refrigeration cycle. The investigation of the refrigeration cycle system has been achieved by utilizing very low GWP (from 1 to 4) refrigerants type viz. HFO- ferion 1234ze, ferion 1234yf and zero ODP to check the performance of another refrigerant (HFC- ferion 134a). The results reveals that the coefficient of performance of dedicated sub cooled vapour compression refrigeration system are higher than that of simple vapour compression refrigeration system. Also, refrigerant (ferion1234ze) performs superior than another refrigerant: ferion 1234yf and comparable to ferion 134a.

The goal of this study is to investigate the effect of rising ambient temperatures especially in the Middle East countries for twelve types of refrigeration fluids (R134a, R1234ze, R22, R12, R290, R1234yf, R152a, R245fa, R407C, R600a, R410A, and R507A) and the performance of environmentally friendly ferion.

2. ENERGY ANALYSIS

The main objective of the refrigeration cycle system is to transfer the heat outside the cooled space. VCR cycle is the most frequently used among the refrigeration cycles. A basic refrigeration cycle system involves of a compressor, an expansion device, condenser and an evaporator [10]. From the view of the first law point, the coefficient of performance (COP) is the measure of performance of the refrigeration cycle, which is expressed as the amount of net refrigeration effect yield per work unit required. It is explained as:

$$COP = \frac{Q_{evap}}{W_{comp}} \tag{1}$$

$$Q_{evap} = m_r\left(h_1 - h_4\right) \tag{2}$$

$$W_{comp} = m_r\left(h_2 - h_1\right) \tag{3}$$

To calculate the subcooling temperature:

$$\frac{1}{UA} = \frac{1}{\eta_{s.a} h_a A_a} + \frac{R_{fa}}{\eta_{s.a} A_a} + R_w + \frac{R_{fr}}{\eta_{s.r} A_r} + \frac{1}{\eta_{s.r} h_r A_r} \tag{4}$$

where,

R_f : fouling factor (°C/W)

R_w : wall resistance (°C/W)

$\eta_{s.a}$, $\eta_{s.r}$: surface efficiency for air and refrigerant side, respectively

h_a , h_r : average coefficient of heat transfer for air side and refrigerant side, respectively (W/m^{2}K)

Since the refrigeration surface efficiency is (1) and there are no fins on the refrigerant side of the tubes and neglecting the fouling factor and the wall resistance, then the overall coefficient of heat transfer is reduced to

$$\frac{1}{UA}=\frac{1}{\eta_{s.a}h_a A_a}+\frac{1}{h_r A_r} \tag{5}$$

A_r : area of heat transfer on the refrigerant side (m^2)
A_a : area of heat transfer on the air side (m^2)

$$\eta_{s.a}=1-\frac{A_{fin}}{A_o}\left(1-\eta_{fin}\right) \tag{6}$$

where,
A_{fin} : the fins surface area (m^2)
A_o : the heat transfer area of air side (involving fin and tube area) (m^2)
η_{fin} : the circular fin efficiency, it is defined as:

$$\eta_{fin}=\frac{\tan h\left(m_{es}R_\phi\right)}{\left(m_{es}R_\phi\right)}\phi \tag{7}$$

R : the radius of a circular fin (m)
ϕ : the fin efficiency parameter
m_{es} : the parameter of standard extended surface (m^{-1}) can be showed as:

$$m_{es}=\left(\frac{2h_a}{k_f\delta_f}\right)^{\frac{1}{2}} \tag{8}$$

k_f : the fin thermal conductivity (W/mL)
δ_f : thickness of fin (m)

The predictive model by Cavallini, et al. [15] was used for the two-phase portion of the condenser, due to it was developed to calculate the coefficient of heat transfer during condensation process inside smooth pipes working with blended halogenated or pure refrigerants. As well as, Cavallini model carry into consideration that changes in the flow system containing "annular", slug, "stratified" and wavy-transition flow.

$$h_{TP}=cp_l\frac{\left(\rho_l\tau\right)^{0.5}}{T^+} \tag{9}$$

$$T^+=\delta^+\times prn_l \qquad \delta^+\le 5 \tag{10}$$

$$T^+=5\times\left(Prn_l+\ln\left[1+Prn_l\times\left(\frac{\delta^+}{5}-1\right)\right]\right) \quad 5<\delta^+<30 \tag{11}$$

$$T^+=5\times\left(Prn_l+\ln\left[1+5Prn_l\right]+0.495\times\ln\left(\frac{\delta^+}{30}\right)\right) \quad 5<\delta^+<30 \tag{12}$$

$$Re_{Di}=\left(1-x\right)\frac{GD_i}{\mu_l} \tag{13}$$

$$\delta^+=\left(\frac{Re_{Di}}{2}\right)^{0.5} \qquad Re_{Di}\le 1145 \tag{14}$$

$$\delta^+=0.0504Re_{Di}^{\frac{7}{8}} \qquad Re_{Di}\le 1145 \tag{15}$$

$$\tau=\left(\frac{dp}{dz}\right)_f\frac{D_i}{4} \tag{16}$$

$$\left(\frac{dp}{dz}\right)_f=\phi_{Lo}^2\left(\frac{dp}{dz}\right)_f=\frac{2\phi_{Lo}^2F_{Lo}G^2}{D_i\rho_l} \tag{17}$$

$$\phi_{Lo}^2=E_{cav}+\left(\frac{1.262F_{cav}H_{cav}}{We^{0.1458}}\right) \tag{18}$$

$$E_{cav}=\left(1-x\right)^2+x^2\left(\frac{\rho_l F_{Go}}{\rho_G F_{Lo}}\right) \tag{19}$$

$$F_{cav}=x^{0.6978} \tag{20}$$

$$H_{cav}=\left(\frac{\rho_l}{\rho_v}\right)^{0.3278}\left(\frac{\mu_v}{\mu_l}\right)^{-1.181}\left(1-\frac{\mu_v}{\mu_l}\right)^{3.477} \tag{21}$$

$$We=\frac{G^2D_i}{\rho_v\sigma_l} \tag{22}$$

For $\frac{GD_i}{\mu_G}>2000$

$$F_{Lo}=0.046\left(\frac{GD_i}{\mu_l}\right)^{-0.2} \tag{23}$$

$$F_{Go}=0.046\left(\frac{GD_i}{\mu_v}\right)^{-0.2} \tag{24}$$

For $\frac{GD_i}{\mu_G}\le 2000$

$$F_{Lo}=\frac{16}{\left(\frac{GD_i}{\mu_l}\right)} \tag{25}$$

$$F_{Lo}=\frac{16}{\left(\frac{GD_i}{\mu_v}\right)} \tag{26}$$

where,
h_{TP} : the heat transfer coefficient of two phase convective (W/m^{2}K)
Prn_l : Prandtl number of the liquid phase $\left(\frac{\mu_l cp_l}{k_l}\right)$
τ : the shear stress of wall (N/m)
We : Weber number
ϕ_{Lo}^2 : friction multiplier of the two phases
F_{Lo} : the factor of fanning friction if liquid only flow in the pipe
F_{Go} : the factor of fanning friction if vapor only flow in the pipe.

In general, these "Equations" give the local coefficient of heat transfer. Therefore, the integral must take over the length of piping in the saturation area to achieve the average coefficient of heat transfer, so two-phase region was broken up into quality increments of 1% and the corresponding lengths of the various regions of the condenser is computed.

The study of "McQuiston and Parker" [16] is utilized to determine the convective "heat transfer coefficient" of air-side for pipe heat exchanger with multiple rows-depth shape of staggered pipes and a plain-fin. The "model" is developed for dry type coils. The coefficient of heat transfer is depended on the "Colburn j-factor", express as:

$$J_c = St_h pr_h^{\frac{2}{3}} \quad (27)$$

Substituting the suitable magnitudes for the "Stanton number" ($St_h = \frac{h_a}{Gcp}$) gives the following equation for the coefficient of air-side convective heat transfer.

$$h_{air} = \frac{J_c G_h cp_h}{pr_h^{2/3}} \quad (28)$$

where, G_h is air mass flux $\left(\text{kg/m}^2\text{sec}\right)$, can be expressed

$$G_h = \frac{m_h}{A_{mf,2}} \quad (29)$$

m_h: air mass flow rate (kg/sec)

$A_{mf,2}$: minimum free flow area that available on-air side (m^2), which can be expressed as:

$$A_{mf,2} = \left(W_{cond} H_{cond}\right) - \left(\delta_{fin} H_{cond} N_{fin}\right) - \left(W_{cond} D_o N_t\right) \quad (30)$$

W_{cond}: width of condenser (m)

H_{cond}: height of condenser (m)

δ_{fin}: fin thickness (m)

N_{fin}: number of fins

D_o: outside diameter of condenser tube (m)

N_t: number of tubes

Kadam, et al., [16] utilized a heat exchanger pipe with 4 depth-rows and plain-fin as the baseline equation, and for this equation the Colburn *j*-factor defined is as:

$$j_{c,4} = 0.2675 \times \left(J_p\right) + 1.325 \times 10^{-6} \quad (31)$$

$$J_p = Re_{Do}^{-0.2} \times \left(\frac{A_o}{A_t}\right)^{-0.15} \quad (32)$$

where,

A_t: the surface area of tube outside (m^2),

A_o: the total surface area of air side heat transfer (fin area + tube area - area under fins) (m^2),

Re_{Do}: "Reynolds number" is depend on the outside diameter of pipe, D_o, and the mass flux of air.

The area ratio can be presented as:

$$\frac{A_o}{A_t} = \frac{4}{\pi} \frac{X_l}{D_h} \frac{X_t}{L_c} \Omega \quad (33)$$

The hydraulic diameter D_h, and Ω are the minimum free-flow area to the front area ratio, which are expressed

$$D_h = \frac{4 \times A_{mf,2} \times L_{cond}}{A_o} \quad (34)$$

$$\Omega = \frac{A_{mf,2}}{A_{fr}} \quad (35)$$

where, A_{fr} is the frontal area (m^2).

The heat exchangers j-factor with four or lower rows-depth can then be determined using the following relationship

$$\frac{J_{c,z}}{J_{c,4}} = \frac{1 - 1280 \times (z) \times \left(Re_{x_l}\right)^{-1.2}}{1 - 1280 \times (4) \times \left(Re_{x_T}\right)^{-1.2}} \quad (36)$$

where, z is the number of rows-depth of pipes (m) and Re_{x_l} is Reynolds number of the air-side depend on the longitudinal pipe spacing.

$$Re_{x_l} = \frac{G_a X_l}{\mu_a} \quad (37)$$

For a heat exchanger of cross flow with unmixed fluids, the effectiveness based on heat capacity ratio and number of transfer unit (NTU) and is expressed as:

$$\varepsilon = 1 - \exp\left\{\left(\frac{1}{C_T}\right) NTU^{0.22} \left[\exp\left(-C_r \left(NTU^{0.78}\right)\right) - 1\right]\right\} \quad (38)$$

where, C_r is heat capacity ratio defined by

$$C_r = \frac{c_{\min}}{c_{\max}} \quad (39)$$

The effectiveness can also be related to a quantity known as (NTU), basing on the flow configuration, expressed as:

$$NTU = \frac{UA}{c_{\min}} \quad (40)$$

where, UA is the coefficient of "overall heat transfer" U times heat transfer area (W/K)

$$\varepsilon = \frac{\dot{Q}}{Q_{\max}} \quad (41)$$

$$Q_{\max} = C_{\min}\left(T_{h,i} - T_{c,i}\right) \quad (42)$$

Then to find the sub cooling at the condenser outlet from the enthalpy at the condenser outlet:

$$Q_{cond} = m_r\left(h_2 - h_3\right) \quad (43)$$

3. RESULTS AND DISCUSSION

According to the results by using Engineering Equation Solver (EES), the comparison between the twelve types of refrigeration fluids (R134a, R1234ze, R22, R12, R290, R1234yf, R152a, R245fa, R407C, R600a, R410A, and R507A) under different outside temperature varied from 35 °C to 55 °C. The best performance can be obtained with R245fa where the coefficient of performance (COP) reached to 4.753 at 35 °C. R245fa can be used as refrigerant in new centrifugal chillers, replacing R123 and R22. Figure 1 displays that, the coefficient of performance of R245fa is higher than R22 by 1.7 and the performance decreases by 7% approximately with the increment of ambient temperature. The performance of R410A and R507A show that these gases do not effective at the higher ambient temperature, while R290 has the lower performance.

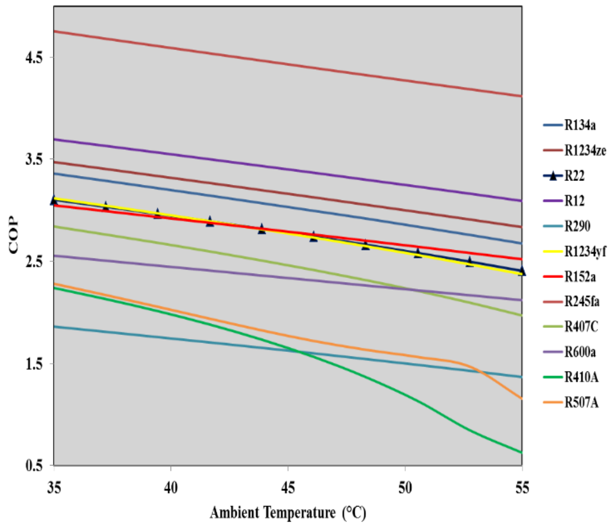


Figure 1. The COP for ferions at different ambient temperatures

The ferion R245fa which has the best performance absorbed the lower power in compressor as shown in Figure 2(a). The ambient temperature has a little effect on compressor work reached to 2.5% increment with the increase of ambient temperature. Figure 2(b) shows that, R152a has the higher refrigeration capacity, while R410A and R507A refrigeration capacity dropped with the rising of ambient temperature. So that R152a considered a good alternative for R134a for higher ambient temperature, which has 30% increment in capacity over R134a. The subcooling temperature of ferions has been illustrated by Figure 3 for different ambient temperature. R12 has the higher subcooling temperature reached to 14.68 °C at 35 °C ambient temperature, while the ferions (R290 and R600a) do not work with higher ambient temperature.

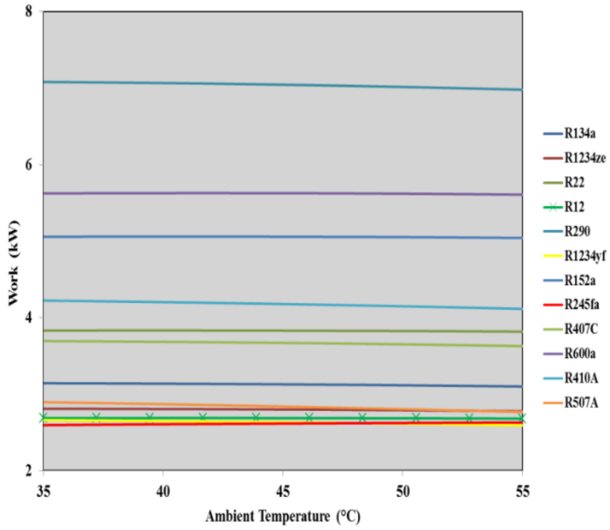
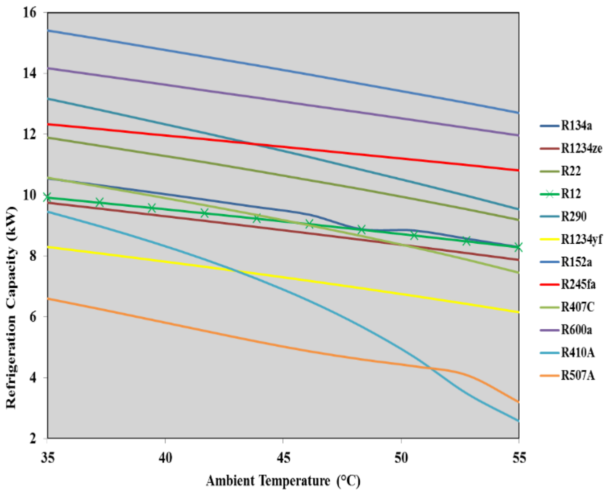


Figure 2. (a) The work of compressor for ferions at different outside temperatures



(b)

Figure 2. (a) The work of compressor for ferions at different outside temperatures, (b) The refrigeration capacity for ferion at different outside temperatures

The ferion R152a and R410A as shown in Figures 4 and 5 has been more efficient for the ambient temperature less than 50 °C. Figures 6 and 7 shows that, R600a and R290 the ferion cannot be condensate and reached to saturated liquid for the all outside temperature from 35 °C to 55 °C.

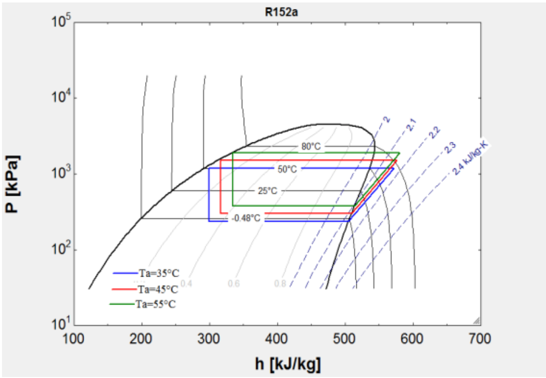


Figure 4. P-h diagram of R152a

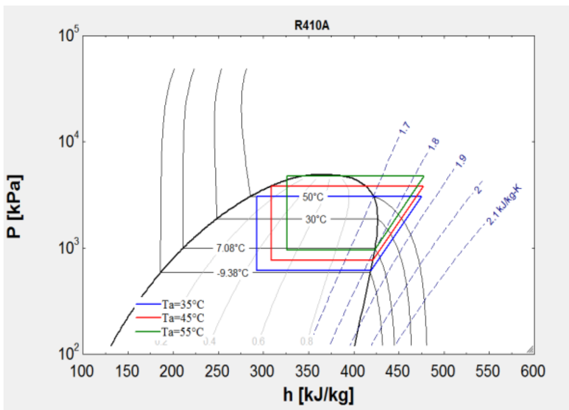


Figure 5. P-h diagram of R410A

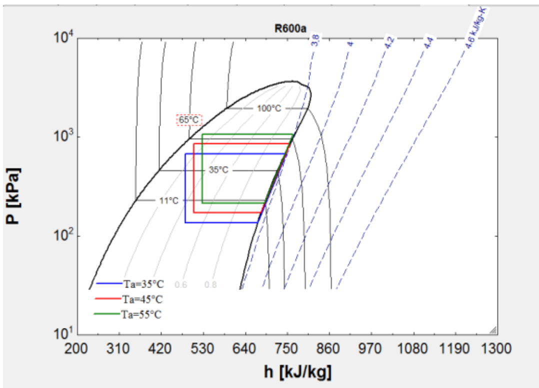


Figure 6. P-h diagram of R600a

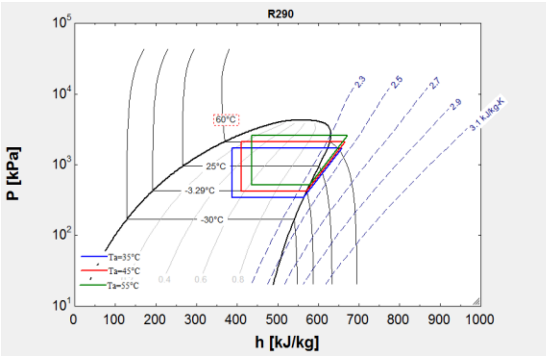


Figure 7. P-h diagram of R290

The ferion R507A has been found that more efficient for the ambient temperature less than 50 °C, as shown in Figure 8. The sub cooling value increases with the increment of ambient temperature for ferion R407C, as shown in Figure 9. While Figure 10 showed that ferion R245fa was efficient for all ambient temperatures 35 °C, 45 °C, 55 °C and for its higher COP values, it has been considered the best choice for new refrigeration systems. R134a has a stable performance with different ambient temperature, as shown in Figure 11. So that R134a, has been considered a best choice instead of R12 for the same performance, as shown in Figure 14. Recently, and due to the growing concerns about the relatively high global warming potential of current ferions R134a can be replaced by R1234yf and R1234ze with same performance, as shown in Figures 12 and 13.

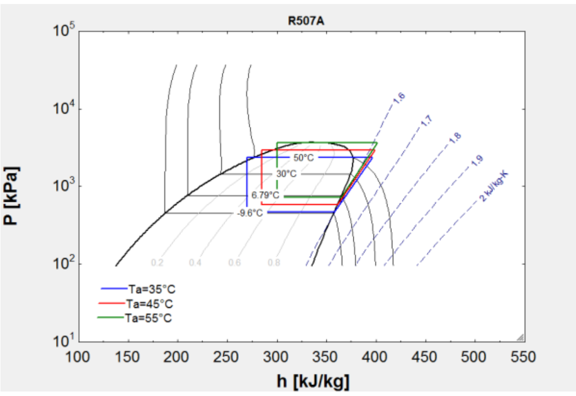


Figure 8. P-h diagram of R507A

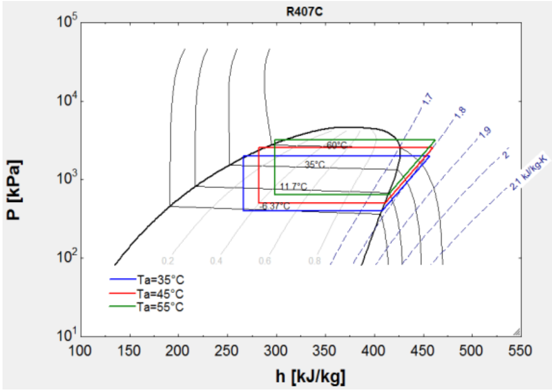


Figure 9. P-h diagram of R407C

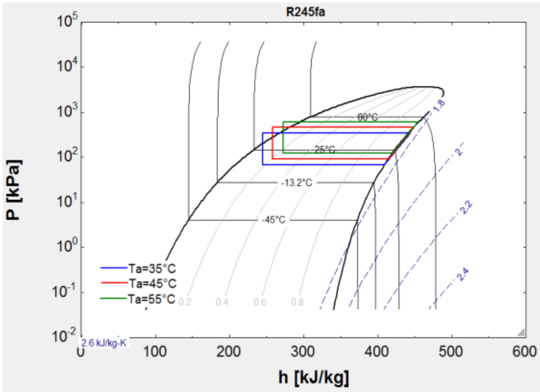


Figure 10. P-h diagram of R245fa

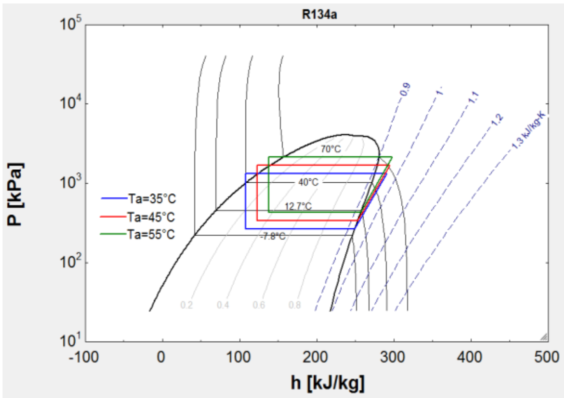


Figure 11. P-h diagram of R134a

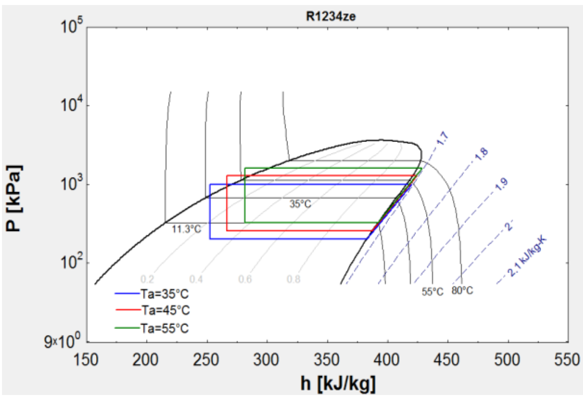


Figure 12. P-h diagram of R1234ze

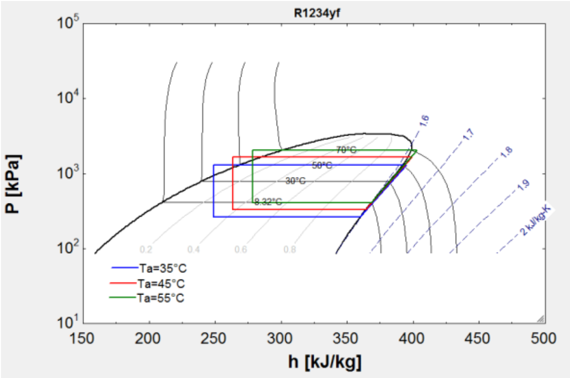


Figure 13. P-h diagram of R1234yf

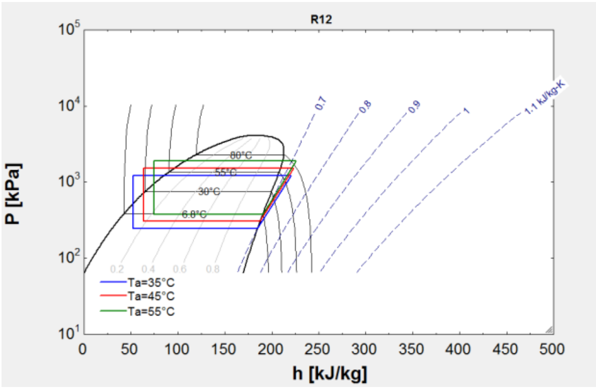


Figure 14. P-h diagram of R12

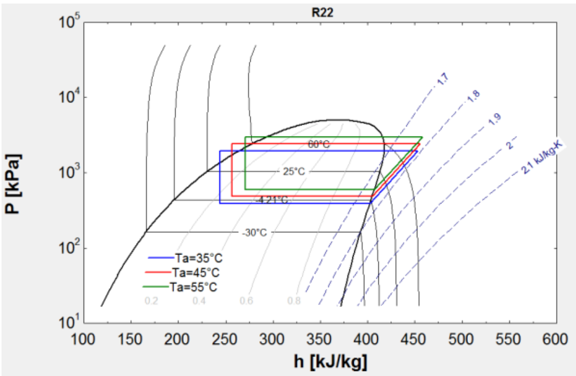


Figure 15. P-h diagram of R22

4. CONCLUSIONS

A numerical study of vapour compression system performance with twelve types of ferion (R134a, R1234ze, R22, R12, R290, R1234yf, R152a, R245fa, R407C, R600a, R410A, and R507A) under different ambient temperature varied from 35 °C to 55 °C studied in the present work. The numerical results showed that, the best performance can be obtained with R245fa which reached to 4.753 at 35 °C. The performance of R410A and R507A show that these gases do not effective at the higher ambient temperature, while R290 has the lower performance.

The ambient temperature has a little effect on compressor work reached to 2.5% increment with the increase of ambient temperature. R600a and R290 the ferion cannot be condensate and reached to saturated liquid for the all-ambient temperature from 35 °C to 55 °C. R410A and R507A do not effective at higher ambient temperature. R134a can be replaced by R1234yf and R1234ze with same performance due to the growing concerns about the relatively high global warming potential of current ferion.

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