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ENERGY PERFORMANCE OF ENVIRONMENTAL-FRIENDLY R435A AND R161 REFRIGERANTS IN SUB-COOLING REFRIGERATION SYSTEMS

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The synthetic chlorofluorocarbon (CFC) and hydro-chlorofluorocarbon (HCFC) refrigerants are being phased out due to their high Ozone depletion Potentials (ODPs) and Global Warming Potentials (GWPs). In this study, the effects of sub-cooling on the energy performance of two eco-friendly refrigerants (R435A and R161) with zero ODP and negligible GWP are investigated theoretically and compared with those obtained using R413A. The results obtained showed that the thermodynamic properties of R435A matched those of R413A and it could be used as substitute for R413A. The two alternative refrigerants (R435A and R161) exhibited very high refrigerating effects and coefficient of performance (COP) with the average COPs of 9.17 and 7.68 % respectively higher than that of R413A. They also exhibited lower power per ton of refrigeration (PPTR), but R435A emerged as the most energy efficient refrigerant with the highest COP and lowest PPTR. Generally, incorporation of sub-cooling heat exchanger enhanced the system's performance.
Key words: sub-cooling, performance, eco-friendly, refrigeration system.

Energetska učinkovitost ekološko prihvatljivih R435A i R161 radnih tvari u pothlađenim rashladnim sustavima. Sintetske radne tvari, klorofluorougljik (CFC) i hidro-klorofluorugljik (HCFC), se postupno ukidaju zbog njihovog visokog potencijala na oštećenje ozonskog omotača (ODPs) i potencijala na globalno zagrijavanje (GWPs). U ovom radu teoretski su istraživani učinci pothlađivanja na energijsku učinkovitost dviju eko-prihvatljivih radnih tvari (R435A i R161) s nula ODP i zanemarivog GWP koji su uspoređeni s onima dobivenim upotrebom R413A. Dobiveni rezultati su pokazali da termodinamička svojstva R435A odgovaraju onima R413A i mogu se koristiti kao zamjena za R413A. Dvije alternativne radne tvari (R435A i R161) pokazuju vrlo visoki rashladni efekt i koeficijent učinkovitosti (COP) s prosječnim COP-om od 9,17% za R435A i 7,68% za R161 više od R413A. Oni također imaju nižu snagu po toni hlađenja (PPTR), ali R435A se pokazao kao energetski najučinkovitije sredstvo za hlađenje s najvišim COP-om i najnižim PPTR-om. Općenito, ugradnja pothlađivanog izmjenjivača topline poboljšava učinkovitost sustava.
Ključne riječi: pothlađivanje, učinkovitost, eko-prihvatljiv, rashladni sustav.

INTRODUCTION

Refrigeration and air-conditioning systems represent a key area of energy usage in both modern urban life and rural households [1]. Energy costs and environmental concerns have made energy optimisation a critical issue for households. Energy efficiency is a prime mover in reducing global warming emissions. The demand for energy has increased due to industrial development and the improvement of living standards. This demand has so far been principally supplied by fossil fuels such as coal, gas and oil. However, energy consumption has produced global environmental concerns, with prevalent calls to decrease greenhouse gas emissions and improve the energy efficiency of engineering systems [2, 3].

The ozone depleting and global warming issues are currently the most serious global environmental problems and they are the most significant conditions in the development of new refrigerants [4, 5]. Since the recognition of the ozone depletion caused by the reaction of chlorine containing Chlorofluorocarbons (CFCs) and Hydro-chlorofluorocarbons (HCFCs) with the ozone layer in the stratosphere, the international community decided to limit the production of CFC and HCFC refrigerants and to ultimately ban them. In agreement with the Montreal protocol, sustainable solutions have been obtained to solve the ozone-depletion issue [6, 7].

Several chemicals were considered as substitute refrigerants and the most prominent among them are the Hydro-fluorocarbons (HFCs) and their blends which have zero Ozone Depletion Potential (ODP) such as R134a, R413A, R404A, R407C and R410 [8, 9]. Since the introduction HFC refrigerants, they have become the most widely utilized refrigerants, employed in stationary and automotive air-conditioning systems as well as in refrigeration and heat pump systems. Consequently, HFC refrigerants which were once regarded as eco-friendly are now considered as one of the six target greenhouse gases under Kyoto protocol for their contribution to greenhouse gases, while they were entirely harmless to ozone layer, most of them did have high Global Warming Potential (GWP) [10, 11].

R134a and R413A were developed as alternative refrigerants to ozone depleting R12. R134a is an HFC refrigerant and does not deplete the ozone layer but it does have properties that contribute to global warming (greenhouse effect). R413A is a zeotropic mixture of 88% of R134a, 9% of R218 and 3% of R600a by mass. It is a drop-in substitute for R12 and R134a in refrigeration systems because it is compatible with mineral oil and polyol-ester (POE) oil used respectively in the compressors of both systems. According to Padilla *et al.* [8], the performance of R413A in terms of power consumption, irreversibility and exergy efficiency is better than that of R12.

R413A has an ozone depleting property of zero, therefore, it does not damage the ozone layer, but it has high global warming potential similar to R134a. This issue of global warming is very serious since it threatens safety and benefit of our lives. According to Kayukawa [12], the rise of the temperature may cause the polar glacier melt, to result in the elevation of the ocean level. This will destroy uncountable coastlines in the world and take away the bulk of the low sea-level islands. It may also turn many regions into deserts when the speed of the climate change is faster than that for migration of the vegetation zones.

Consequently, besides zero ODP, the working fluids in refrigeration systems must also have very low GWP [13, 14]. In developed countries, refrigerants with high GWP are been phased out. The European Union's F-gas regulation and the directive 2006/40/EC ban fluorinated gases having GWP greater than 150 in new mobile models from January 1, 2011 and in new vehicles from January 1, 2017[15]. In line with this new constraint, once again, refrigeration industries will be forced to change refrigerants. This time from the newly introduced chlorine-free alternative refrigerants to those do not absorb the infrared re-radiation from the earth's surface. Also, the improvement of the system's efficiency is essential so that the new refrigerants do not result in additional CO_2 generation at the power source.

A sub-cooling heat exchanger is normally incorporated in a refrigeration system to facilitate heat transfer between the hot liquid refrigerant at the exit of the condenser with the cold vapour refrigerant leaving the evaporator. This causes a high quality of refrigerant to enter the evaporator. As a result of adding the sub-cooling heat exchanger to the basic cycle the sub-cooling upgrades significantly and it is also helps to protect the compressor from two-phase flow [16]. Many studies [17-19] have shown that cooling of liquid refrigerant leaving condenser can significantly reduce the power consumption and improve the system COP. Therefore, in the present study, sub-cooling effects on the energy performance of eco-friendly R161 with GWP of 12 and R435A an azeotropic refrigerant mixture composed of RE170 and R152a (80 and 20 % in weight, respectively) with GWP of 27 (Table 1) were evaluated theoretically using a sub-cooling heat exchanger vapour compression refrigeration system and compared with performance of R413A in the system.

Table 1. Environmental and thermophysical properties of investigated refrigerants [20, 21]
Tablica 1. Okolišna i termofizikalna svojstva ispitivanih radnih tvari [20, 21].

Properties	Refrigerants			
	R134a	R413A	R161	R435A
Molar mass (kg/kmol)	102.03	103.95	48.06	49.04
Normal boiling point (^oC)	-26.07	-33.40	-37.10	-26.10
Critical temperature (^oC)	101.06	96.60	102.02	125.20
Critical pressure (MN/m^2)	4.06	4.02	4.70	5.39
ODP	0	0	0	0
GWP)	1430	2100	12	27

MATERIALS AND METHODS

Analysis of refrigeration cycle with sub-cooling heat exchanger

Higher proportion of the world's modern refrigeration, air-conditioning and heat pump systems work on the principles of vapour compression refrigeration system. These systems use circulating fluid refrigerants that undergo a closed refrigeration cycle through a series of evaporation, compression, condensation, throttling and expansion processes. The refrigerant absorbs heat from a lower-temperature reservoir and releases the heat to a higher-temperature reservoir in such a way that the final state is equal in all respects to the initial state. In a vapour compression cycle (Fig. 1), the circulating refrigerant enters the compressor as low-pressure vapour and exits the compressor as high-pressure superheated vapour.

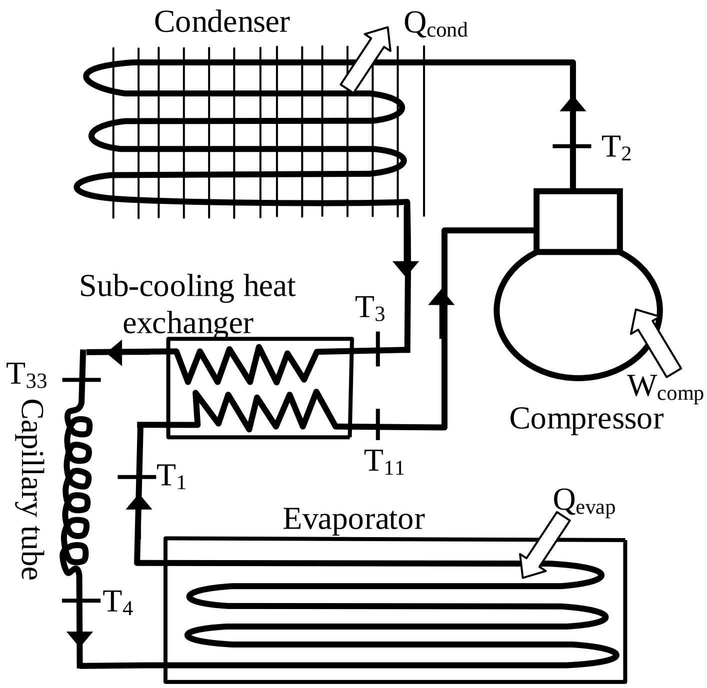

Figure 1. Refrigeration system with a sub-cooling heat exchanger
Slika 1. Rashladni sustav s pothlađenim izmjenjivačem topline

Refrigerant gives up its latent heat to the surrounding condensing medium and turns to liquid while passing through the condenser. The high-pressure liquid refrigerant from the condenser is forced through an expansion device that regulates the flow and reduces the pressure to low-pressure existing in the evaporator. In the evaporator, the liquid refrigerant vaporizes by absorbing latent heat from the material being cooled, and the resulting low pressure vapour refrigerant then passes from the evaporator to the compressor.

Sub-cooling in refrigeration involves cooling the liquid refrigerant leaving the condenser at uniform pressure, to a temperature that is less than the saturation temperature, which corresponds to condenser pressure. It increases the amount of heat extracted from the evaporator without increasing the power input to the compressor [22]. Generally, sub-cooling is accomplished using a water cooled heat exchanger or by employing a liquid-suction vapour heat exchanger. These sub-cooling methods improve efficiency and cooling capacity without adding moving parts. The schematic diagram of vapour compression refrigeration system with a sub-cooling heat exchanger is shown in Fig. 1. In this system, the vapour leaving the evaporator is heated up by the condensate, the temperature of condensate decreases from T_3 to T_{33} and the vapour is superheated before suction (the temperature increased from T_1 to T_{11}). The sub-cooling heat exchanger is essentially a concentric type counter-flow heat exchanger which causes the sub-cooling of the liquid refrigerant before throttling. The ideal vapour compression refrigeration system with sub-cooling heat exchanger is shown on the Pressure-Enthalpy diagram in Fig. 2.

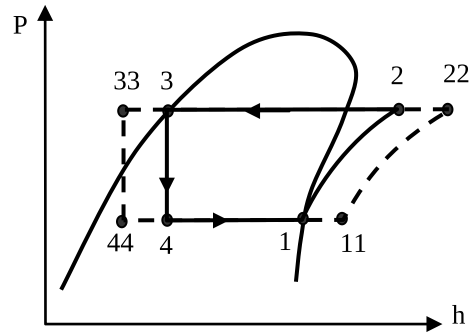

Figure 2. Pressure-enthalpy diagram showing effect of an idealized sub-cooling
Slika 2. Dijagram tlak-entalpija prikazuje učinak idealiziranog pothlađivanja

Considering the cycle on p-h diagram in Fig. 2, without a sub-cooling heat exchanger, the heat absorbed by the refrigerant in the evaporator or refrigerating effect per unit mass flow rate of circulating refrigerant is the difference in enthalpy between states 1 and 4. After the heat exchanger is installed, the refrigeration effect per unit mass flow rate increases to the difference in enthalpy between states '1' and '44'. Therefore, the refrigerating effect without sub-cooling (Q_{evap}, kJ/kg) is calculated as:

$$Q_{evap} = (h_1 - h_4) \quad (1)$$

where, h_1 = specific enthalpy of refrigerant at the outlet of evaporator (kJ/kg); and h_4 = specific enthalpy of refrigerant at the inlet of evaporator (kJ/kg). The refrigerating effect with sub-cooling (Q'_{evap}, kJ/kg) is calculated as:

$$Q'_{evap} = (h_1 - h_{44}) \quad (2)$$

where, h_{44} = specific enthalpy of refrigerant at the inlet of evaporator after passing through the heat exchanger and the capillary tube (kJ/kg). The compressor work input of the system without heat exchanger (W_{comp}, kJ/kg) is obtained as:

$$W_{comp} = (h_2 - h_1) \quad (3)$$

where, h_2 = specific enthalpy of refrigerant at the outlet of compressor (kJ/kg). The compressor work input of the system with heat exchanger (W'_{comp}, kJ/kg) is obtained as:

$$W'_{comp} = (h_{22} - h_{11}) \quad (4)$$

where, h_{11} = specific enthalpy of refrigerant at the inlet to compressor after superheated by the heat exchanger (kJ/kg); h_{22} = specific enthalpy of refrigerant at the outlet of compressor of the system with heat exchanger (kJ/kg). The pressure ratio (P_R) of the cycle is obtained as:

$$P_R = \frac{P_{cond}}{P_{evap}} \quad (5)$$

where, P_{cond} = absolute condensing pressure (MPa) and P_{evap} = absolute evaporating pressure (MPa). The Coefficient of Performance (COP) is the refrigerating effect produced per unit of work required; therefore, COP is obtained as the ratio of Eq. (1) to Eq. (3) for the system without sub-cooling and ratio of Eq. (2) to Eq. (4) for the system with sub-cooling:

Without sub-cooling,

$$COP_{ref} = \frac{Q_{evap}}{W_{comp}} \quad (6)$$

With sub-cooling,

$$COP_{ref} = \frac{Q'_{evap}}{W'_{comp}} \quad (7)$$

Determination of Thermodynamic Properties of Refrigerants

Thermodynamic and transport properties of refrigerants are necessary for predicting system behaviour and performance of refrigeration components. The pressure, volume and temperature (PvT) in an equilibrium state are the most important thermal properties that are needed for the prediction of a refrigerant system's performance. Other properties, such as enthalpy and entropy as well as the Helmholtz and Gibbs functions, may be derived from a PvT correlation utilizing specific heat [14]. The most extensively used refrigerant database software known as REFPROP was developed and maintained by the National Institute of Standards and Technology (NIST) using combinations of several equations-of-state to correlate various single component refrigerants and predefined mixtures, along with the ability to construct virtually any desired mixture of up to five components [23]. This software was employed in this work to compute the properties of the investigating refrigerants.

RESULTS AND DISCUSSION

Fig. 3 shows the saturation vapour pressure curves for R413A and its two investigated alternative refrigerants (R435A and R161). As shown in this figure, the saturation vapour pressure curve for R435A is very close to that of the base refrigerant (R413A) with average value of the R435A between the temperature range of -30 to 40 ^{o}C deviated by 8.86 % lower than that of the R413A. This indicates that this refrigerant can exhibit similar properties. The average saturation vapour pressure for R161 between the same temperature range deviated by 31.13 % higher than that of R413A.

Fig. 4 shows the discharge pressures at condensing temperature of 40 ^{o}C for the three investigated refrigerants. The discharge pressure is a vital factor that affects the performance of a refrigerating system. It affects the stability of the lubricant and compressor components. Therefore, low discharge pressure is beneficial to the system's performance. R435A and R161 exhibited close discharge pressure with R413A. The lowest discharge pressure was obtained using R435A in the system. The average discharge pressures for R435A and R161 are 15.66 % lower and 24.57 % higher respectively than that of R413A.

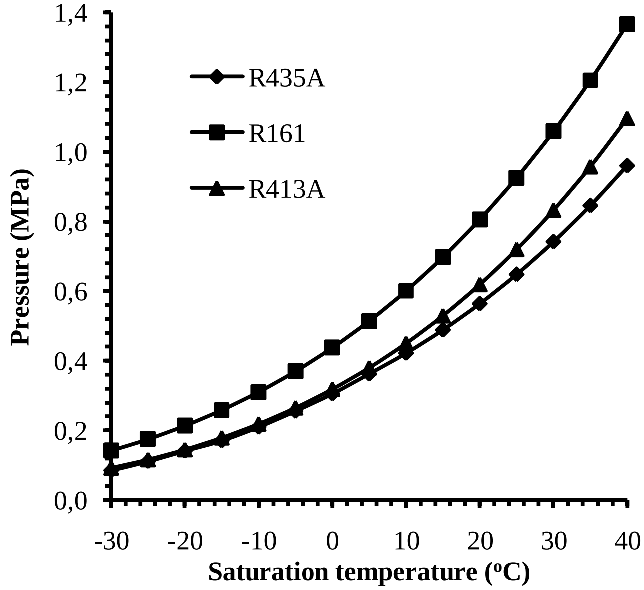

Figure 3. Saturation vapour pressure curves
Slika 3. Krivulje tlaka zasićene pare

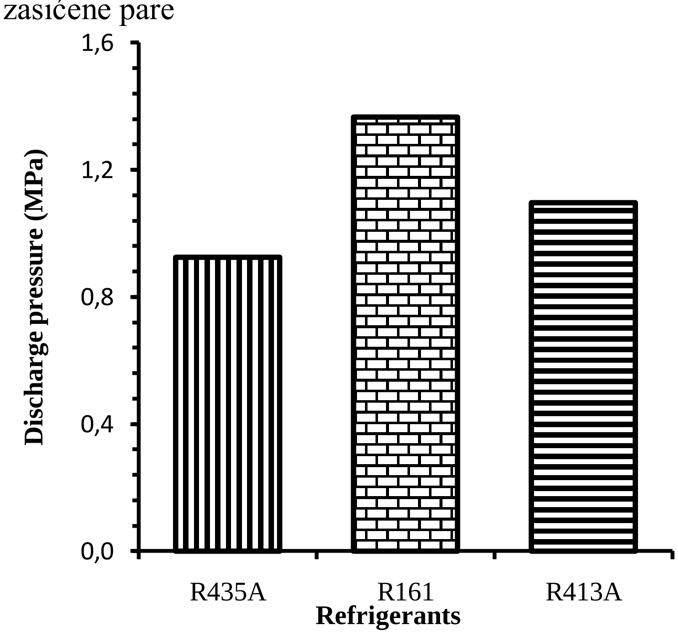

Figure 4. Discharge pressure at condensing temperature of 40 ^{o}C
Slika 4. Izlazni tlak pri temperaturi kondenzacije od 40 ^{o}C

The compressor pressure ratios for R413A and the two alternative refrigerants (R435A and R161) at condensing temperature of 40 ^{o}C are shown in Fig. 5. Compressor pressure ratio is one of the conditions use for choosing suitable alternative to any refrigerant in refrigeration system. Refrigerants with lower pressure ratio are more suitable and better than those with high pressure ratio, because high pressure ratio is detrimental to the performance of the system. As shown in Fig. 5, the pressure ratios for R435A and R161 are lower than that of the baseline refrigerant (R413A) with average values of 5.56 and 15.23 % respectively.

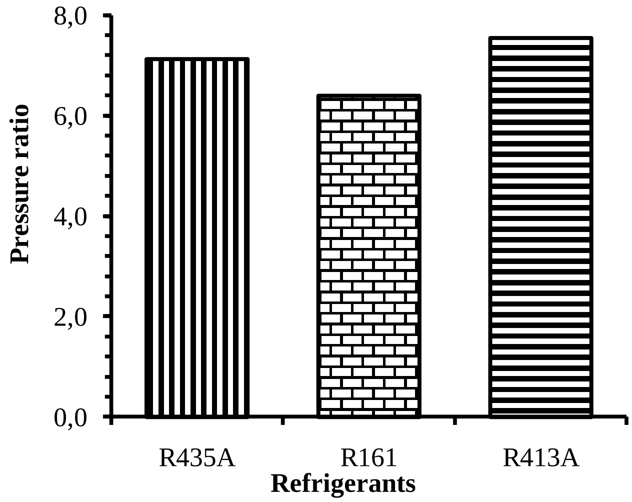

Figure 5. Pressure ratio at condensing temperature of 40 °C
Slika 5. Omjer tlaka pri temperaturi kondenzacije od 40 °C

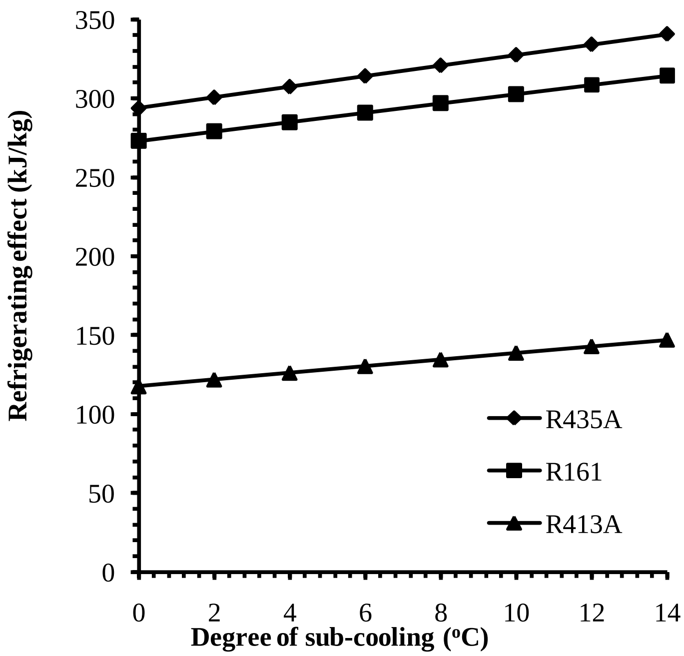

Figure 6. Variation of refrigerating effect with the degree of sub-cooling at 40 °C condensing and -20 °C evaporating temperatures
Slika 6. Promjena rashladnog efekta sa stupnjem pothlađivanja pri temperaturi kondenzacije na 40 °C i temperaturi isparavanja na -20 °C

Fig. 6 shows the variation of refrigerating effects with the degree of sub-cooling at 40 °C condensing and -20 °C evaporating temperatures for R413A, R435A and R161 refrigerants. As shown in the figure, the refrigerating effect increases as the degree of sub-cooling increases for all the investigating refrigerants. The latent heat of refrigerant increases as its degree of sub-cooling increases and hence, the refrigerating effect increases. Very high latent heat energy is desirable since the mass flow rate per unit of capacity is less. When the latent value is increased, the energy efficiency and capacity of the compressor are significantly increased. R435A and R161 exhibited much higher refrigerating effect than R413A as clearly shown in Fig. 6. The average refrigerating effects obtained using the alternative refrigerants (R435A and R161) are 184.9 and 161.4 kJ/kg respectively higher than that of R413A. The effects of the degree of sub-cooling on the power consumption per ton of refrigeration at 40 ^{o}C condensing and -20 ^{o}C evaporating temperatures is shown in Fig. 7. The figure revealed that the power per ton of refrigeration (PPTR) reduces as the degree of sub-cooling increases for all the investigating refrigerants. The figure also showed that reduction in power consumption increases as the degree of sub-cooling increases. This result has revealed the two alternatives (R435A and R161) as energy efficient refrigerants. They exhibited lower PPTR with the average values of 9.32 and 7.99 % lower than that of R413A, respectively.

Fig. 8 shows the influence of degree of sub-cooling on the coefficient of performance (COP) at 40^{o}C condensing and -20^{o}C evaporating temperatures for R413A and the two alternative refrigerants. This figure clearly shows the effect of sub-cooling on the refrigerant performance; the COP increases with increase in the degree of sub-cooling. Again, the two alternative refrigerants exhibited high performance than the baseline refrigerants. The average COPs obtained for R435A and R161 were 9.17 and 7.68 % respectively higher than that of R413A.

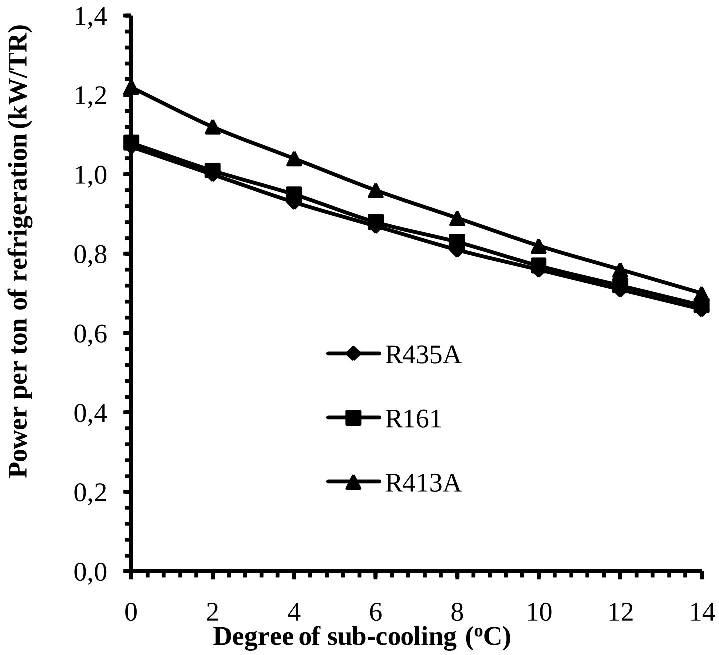

Figure 7. Variation of power per ton of refrigeration (PPTR) with the degree of sub-cooling at 40 ^{o}C condensing and -20 ^{o}C evaporating temperatures
Slika 7. Promjena snage po toni rashlađivanja (PPTR) sa stupnjem pothlađivanja pri temperaturi kondenzacije na 40 ^{o}C i temperaturi isparavanja na -20 ^{o}C

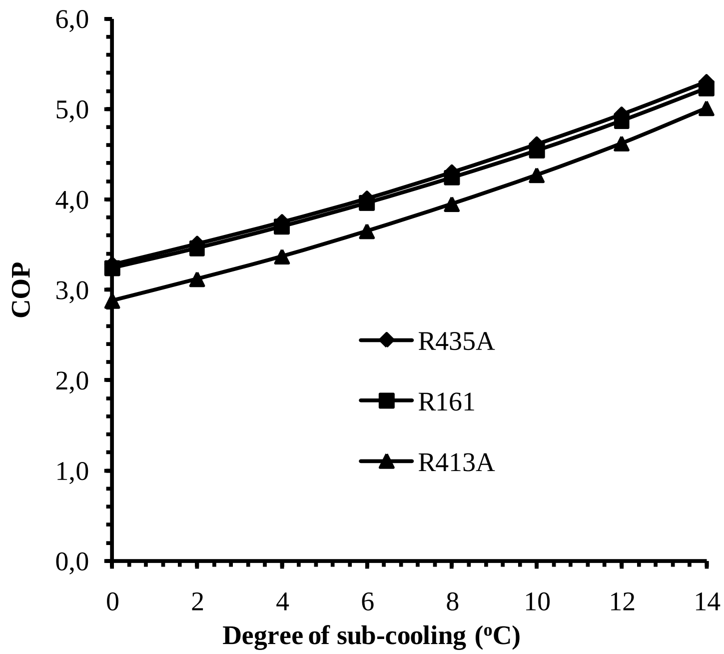

Figure 8. Variation of coefficient of performance (COP) with the degree of sub-cooling at 40 ^oC condensing and -20 ^oC evaporating temperatures

Slika 8. Promjena koeficijenta učinkovitosti (COP) sa stupnjem pothlađivanja pri temperaturi kondenzacije na 40 ^oC i temperaturi isparavanja na -20 ^oC

CONCLUSION

The performance of R435A and R161 as alternatives to R413A in a refrigeration system with sub-cooling at condensing temperature of 40 ^oC and evaporator temperature of -20 ^oC was investigated theoretically. The following conclusions are drawn from the analysis and discussion of the results:

(i) The two alternative refrigerants (R435A and R161) are more environmentally-friendly than R413A. The saturated vapour pressure for R435A is very close to that of R413A, which indicates similar properties, while the saturated vapour pressure for R161 between the temperature range of -30 to 40^oC deviated with 31.13 % from that of R413A.

(ii) R435A and R161 exhibited low pressure ratio with average values of 5.56 and 15.23 % respectively lower than that of R413A.

(iii) R435A and R161 showed higher refrigerating effect than R413A. Therefore, lower mass of R435A and R161 refrigerants will be required to produce the same capacity with R413A.

(iv) Energy performance of the two alternative refrigerants was better than that of R413A. They exhibited lower power per ton of refrigeration (PPTR) but R435A emerged as the most energy efficient refrigerant among all the investigated refrigerants with average PPTR of 9.32 % lower than that of R413A.

(v) The two alternative refrigerants showed higher coefficient of performance (COP) than R413A, which shows high efficiency and low operating cost for the systems working with these alternative refrigerants. The highest coefficient of performance (COP) of the system was obtained using R435A.

Generally, R435A and R161 performed better than R413A; their specific power consumptions were less than that of R413A. Therefore, they can be used as alternative refrigerants in the existing R413A refrigeration systems. The best performance is obtained using R435A in the system. Incorporation of sub-cooling heat exchanger in the refrigeration system significantly boosted the performance of the system; it increases the COP, reduces the pressure and the specific power consumption of the system.

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