Effect of Sub-Cooling on the Performance of a Reftrofitted Domestic Refrigerator Using Eco-Friendly Refrigerants

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***Abstract* - Stratospheric ozone depletion and the atmospheric greenhouse effect due to refrigerant emissions have led to drastic changes in the refrigeration and air-conditioning technology. Chlorofluorocarbon (CFC) and Hydro-chlorofluorocarbon (HCFC) refrigerants, which are the major causes of global warming and ozone depletion, have been scheduled for total phase out. In this study, the effects of sub-cooling on the performance of two hydrocarbon refrigerants (R510A and R600a) were investigated. The sub-cooling heat exchanger was incorporated into an existing R134a refrigerator to improve its performance. The system was incorporated with two pressure gauges at the inlet and outlet of the compressor to measure the suction and discharged pressures. The refrigerant temperatures at the inlet of the evaporator, compressor, condenser and expansion valve of the system were measured with thermocouple. The thermodynamic properties of the refrigerants were determined using REFPROP software. The results obtained showed that R600a and R510A exhibited higher refrigerating effect than R134a at sub-cooling temperatures ranging from 0 to10ºC. The refrigerating effects of 220.66, 286.86 and 108.70 kJ/kg were obtained using R600a, R510A and R134a respectively at sub-cooling temperature of 7ºC. The highest refrigerating effect (289.30 kJ/kg) was obtained using R510A at 10ºC sub-cooling. The compressor work inputs were 88.96 and 85.90 kJ/kg for R600a and R510A respectively at 5ºC sub-cooling temperature, while the value of R134a was 49.71 kJ/kg at the same sub-cooling temperature. The sub-cooling increased the Coefficient of Performance (COP) of all three refrigerants with the highest COP of 3.8 obtained using R510A at sub-cooling temperature of 9ºC. The COPs of R510A and R134a were 1.08 and 0.18 higher than that of R600a respectively. Finally, sub-cooling enhanced the system performance by reducing the compressor work input and increasing the system refrigerating effect.**

Keyword: Sub-ooling, Cop, Hydrocarbon, Refrigerant, thermocouple, Chlorofluorocarbon, Hydro chlorofluorocar

1.0 INTRODUCTION

Majority of air-condition, refrigeration and heat pump systems are running at the vapour compression cycle that rely on the performance of refrigerants which can be safe, chemically stable, have desirable thermodynamic and thermo physical properties. Chlorofluorocarbons (CFCs) and hydro- chlorofluorocarbons (HCFCs) were used notably as refrigerants in those structures for the reason that Thirties because of their tremendous thermodynamic, thermo physical and safety houses (Pham and Sachs, 2010).

Since 1987, refrigerants had been experiencing varying demanding situations because of their global environmental issues. These chlorine containing fluorinated hydrocarbon refrigerants (CFCs and HCFCs) had been located to diffuse up into the stratosphere. The chlorine content material of the refrigerants become the important purpose of destruction of the stratospheric ozone which absorbs the solar's excessive power ultraviolet rays and protects each human and different dwelling matters from publicity to ultraviolet radiation. This chance is represented by means of the refrigerant Ozone Depletion potential (ODP) wide variety (McMullan, 2002; Padilla et al., 2010; Kim et al., 2011)

Refrigeration and air-conditioning industries, like other technology primarily based industries, are driven via new technologies, environmental and economic factors. growing difficulty over international Warming ability (GWP), or extra precisely over the whole equivalent Warming effect (TEWI) of CFC, HCFC and HFC refrigerants and their consequences on the surroundings, has led the refrigeration industries to study possible approaches of changing those offensive refrigerants in the refrigeration systems. al. 2010). The environmental effects of a few common refrigerants are proven in desk 1. Also, HCs are well matched with commonplace materials discovered in refrigerating systems and are soluble in traditional mineral oils.

1.1 utility of Sub-Cooling in Refrigeration gadget

Very not unusual utility of sub-cooling is its indirect use on the superheating manner. Superheating has similarities to sub-cooling in an operative way, and each methods can be coupled the usage of an inner warmness exchanger (figure 1). Sub-cooling,as shown in the discern, serves itself from the superheating and vice versa, permitting warmness to flow from the refrigerant at a higher stress (liquid) to the one with decrease pressure (gasoline). This creates an active equivalence between the sub-cooling and the superheating phenomena whilst there is no power loss.

Normally, the fluid this is being sub-cooled is warmer than the refrigerant that is being superheated, allowing an energy flux within the wished path. Superheating is critical for the operation of compressors due to the fact a gadget lacking it may provide the compressor with a liquid-gas aggregate, scenario that normally ends in the destruction of the fuel compressor because liquid is incompressible. This makes sub-cooling an easy and substantial supply of heat for the superheating system.

1.1.1 Sub-cooling for optimization and power saving

Sub-cooling procedure outdoor the condenser (as with an internal heat exchanger) is a great manner of using up all of the condensing tool's heat exchanging capacity. A massive portion of refrigeration systems use part of the condenser for sub-cooling which, even though very effective and easy, can be considered as a diminishing issue inside the nominal condensing potential. A similar state of affairs can be found with superheating taking area in the evaporator, as a result, an internal heat exchanger is a great and relatively reasonably easy solution for the maximization of heat exchanger capability (Klein et al., 2000).

1.1.2 Sub- cooling for enhancing refrigeration capability

Every other large application of sub-cooling is for enhancing refrigeration ability. Inversely to superheating, sub-cooling, or the quantity of heat withdrawn from the liquid refrigerant on the sub-cooling technique, manifests itself as an increase inside the refrigeration capacity of the device. This means that any extra heat removal

After the condensation (sub-cooling) allows a better ratio of heat absorption on in addition ranges of the cycle. it is to be stated that superheating has precisely the inverse effect, and that an internal heat exchanger alone, is not capable of boom the ability of the device due to the fact the boosting effect of sub-cooling is dimmed by using the superheating, making the internet ability advantage identical to zero (Bolaji and Huan, 2013).

1.1.3 Natural and synthetic sub-cooling

The sub-cooling procedure can happen in many unique approaches; therefore, it's far feasible to differentiate among the specific parts in which the system takes region. commonly, sub-cooling refers back to the significance of the temperature drop that's without problems measurable, however it's miles feasible to talk of sub-cooling in terms of the full heat being removed. The most commonly known sub-cooling is the condenser sub-cooling, which is usually known as the total temperature drop that takes place inside the condenser, immediately after the fluid has totally condensed,un till it leaves the condensing unit (Selbas et al., 2006).

Condenser sub-cooling differs from total sub-cooling generally due to the fact after the condenser, during the piping, the refrigerant may naturally have a tendency to cool even more, before it arrives to the enlargement valve, but also due to artificial sub-cooling. The entire sub-cooling is the whole temperature drop the refrigerant undergoes from its actual condensing temperature, to the concrete temperature it has while attaining the enlargement valve: this is the effective sub-cooling (Del-Colet al., 2010).

Natural sub-cooling is the call typically given to the temperature drop produced inside the condenser (condenser sub-cooling), mixed with the temperature drop happening thru the pipeline alone, with the exception of any heat exchangers of any kind. Whilst there may be no mechanical sub-cooling (i.e. an internal heat exchanger), natural sub-cooling must same total sub-cooling. However, mechanical sub-cooling is the temperature decreased by using any synthetic technique that is intentionally located to create sub-cooling. This concept refers mainly to devices such as internal heat exchangers, independent sub-cooling cascades, economizers or boosters (Bolaji., 2010).

1.1.4 Sub-cooling warmness exchangers

Sub-cooling heat exchangers are commonly installed in refrigeration systems with the intent of ensuring proper system operation and increasing system performance. Specially, ASHRAE (1998) states that sub-cooling

Heat exchangers are effective in: (i) increasing the system performance; (ii) sub-cooling liquid refrigerant to prevent flash gas formation at inlets to expansion devices; and (iii) absolutely evaporating any residual liquid that may remain within the compressor suction line. Therefore, sub-cooling heat exchanger is a device that can be used to evaluate the impact of refrigerants on refrigeration capacity and overall performance. This device has been utilized by some researchers to evaluate alternative refrigerants to R22 and R12. (Domanskiet al., 1994; Klein et al., 2000; Bolaji, 2010).

In refrigeration, sub-cooling is the technique via which a saturated liquid refrigerant is cooled under the saturation temperature, forcing it to alternate its section absolutely. The resulting fluid is called a sub-cooled liquid and is the convenient nation wherein refrigerants may also go through expansion and evaporation technique of a refrigeration cycle. Sub-cooling is usually used so that when the cycling refrigerant reaches the thermostatic expansion valve, its totality is in its liquid form, thus, allowing, the valve to work properly (Domanski et. al, 1994).

2.0 EXPERIMENTAL SETUP AND APPARATUS

2.1 Experimental setup

The test rig was a Prestige brand of a domestic refrigerator having detailed specifications shown in Table 5, originally designed to work with R134a was retrofitted to incorporate a sub-cooling heat exchanger and the three investigated refrigerants were introduced into it one after the other. The refrigerator was fitted with thermocouples and a manifold gauge. A sub-cooling heat exchanger was installed on this refrigerator at the condenser outlet while the capillary tube was increased base on ASHRAE Standard from 1000mm to 1800mm (ASHRAE, 1998). Measuring instrument were installed in their respective places to obtain the performance parameters. As shown in the schematic diagram of Figure 5, the following readings were obtained during the test:-

i. The mass of refrigerants introduced into the system ranged from 0.16 to 0.3 kg for each of the refrigerants R134a, R600a and R510A respectively.
ii. The suction pressure (P_1) of the compressor in kN/m^{2}at the evaporating temperature $T_1$was determined.
iii. The temperature $T_{44}$of the refrigerants in ^{o}C at the evaporator inlet, T_1 at the compressor inlet, T_2 at the outlet and T_3 and T_{33} at the sub-cooling heat exchanger inlet and outlet were measured respectively.
iv. The degrees of sub-cooling ranging from 2 to 10^{o}C were obtained between condenser and the sub-cooling heat exchanger by calculating the difference in their respective outlet temperatures.

The above readings were obtained and evaluated with the aid of software called REFPROP to calculate the enthalpy of the refrigerants h_1, h_2, and h_3 at the compressor inlet, compressor outlet and condenser outlet respectively after the sub-cooling heat exchanger. The pressure ratio, the compressor work input and the coefficient of performance were also calculated for the three refrigerants. The temperatures of the refrigerants at inlet and outlet of each component of the refrigerator were measured with the thermocouples. The thermocouples sensors were fitted at inlet and outlet of the compressor, condenser and evaporator. Temperature and pressure measurements are necessary in order to determine the enthalpy in and out of each component of the system to investigate the performance of the system working with different type of refrigerants. The manifold gauge was fitted at the inlet and outlet of the compressor and expansion valve . The manifold gauge is fitted with the T- joint and then brazed with the tube to measure the pressure at desired position as mentioned before .The ranges of the manifold gauges were -1 to + 39 bars. A charging manifold gauge and a vacuum pump (or spooter) was installed at the inlet of expansion valve and compressor for charging and recovering the refrigerants respectively. The evacuation has also been carried out through this service port. Amperage was connected with compressor to measure the power and digital compute charging scale to measure the energy consumption of the unit .

The refrigerator was tested using R134a as the baseline and performance data were obtained. The system was evacuated with the help of a Blue VAC vacuum pump to remove the non-condensable particle from the system. The system was charged with the help of manifold gauge. The pressure gauge and thermocouples were connected to the system. The system was instrumented with two pressure gauges with accuracy of $\pm$ 0.5kPa at the inlet and outlet of the compressor for measuring the suction and discharge pressures. The REFPROP software and equations (1) to (6) were used to compute various performance parameters at different sub-cooling temperatures.

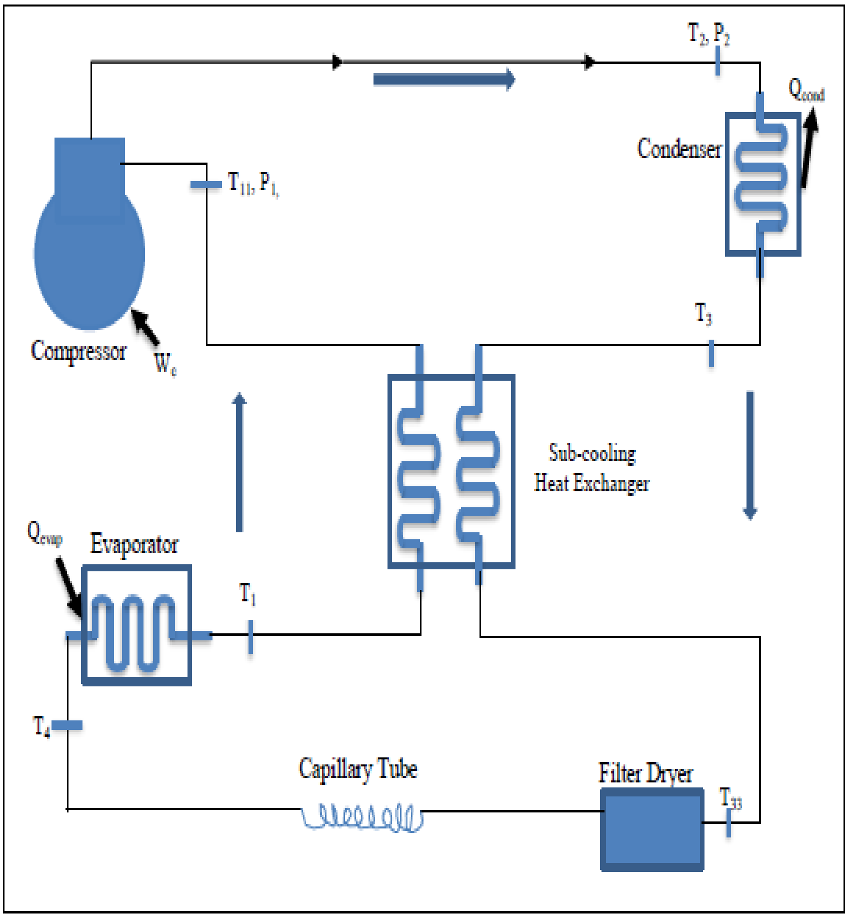

Figure 5: Schematic diagram of the test rig equipment and parameters obtained

Table 5: The specification of the Prestige refrigerator used

Specifications	Value
Freezer compartment capacity (liters)	130
Fresh food compartmentcapacity (liters)	320
Power rating (W)	80
Current rating (A)	0.60
Voltage (V)	220
Frequency (Hz)	50
No of door	1
Refrigerant type	R 134a
Defrost system	Auto Defrost

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2.2 Heat exchanger without sub-cooling

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

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 work input to compressor (W_c, kJ/kg) is expressed as:

$$W_c = (h_2 - h_1) \quad (2)$$

where, h_2 = specific enthalpy of refrigerant at the outlet of compressor (kJ/kg). The coefficient of performance (COP) is the refrigeration effect produced per unit of work required. It is expressed as:

$$COP = \frac{Q_{evap}}{W_c} \quad (3)$$

2.3 Heat exchanger with sub-cooling

Refrigeration cycle performance calculations without sub-cooling were carried-out with assumption that refrigerant exits the evaporator as a saturated vapour at the evaporator pressure (state 1) and exits the condenser as a saturated liquid at the condenser pressure (state 3). With reference to Figure 5, when a sub-cooling heat exchanger is introduced, the refrigerant entering the compressor (state 11) has been superheated by heat exchange with the liquid exiting the condenser which causes the liquid to enter the expansion device in a sub-cooled state (state 33). Therefore, refrigerating effect with sub-cooling heat exchange (Q_{evap}, kJ/kg) is expressed as:

$$Q_{evap} = (h_1 - h_{44}) \quad (4)$$

The work input to the compressor in the cycle with sub-cooling heat exchanger

(W_c', kJ/kg) is given as:

$$W_c' = (h_{22} - h_{11}) \quad (5)$$

The co efficient of performance of the vapour compression cycle with sub-cooling heat exchanger (COP') is the ratio of equation (4) to equation (5):

$$COP' = \frac{h_1 - h}{h_{22} -} \quad (6)$$

44

3.0 RESULTS AND DISCUSSION

Various performance parameters obtained during the experiment were tabulated and analyzed graphically and statically as follows:

3.1 Effects of Sub-cooling on the Performance of the Refrigerating system

Figure 1 shows graphical representation of the variation of refrigerating effect as a function of degree of sub-cooling for R600a, R510A and R134a. The refrigerating effects at sub-cooling temperature of 7ºC were 220.66, 286.86 and 108.70 kJ/kg respectively; while the values of 120.62, 145.20 and 95.17 kJ/kg were obtain without sub-cooling. Figure 2 shows the variation of compressor work input as a function of degree of sub-cooling for the investigated refrigerants. At sub-cooling temperature of 5ºC the compressor work input for R600a, R510A and R134a were 88.96, 85.90 and 49.71 kJ/kg respectively.

The effect of the degree of sub-cooling on pressure ratio for the three investigated refrigerants is shown in Figure 3. The pressure ratios of 1.4, 2.2 and 1.3 at sub cooling temperature of 8ºC were obtained for R600a, R510A and R134a respectively. The effect of degree of sub-cooling on the Coefficient of Performance (COP) is shown in Figure 4. The data obtained on the effect of sub-cooling of the system were analyzed statistically using Analysis of Variance (ANOVA) and the result are shown in table 1 to 4.

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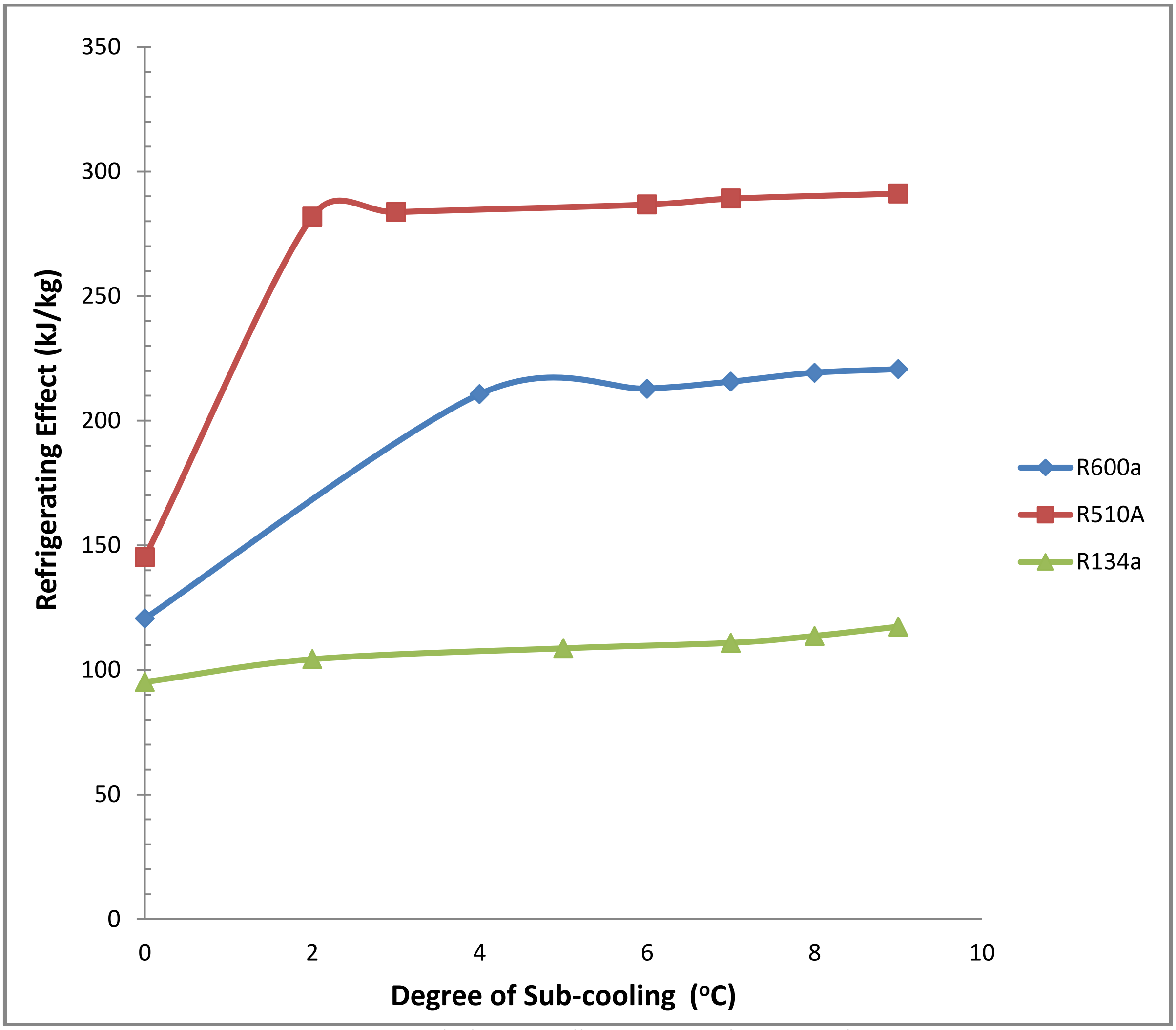

Figure 1. Variation of refrigerating effect with degree of sub-cooling from 0°C

Table 1: Post Hoc Tests (Multiple comparisons for refrigerating effect with degree of sub-cooling)

(I) Refrigerants	(J) Refrigerants	Mean Difference (I-J)	Std. Error	Sig.
R600a	R510A	-70.74000*	2.74022	.000
	R134a	104.83200*	2.74022	.000
R510A	R600a	70.74000*	2.74022	.000
	R134a	175.57200*	2.74022	.000
R134a	R600a	-104.83200*	2.74022	.000
	R510A	-175.57200*	2.74022	.000

*. The mean difference is significant at the 0.05 level.

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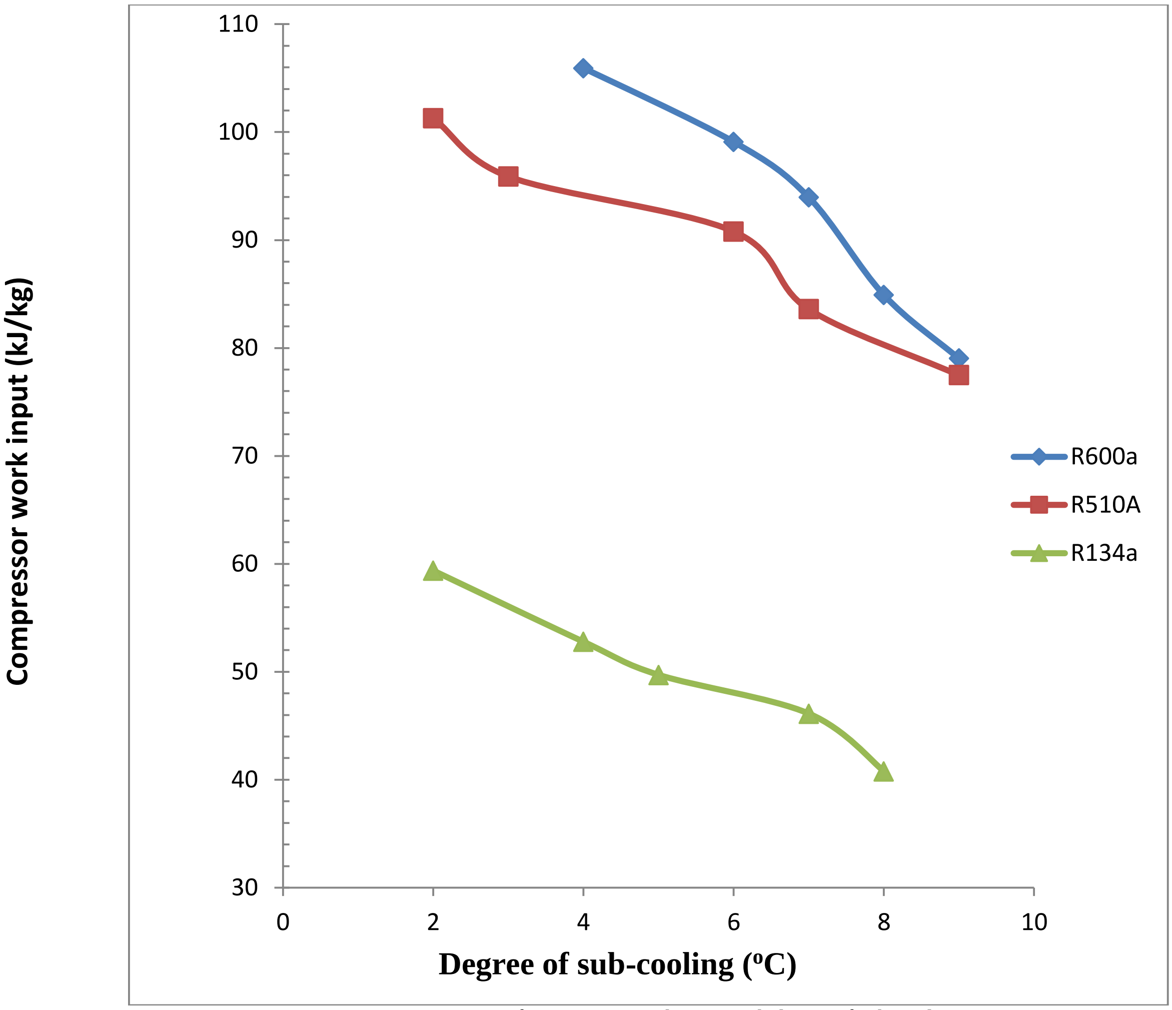

Figure 2: Variation of compressor work input with degree of sub-cooling

Table 2: Post Hoc Tests (Multiple comparisons for compressor work input with degree of sub-cooling)

(I) Refrigerants	(J) Refrigerants	Mean Difference (I-J)	Std. Error	Sig.
R600a	R510A	2.76800	5.83257	.644
	R134a	42.83200*	5.83257	.000
R510A	R600a	-2.76800	5.83257	.644
	R134a	40.06400*	5.83257	.000
R134a	R600a	-42.83200*	5.83257	.000
	R510A	-40.06400*	5.83257	.000

*. The mean difference is significant at the 0.05 level.

a. Parameters = compressor work input

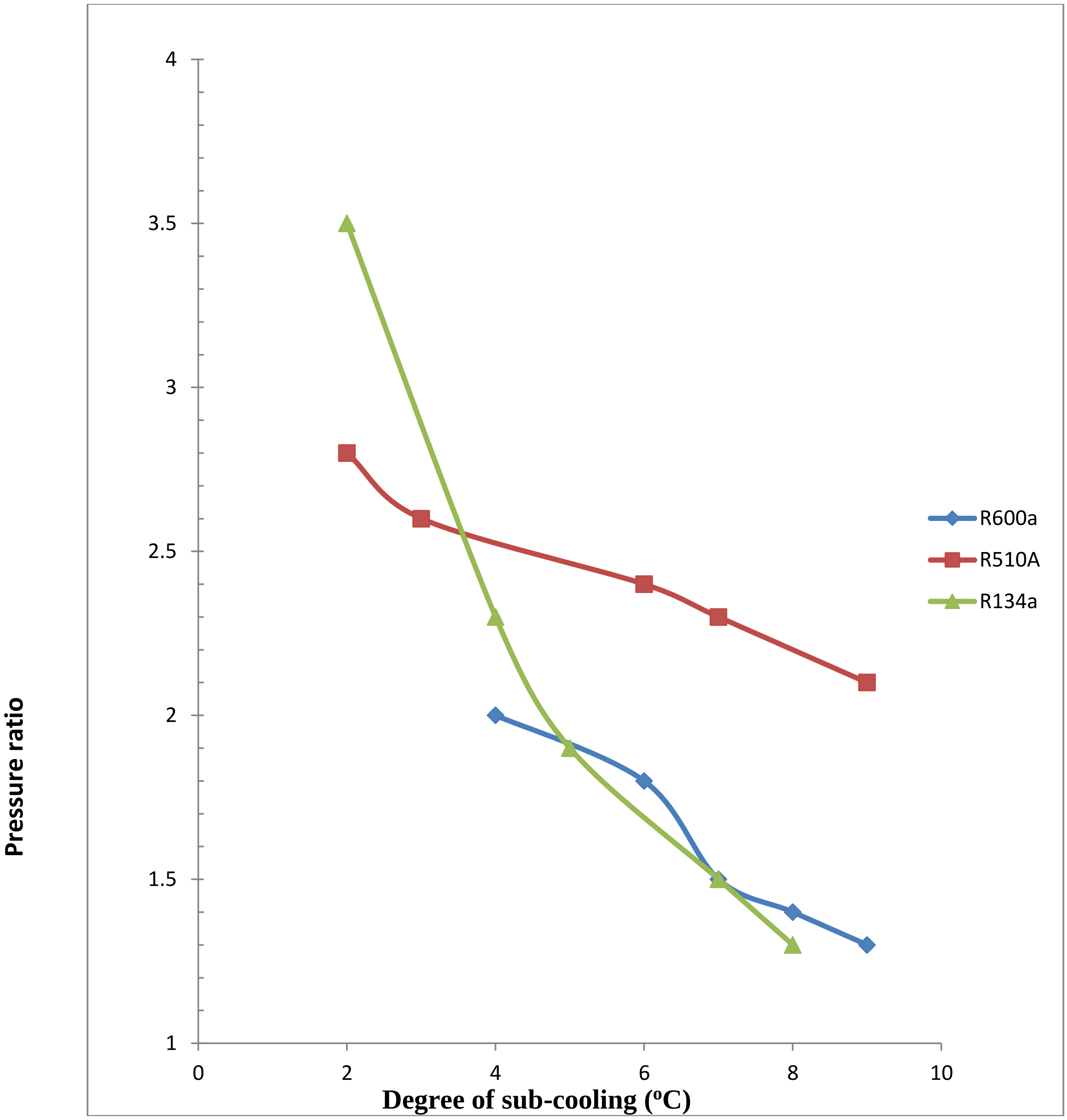

Figure 3: Variation of pressure ratio with degree of sub-cooling

Table 3: Post Hoc Tests (Multiple Comparisons for pressure ratio with degree of sub-cooling)

(I) Refrigerants	(J) Refrigerants	Mean Difference (I-J)	Std. Error	Sig.
R600a	R510A	-.88000*	.35214	.028
	R134a	-.52000	.35214	.166
R510A	R600a	.88000*	.35214	.028
	R134a	.36000	.35214	.327
R134a	R600a	.52000	.35214	.166

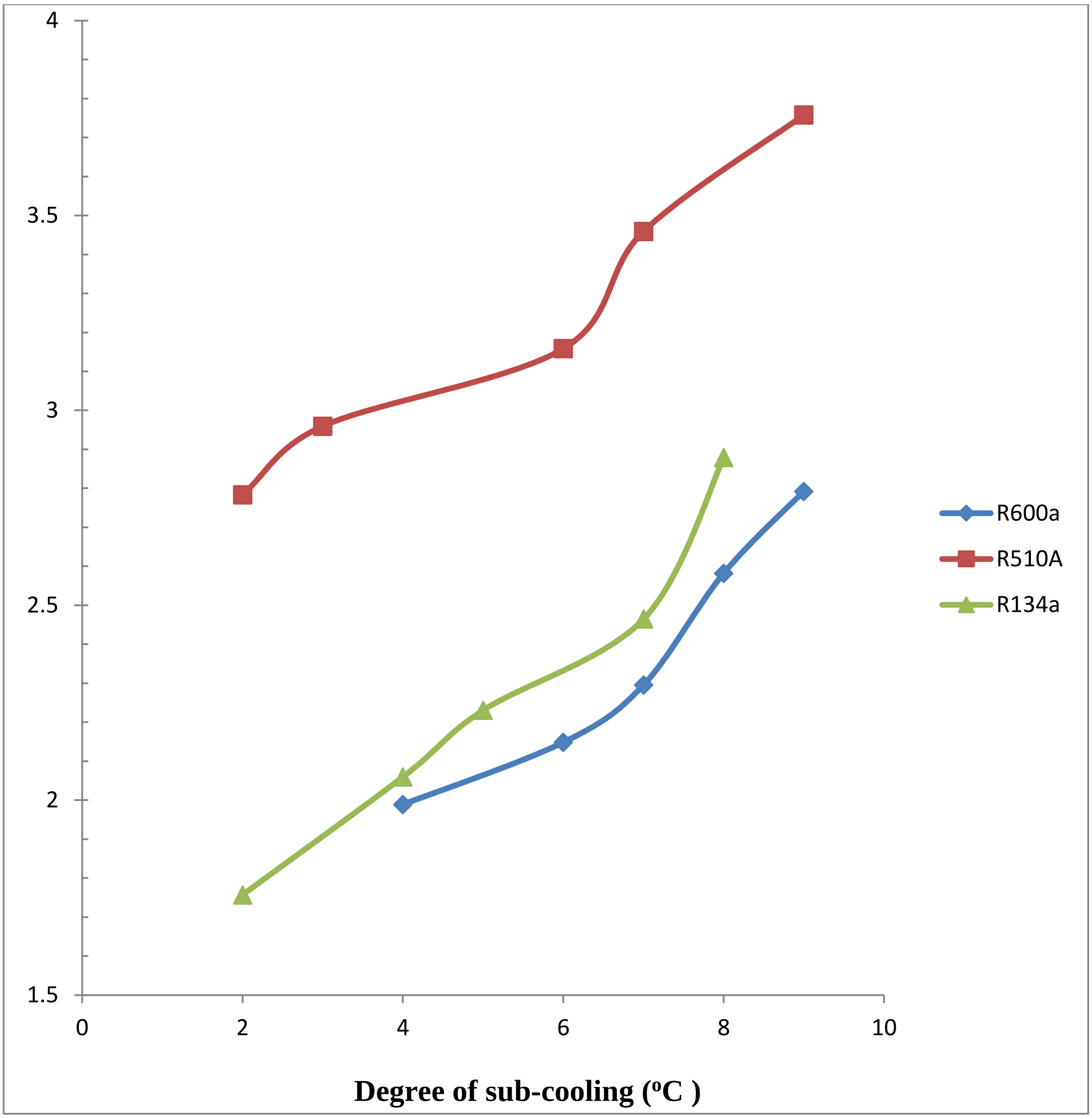

Figure 4: Variation of Coefficient of Performance (COP) with degree of sub-cooling

Table 4: Post Hoc Tests (Multiple comparisons for coefficient of performance with degree of sub-cooling)

(I) Refrigerants	(J) Refrigerants	Mean Difference (I-J)	Std. Error	Sig.
R600a	R510A	-.85800*	.24140	.004
	R134a	.08600	.24140	.728
R510A	R600a	.85800*	.24140	.004
	R134a	.94400*	.24140	.002
R134a	R600a	-.08600	.24140	.728
	R510A	-.94400*	.24140	.002

*. The mean difference is significant at the 0.05 level.

4.0 DISCUSSION

The results showed the effect of degree of sub-cooling on the refrigerating effect for R134a and the two hydrocarbon refrigerants. The refrigerating effect increases with increase in degree of sub-cooling as a result of increase in latent heat value of the refrigerants. A very high latent heat value is desirable. High values greatly increase the efficiency and capacity of the compressor. R600a and R510A exhibited higher refrigerating effect than R134a therefore; very low mass of refrigerant will be required for the same capacity. The highest average refrigerating effect was obtained using R510A with a value of 289.30 kJ/kg compared with 110.98 kJ/kg of R134a.

The compressor work input decreases with increase in degree of sub-cooling for R134a and the two hydrocarbon alternatives. The two hydrocarbon refrigerants exhibited higher compressor work input than R134a, but they equally exhibited very high refrigerating effect which is a form of compensation for their high compressor work input.

The result obtained showed that pressure ratio decreased with increased in degree of sub-cooling. This is an indication that compressor work input for the three refrigerants reduced accordingly due to sub-cooling effect. The average pressure ratios obtained using R510A and R600a were 17% higher and 25% lower, respectively than when R134a was used. Also at sub-cooling below 3.5ºC, R134a manifest higher pressure ratio than the two hydrocarbon refrigerants (R510A and R134a). Therefore, heavy work is required for employing R134a in the system at sub-cooling temperature below 3.5ºC; while the same compressor with little or no modification can be used for R600a in the system at sub-cooling temperature above 3.5ºC. The COP of refrigeration cycle shows the cycle performance and is the major criterion for selecting a new refrigerant as a substitute. At sub-cooling temperature of 4 to 8ºC the average COPs for R510A, R600a and R134a were 3.31, 2.23 and 2.41 respectively. Highest COP of 3.8 was obtained using R510A at sub-cooling temperature of 9ºC.

The homogeneous subsets obtained from ANOVA Post Hoc Tests show which groups (R600a, R510A and R134a) differ from each other based on the performance of pressure ratio. Also it reveals that there is **no** statistically significant difference in the performance of R510A and R600a when compared with R134a, since P-value is above 0.05. Hence, we can say that at 95% confidence level, there is no significant difference in the system performance of the pressure ratio between R134A when compared with both R600a and R510A. The ANOVA Post Hoc Tests based on the COPs of R600a, R510A and R134a revealed that there is statistically significant difference in the performance of R510A when compared with R600A, since P-value is below 0.05. In addition, there is no significant difference in the performance of R134a when compared with R600a, and there is significant difference in the COP of R510A and R134a.

5.0 CONCLUSION

In this research, the performance analysis of two hydrocarbon refrigerants (R600a and R510A) as alternatives to R134a in a domestic refrigerating system with sub-cooling heat exchanger were experimentally conducted and analyzed. The effects of sub-cooling on the performance of these refrigerants in a R134a refrigeration system were evaluated and compared with that of R134a. The following conclusions were drawn from the analysis and discussion of the results:

i. R600a and R510A exhibited higher refrigerating effect than R134a. The highest mean refrigerating effect of 289.30kJ/kg was obtained using R510A.
ii. The compressor work input decreases with increase in degree of sub-cooling for R134a and the two hydrocarbon alternatives (R600a and R510A).
iii. The average pressure ratios obtained using R510A and R600a were 17% higher and 25% higher respectively than when R134a was used.

The COP obtained using R510A was higher than those of R600a and R134a with average value of 1.08 higher than that of R134a

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