Experimental comparison of the Performance of a Domestic Vapour Compression Refrigeration system using hybrid (circular-triangular) finned- and single-finned Condensers

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Abstract: The performance of a domestic refrigeration system using hybrid and single geometry finned condensers is evaluated in this paper. The Test Rig comprised a 1/8 sized domestic refrigerator working on R600a refrigerant, developed hybrid condensers and instrumentation part (temperature data logger 3 pressure gauges, temperature recorders) Results showed that circular-triangular hybrid finned condenser is more efficient than the existing conventional type with average coefficient of performance of 0.683, 0.710, 0.727, 0.710 (for hybrid), and 0.683, 0.721, 0.710, 0.058 (single) over the load variations of ambient, 30 °C, 40 °C, and 50 °C. Hybridization of extended surfaces is another passive method of augmenting the heat transfer rate of heat exchangers

Index Terms: Refrigeration system, condenser, fins, augmentation technique, hybrid, energy, performance

Nomenclature

Q' : Heat flux (KJ/m²)
h'_g : Refrigerant heat transfer coefficient (W/m² °C)
T'_g : Refrigerant mean Condensing Temperature (°C)
T'_i : Condenser nodal internal wall temperature (°C)
$\Delta t'_g$: Nodal differential temperature (°C)
Cp_g : Refrigerant specific heat at constant pressure (kJ/kg °C)
T_g : Refrigerant temperature (°C)
A_f : Total finned area (m²)
P_{abs} : Absolute Pressure (kPa)
P_{cpd} : Discharge gauge Pressure (kPa)
P_{cps} : Suction gauge pressure (kPa)
V_p : Piston displacement (m³/s)
ρ_{rs} : Refrigerant density at suction pressure (kg/m³)
h_{sccp} or h_{sp}: Refrigerant enthalpy at suction (kJ/kg/°C)
h_{ccd} : Refrigerant enthalpy at condenser discharge pressure (kJ/kg/°C)
A_{fc}, A_{ft}: surface area of circular fin, triangular fin (m²)
h'_g : Refrigerant heat transfer coefficient (W/m² °C)
N_{f1} , N_{f1} : number of similar fins (24)
$l_{cd}{}'$: length of unfinned coiled part of condenser tube (m)
$l_d{}^i$: extending length of condenser from it exit to dryer inlet (m)
l_{fc} : horizontal finned length of coiled condenser tube (m)
l_{usc} : unfinned length of coiled condenser tube (m)
l_{ecn} : extending length of condenser from compressor exit (m)
$\frac{h_c}{n}$: Condenser tube pitch (m)
W_{cm} : Work of compression (W)
h_{cd}, : Refrigerant enthalpy at Compressor discharge pressure (kJ/kg/°C)
h_{cs} : Refrigerant enthalpy at Compressor suction pressure (kJ/kg/°C)
T_{av} : Average air Temperature (°C)
n : Number of condenser loop
η_f , η_{f1}, η_{f2}: efficiency of the extended surface fin (%)
L : Characteristic length of fin (m)
p_c, p_s, or p_t: Fin Perimeter (m)
d_i : Condenser tube inside diameter (m)
d : diameter of a fin (m)
s : side length of a square fin (m)
t : length of side of a triangular fin (m)
V_p : Compressor piston displacement (m^3

1 Introduction

Energy is one of the fundamental requirements for economic growth but its generation is at low pace. Refrigeration systems consume approximately 15 % of the available energy; therefore, prudent use of the available energy is vital. The economic importance of refrigeration systems, both for domestic household and industrial applications has been on the increase in recent decades and the day-by-day improvement of existing and innovation of new technology has made the living standard of the people to be on increase. International Institute of Refrigeration (IIR) estimated that refrigerators and air-conditioners use approximately 15 % of the world production of electricity (IIR, 2002). Hence, improvement to technical elements of modern refrigeration is imperative to make the design process of refrigeration systems more efficient and the product performance better for effective utilization of the available electrical energy.

Quite a number of studies have been done on the various components of the refrigerating systems so as to produce more-efficient and effective refrigeration systems. Some researchers worked on the auxiliaries (refrigerants, lubricants), and some on the basic components (Compressor, capillary tube, evaporator, and condensers.). Mao-Gang et al (2005) experimentally tested two pure hydrocarbons and seven mixture composed of propylene, propane, HFC 152a and dimethylether as an alternative to HCFC22 in residential air conditioners and heat pumps. Their experimental results showed that the coefficient of performance (COP) of these mixtures was up to 5.7% higher than that of HFC22. Mani and Cho et al. (2006) evaluated the performance of an air conditioning system, under steady-state conditions without the aid of any type of controller, by varying the refrigerant charge, the compressor speed, the expansion valve opening and the internal heat exchanger (IHX) length using Carbon dioxide (CO_2) as the working fluid. They found that any increase in the compressor speed increased the optimum valve opening but reduced the system coefficient of performance (COP), independent of refrigerant charge, and that the system optimum discharge pressure could be attained by the simultaneous adjustment of valve opening and compressor speed. Rani and Reddy (2014) evaluated the performance of a domestic refrigerator equipped with three layers of zig-zag shaped evaporator. They attempted to compare the performance of a domestic refrigerator equipped with zig-zag shaped evaporator with that of existing conventional (single) evaporator using R600a as the working fluid. Their results showed that the coefficient of performance of domestic refrigerator increased by 13 % and the heat rejection rate in the condenser increased by 1.3 % over the existing conventional evaporator. Elsayed and Hariri (2011) experimentally investigated the performance of a direct expansion air conditioning (A/C) unit having a variable speed condenser fan. Their results showed that maximum rated air flow of the fan is 0.43 m^3/s at 42 ºC outdoor air temperature and the minimum is 0.28 m^3/s, and found that a 10 % reduction in compressor power consumption was achieved by increasing the condenser air flow by about 50 % Al-Doori (2011) investigated the heat transfer by natural convection in a rectangular fin plate by circular perforations heat sinks. The results of the experimentations showed that the temperatures along the non-perforated fin dropped from 30 to 25 ºC and that of perforated from 30 to 23.7 ºC in low power (6 W) heat sink, whilst in high powered heat sink (220 W) the temperature was from 250 ºC 49 ºC but the temperature drop increases from the fin base to the tip as the diameter from also the temperature drop between the fin base and tip increases as the perforation diameter increased. The temperature drop with high power (220 W) was from 250 to 49 ºC for non-perforated fin

Various studies reviewed above focused mostly on the refrigeration system using conventional single finned condensers, with little information on the performance of refrigeration systems equipped with hybrid finned condenser. Therefore, in this paper, performance of a domestic refrigeration system equipped with circular-triangular finned condenser would be compared with that of a similar domestic refrigerator equipped with existing conventional single finned condenser using R600a refrigerant

2 Materials and Methods

2.1 Materials and Equipment

The materials and equipment used for this study include: 48 pieces of fins of two geometries

(Circular and triangular), 3 pieces of 1/8 sized conventional single finned condenser, bundle of twain rope, a packet of paper-tape, unit mass of water sample, 1/8 sized upright freezer, Temperature data-logger, 8 pieces of threaded Union, Manifold gauge, micro-computer temperature controller, Thermo-Hygrometer, Multi-thermometer, one low and two high pressure Gauges, 1.0 kW MasterCheff Hot-plate, 2.0 kVA Voltage Regulator, 6 pieces of Cfoss and Brass rods. The two sets of fin geometries were produced by extrusion process from Aluminium material (with elemental compositions of: 99.10 % Al; 0.2842 % Si; 0.2201 %Fe; 0.1961 %Mg; 0.0927 %V; M0.0302 %Cr; 0.0202 %Zn; 0.0125 %Ti; 0.0109 %Mn; 0.0101 %Ni; 0.004 %Cu; 0.0021 %Sn; 0.0015 %Ca; 0.0011 %Pb; 0.0009 %Na; and 0.0003 %Cd) at Foundry Workshop of the Federal Institute of Industrial Research, Oshodi (FIIRO), Lagos. Table 1 shows the fins specifications. The conventional condensers were procured from the Refrigeration and Air-conditioning Section of King Adesida Market, Akure, Nigeria.

Table 2: Fins Specifications

s/n	Fin Geometry	Dimension
1	Circular	3.0 mmØ x 418 mm length
4	Triangular	4.04 mm x 4.04 mm x 4.04 mm 418 mm

2.2 Method

2.2.1 Development of circular-triangular (CT) hybrid finned Condenser

A bare coiled condenser tube was obtained by definning the fins lined on the procured 1/8-sized single-finned coiled condenser with the aid of a Plier and File. 24 pieces of similar fin geometry (circular) were picked, lined at equal fin pitch and fastened on half of the finned length (X_1) of the bare coiled condenser tube. This procedure was repeated with triangular fin for the remaining half finned length of the bare tube (X_2). Figure 1 shows a schematic diagram of the developed finned condenser

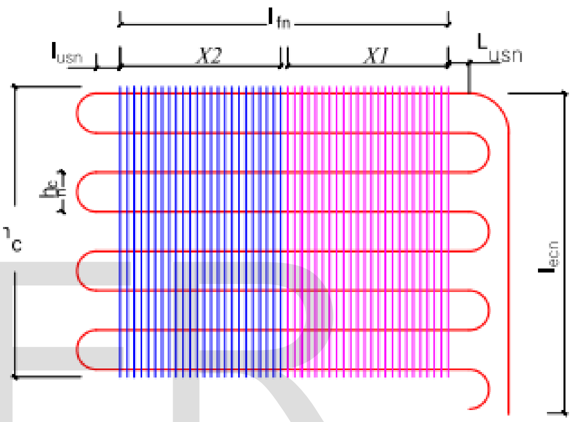

Figure 1: Schematic diagram showing parts of developed hybrid condenser

2.2.2 Experimental Theory

The heat flow per unit area along the local point of the condenser can be expressed equation (1), Kakac, 2001

$$Q^{"} = h_g^o\left(T_g^* - T_i\right) + \sum_{i=1}^{n} n_i \ddot{\Delta} t'_{lg,i} \qquad (1)$$

where $\ddot{\Delta} t'_{lg} = \ddot{\Delta} t_{lg,i} + \ddot{C}_{pgi}\left(T_g - T_i\right))$

This heat flux relied on three factors: (1) sensible cooling of the bulk vapour mixture as it moves through the condenser, (2) sensible cooling of the bulk vapour mixture as it flows from the local bulk conditions to the interface at a temperature T_i, and (3) latent heat of condensation of the vapour condensing species

2.2.3 Determination of the Fin Pitch

The fin pitch Consider (p_f) was considered using a finned tube condenser of finned length X_n (in metre) carrying N_f number of fins of base thickness ∅ (in metre) corresponding to the length of side of the extended surface. Hence, the fin pitch is expressed using equation (2):

$$p_f = \frac{X_n - \emptyset}{N_f - 1} \qquad (2)$$

The total length of the finned condenser consists of four parts, namely: (i) finned part, (ii) unfinned curved or looped part, (iii) unfinned straight part connecting the horizontally finned with the looped part, and (iv) unfinned part extending from the looped part to the discharge line. Consider a condenser tube of outer and inner diameters d_o and d_i bent with tube pitch p_t into n_l number of loops so its total vertical height is h_c as shown in figure (1): Therefore, the total length ($\boldsymbol{l_{tb}}$) of the bare condenser tube was obtained using equation (3)

$$\boldsymbol{l_{tb}} = \frac{\pi\, n_l}{2}\left|\frac{h_c}{n_l} - d_o\right| + h_c + N_f\left(t_{f1} + t_{f2}\right) + l_{ecn} \qquad (3)$$

2.2.4 Areas of the Condenser

The area of the condenser comprises the: (i) bare area (A_{po}) and (ii) finned area (A_f)

(i) The bare area is given by equation (4):

$$A_{po} = \pi d_o l_{btc}' \quad (4)$$

where l_{btc} is the total length of the condenser tube

(ii) The finned area using equation (5) adapted from Kuppan, (2003)

2.2.5 Volumetric efficiency and Quantities of refrigerant charged

As a result of the effect of the clearance space, the volumetric efficiency (η_{vol}) can be expressed in terms of the specific discharge and suction volumes using equation (7) proposed by Ballaney (1997 and Ibrahim et al., (2010)

$$\eta_{vol} = `1 + \varepsilon\left(1 - \frac{v_s}{v_d}\right) \quad (7)$$

where ε is the clearance factor with values vary mostly from 6 to 12 % according to Ballaney, (1997) and Arora (2009) Hence, in this study an average value of 9 % was considered. v_d and v_s are the specific volumes at compressor discharge and suction pressures.

The refrigerant mass flow rate (m_r) can be expressed using equation (8), (Dirk, 2010):

$$m_r = 10^{-6}\left[1 + \varepsilon\left(1 - \frac{v_s}{v_d}\right)\right]\frac{V_p}{v_s} \quad (8)$$

2.2.6 Determination of Fin Efficiency (η_f)

According to Arora (2009), fin efficiency (η_f) is ratio of the difference between the fin tip temperature (T_p) and the ambient fluid temperature (T_a) to the difference between the fin base (e.g. Condenser outer wall, the primary surface) temperature (T_b) and the ambient fluid temperature (T_a); Mathematically, this relation can be expressed, according to Arora, (2009) using equation (9)

$$\eta_f = \frac{T_p - T_a}{T_b - T_a} \quad (9)$$

The overall surface efficiency (η_T), according to Kuppan (2010), is given by equation (10)

$$A_f = N_{f1}A_{f1} + N_{f2}A_{f2} \quad (5)$$

so that the total heat transfer area, according to Incropera and DeWitt (2012), can be expressed with equation (6):

$$A_o = A_{po} + N_{f1}A_{f1} + N_{f2}A_{f2} \quad (6)$$

$$\eta_T = 1 - \frac{N_{f1}A_{sf1}}{A_{sT}}(1 - \eta_{f1}) - \frac{N_{f2}A_{sf2}}{A_{sT}}(1 - \eta_{f2}) \quad (10)$$

However, for single finned condenser, the overall surface efficiency, Incropera and DeWitt (2007); Kakac and Pramuanjarownkij (2012) is as in equation (11):

$$\frac{N_f A_{sf}}{A_{sT}}(1 - \eta_f) \quad (11)$$

2.2.7 Compression Ratio

The compression ratio (C_R) is given by equation (12) (Dirk, 2010)

$$C_R = \frac{P_{abs} + P_{cpd}}{P_{abs} + P_{cps}} \quad (12)$$

Also, according to Ibrahim and Mehtec (2003) the refrigeration capacity (R_{ev}) can be expressed as in equation (13):

$$R_{ev} = V_p \eta_{vol} \rho_{rs}(h_{sccp} - h_{ecd}) \quad (13)$$

2.2.8 Work of Compression

The work of compression, defined by and from the Temperature-entropy relation of figure 2, and from the formula adopted by (Ballaney, 1997), can be expressed with equation (14)

$$W_i = m_r(h_{cd} - h_{cs}) \quad (14)$$

2.2.8 Coefficient of Performance (COP)

The COP is the ratio of the refrigerating effect to the isentropic work (Arora, 2009) The coefficient of performance of the refrigerating system, according to Ibrahim and Mehtec (2003) andArora (2009), can be expressed using equation (15) adopted from Khurmi and Gupta (2007)

$$COP = \frac{R_{ev}}{W_{cm}} = \frac{V_p \eta_{vol} \rho_{rs}(h_{ecd} - h_{sccp})}{m_r(h_{cd} - h_{cs})} \quad (15)$$

2.3 Description of Experimental Facility and Procedure

The facility used a 100 L sized upright shelf type freezer. The freezer comprised the basic components of vapour compression refrigeration system: a 80 W self-equalization Danfoss hermetic compressor of 5.06 cm³ piston displacement, a finned condenser (the test section), a drier, a capillary tube and shelf tube type evaporator, that works on R600a refrigerant.

The probes of the temperature data-logger were fastened with paper tape to the condenser tube-fin interface for periodically logging of the fin base-tube interface temperatures at the 16 designated nodes of the test section. 3 pressure, (one low (P1) and High (P2 and P3), pressure gauges as depicted in figure 3 were installed at the entry and exits of the compressor and condenser installed to read and record the compressor and condenser suction and discharge gauge pressures

Room temperature and relative humidity were measured with a precision thermo-hygro-meter; the masses and temperatures of water sample (refrigerating item) were measured with weighing balance and a Multi-thermometer during each successive heating of the water sample. Charging of the Test Rig Refrigerant, R600a, was done with the aid of Manifold gauge. Plate 1 shows the pictorial view of the experimental facility

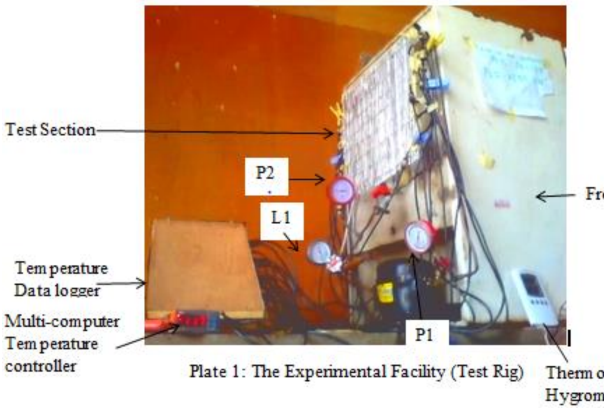

Plate 1: Pictorial view of the Experimental Facility

A known mass of clean Aluminium (0.1345 kg) cooking pot was filled with water and weighed so that the exact mass of the water is 2.0 kg. The initial temperature of the water sample, refrigerator box temperature, and room temperature with the relative humidity were also read and recorded. The refrigerator was loaded with the measured sample and switched on to refrigerate the sample for period of 8 hours. During the refrigeration process data on the: box temperature, room temperature/relative humidity; fin base- tube interface temperatures compressor and condenser inlet and discharge pressures were taken at interval of 1.0 hour to determine the evaporator temperature, condensing temperature, ambient temperature, box temperature, and pressure ratio needed for the evaluation. The procedure was repeated again when the water sample were raised to 30 ^oC, 40 ^oC and 50 ^oC, both for single finned and the developed hybrid finned condensers.

3 Results and Discussions

The performance of the refrigerating system when equipped with conventional single circular and circular-triangular (CT) were analysed in succession based on the data collected to compare their thermal behaviours. The thermal properties used for this evaluation include: Coefficient of Performance (COP), Refrigerating effect, Pressure ratio, temperature lift and condenser split. The results are graphically depicted in figures 2 to 16.

3.1 Pressure Ratio with Evaporator Temperature

Pressure ratio is the ratio of the absolute discharge pressure to the absolute suction pressure. Pressure ratio can be increased either by: increasing the discharge pressure, reducing the suction pressure or combination of the two pressures. A low suction pressure produces and signifies low-pressure vapour which has fewer new atoms or molecules and more re-expand residual vapour per cubic volume to carry heat away from the box and high discharge pressure since the compressor needs to work harder to attain the required condensing temperature; hence, more un-useful energy (heat of compression) is generated. A low suction pressure signifies that less heat is absorbed by the evaporator. Therefore, a low suction pressure is less effective at transferring heat away from the system. A high-pressure ratio, due to low suction pressure and high discharge pressure, would cause high discharge temperature (condensing temperature), low evaporator temperature, cylinder walls to become hotter which increases the temperature of the incoming vapour, more power consumption requirement by the compressor, and low volumetric efficiency. A high-pressure ratio denotes decrease in the COP because (Ibrahim and Dincer, 2010; Sharma, 2015). A low-pressure ratio, arising from high suction pressure and low discharge (appreciable) pressure, means increase in refrigerating capacity per unit mass of refrigerant, less power consumption by the compressor to operate the refrigerating system, increase in the COP due to reduction in the condensing temperature and increase in the evaporator temperature. A low compression ratio causes a system to run with ease (Dincer, 2003). Figures 2-5 present the variations of the pressure ratio with evaporator temperature at the various thermal loads considered. It can be observed that the pressure ratios increased with time across the two kinds of the heat exchangers. It was also observed that the conventional single finned condenser possessed higher compression ratios that the developed hybrid finned condenser. The lower pressure ratio depicted by the refrigerator when equipped with hybrid finned condenser, accounted for why it would be more reliable than when it is working with existing single finned condenser. According to Dincer (2003) and Sharma (2015) and a high-

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pressure ratio would lead to system burnout or poor system reliability.

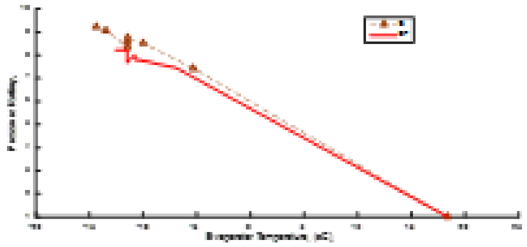

Figure 2:Pressure ratio aginst Evaporator at ambient Thermal Loads

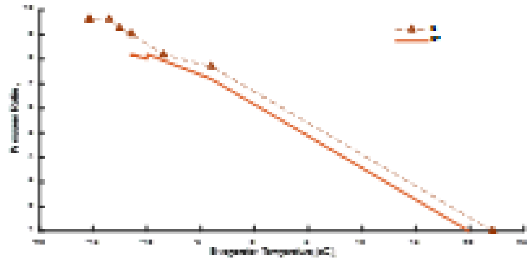

Figure3: Pressure Ratio against Evaporator Temperature Temperature at 30 ^{o}C Thermal Loads

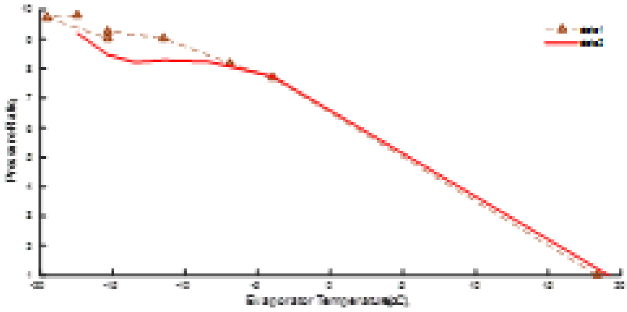

Figure 2:Pressure ratio aginst Evaporator at 40 ^{o}C Thermal Loads

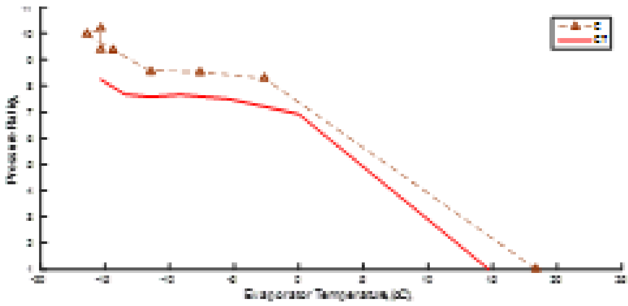

Figure3: Pressure Ratio against Evaporator Temperature Temperature at 50 ^{o}C Thermal Loads

3.2 Temperature Lift (TL) and Condenser Split (CS) with Refrigerant Mass Flow Rate

Temperature lift refers to the difference between condensing temperature and evaporative temperature; and the condenser split refers to the difference between the condensing temperature and the ambient temperature The Temperature lift within a refrigerating system, which is a measure of the reliability index of the compressor (Pearson, 2014), depends on the evaporating temperature and condensing temperature; the condenser split, which is a measure of the performance of the heat exchangers, depends mainly on the condensing temperature, but not affected by the ambient changes (Tomcyzk, 2017). While the evaporating temperature is related to the refrigerating product temperature, the condensing temperature is related to the refrigerating system ambient temperature. A system works best and becomes reliable when it operates with minimum temperature lift and condenser split. To achieve this, the evaporating temperature must be high and the condensing temperature low as possible whilst having the ambient temperature fixed (Dincer, 2003; Food and Drink Federation, 2007). A higher condenser split means heat is absorbed by the working fluid from the refrigerated space due to excess heat load in the evaporator and unchanged prevailing ambient temperature which causes some of the heat (due to work of compression, un-useful superheating to remain in the working fluid) to be running over certain time to reject the heat, hence, an increase in the work of compression and reduction in the COP (Tomczyk, 2012). Also, a higher temperature lift means higher compressor work and higher energy usage, less COP and vis-a-versa

Figures 6 to 9 present plots of the variations of the temperature lift and condenser split with the refrigerant mass flow rate. From the trends, circular-triangular (CT) finned condenser exhibited lower values of condenser split and temperature lift throughout the range of the thermal loads. least increase in temperature lift with time.

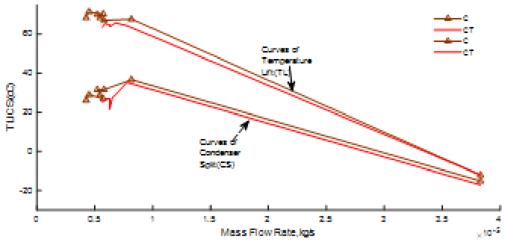

Figure6:Graph of Temperature lift and Condenser split against masss flow rate at ambient thermal loads

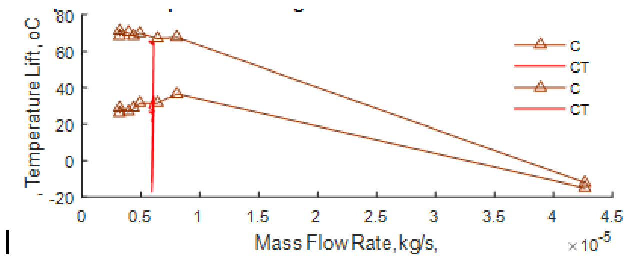

Figure7:Graph of Temperature lift and Condenser split against masss flow rate 30 °C theraml loads

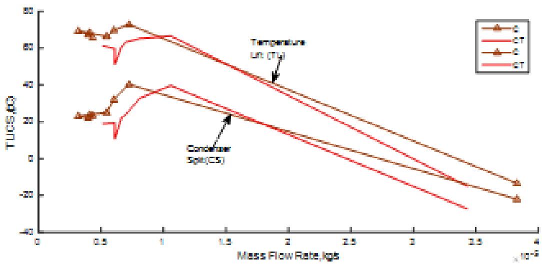

Figure 8:Graph of Temperature lift and Condenser split against masss flow rate at 40 °C thermal lads

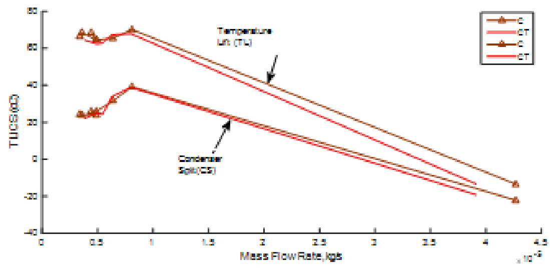

Figure 9:Graph of Temperature lift and Condenser split against masss flow rate 50 °C thermal loads

3.3 Refrigerating Effect and Work of Compression against Evaporator Temperature

Refrigerating effect, otherwise called the cooling effect, of refrigeration system stands for the ability of the refrigerant to remove heat (Dirk, 2010) and work of compression refers to the while the refrigerating effect of a system increases with proportionate increase in evaporator temperature (when there is smaller difference between the pressures), the work of compression decreases with it and vis-a-versa. Figures 10 to 14 present the effect of evaporator temperature on the cooling effect produced and work of compression as the compressor pumps the refrigerant round the system. From the plots, it was observed that the refrigerating effect increased across the two configurations but more significant with hybrid finned setup than in conventional setup. The improvement in the refrigerating effect due to variation in the pressure ratio arose as a result of higher evaporator pressure and discharge pressure (proportionate) which caused newer refrigerant and fewer re-expanded vapour to be circulated with each revolution of the crankshaft.

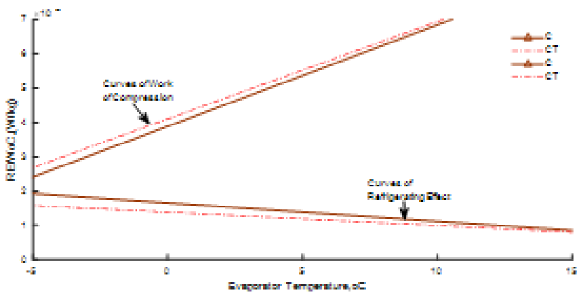

Figure10:Graph of Refrigerating Effect(RE) and Work of Compressoin(WoC) against
against
Evaporator Temperature at ambient thermal loads
loads

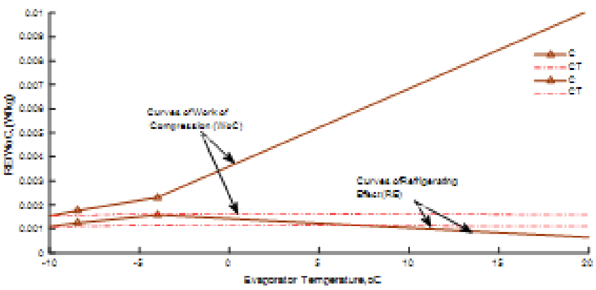

Figure11: Graph of Refrigerating Effect(RE) and Work of Compressoin(WoC)

Evaporator Temperature 30 °C heat

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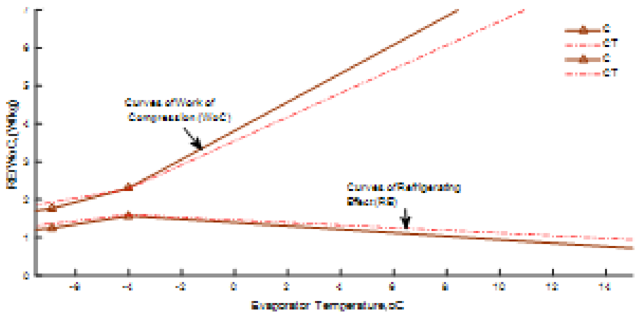

Figure13::Refrigerating Effect(RE) and Work of Compression (WoC) against Evaporator Temperature at 40 °C Thermals

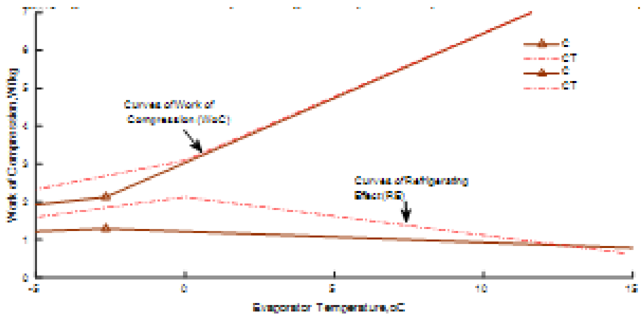

Figure 14 Refrigerating Effect(RE) Work of Compressoin(WoC) against Evaporatore 50 °C Thermal Loads

3.4 Coefficient of Performance (COP) against Temperature Lift (TL)

The Coefficient of performance (COP), according to Armamann (2012) refers to the ratio of the refrigerating capacity to the input power; it is a parameter which has to be optimized in a given refrigeration application for maximum conservation of energy (Dabas et al., 2011). As a result of the similarity of the plot, the variations of the COP with temperature lift at 30 °C and 50 °C were considered. Figures 15 to 16 show the variations of the coefficient of performance (COP) of the circular-triangular hybrid finned condenser and that of the convectional single circular finned condenser at different heat loads with temperature lift. It was observed that the COP of the conventional single finned condenser declines as the heat load increases but the performance increases with hybrid finned condenser. This implies that cooling capacity per work of compression is greater for the hybrid finned than the conventional single finned type refrigerator. This agreed with the findings of: Dossat (1991) that the COP of a refrigerating system is an index for indicating improvement in cycle efficiency, and Ivan et al., (2012) that COP is the ratio of generated cooling capacity (heat extracted by the refrigerator) to the electricity supplied to the compressor. Thus, more heat is extracted from the refrigerating space by refrigerator when working with hybrid finned condenser than conventional single finned condenser.

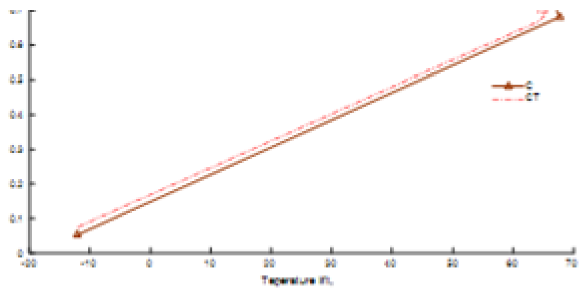

Figure 15: Graph of COP against Time at with water at 30 °C Temperature

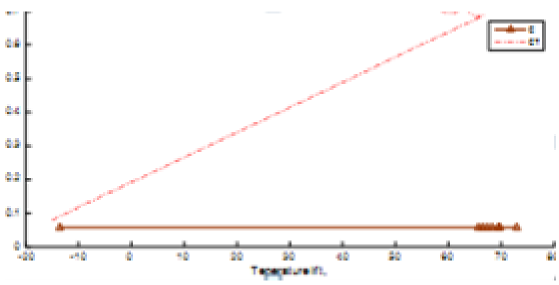

Figure 16: Graph of COP against Time at with water at 50 °C Temperature

4. Conclusion

This study offers some insight into the role hybridization of extended surfaces (fins) would play when used to augment the heat transfer rate of air-cooled condenser used in a domestic vapour compression refrigeration system, or HVAC equipment. It was found that the system is more efficient when it works with hybrid- (circular-triangular) finned condenser than with the existing single- (circular) fined condenser. It was, also, observed that improvement in reliability or elongation of shelf life of system could be achieved with this new innovation as it is characterized with low temperature lift and condenser split. On the average; it was found that the COPs of the refrigerator varied in this order 0.683, 0.710, 0.727, 0.710 with hybrid circular-triangular finned condenser, and 0.683, 0.721, 0.710, 0.058 with single circular finned condenser over the load variations.

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