ISSN 2229-5518

# Statistical Relevance of Variations in Condenser extended surfaces geometries and Heat loads on Properties for estimating Performance of Components of Refrigeration System

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**Abstract:** The performance of any system is a joint function of the performance of its individual components. This paper presents statistical interpretations of the effects of variation in condenser enhancing surface geometries and inherent thermal loads of refrigerating items on measured thermodynamic properties used in estimating the performance of components of household refrigeration system. Statistical analysis of the experimental data obtained was done with Analysis of Variance (ANOVA) technique using Minitab software.. The responses of the measured thermal properties to the configurations of the condenser heat enhancing surface geometries when enhanced with single-finned and hybrid finned over four range of thermal loading conditions (prevailing ambient, 30 $^{o}$C, 40 $^{o}$C and 50 $^{o}$C temperature) for period of 8 hours were also successively studied. Six sets of hybrid-finned and three sets of single-finned air-cooled condensers produced from a mix of three fin geometries (circular, triangular and square). It is found that configurations of extended surfaces of condenser and inherent heat loads of refrigerating item have different influence on the performance of the various components of refrigerating systems



**Index Terms**: measured thermodynamic properties, condensers' enhancing surfaces geometries, refrigeration system's components, performance,

## 1. Introduction2

Refrigeration system, these days, has become one of the household necessities mostly based on some functions it is being used to perform, apart from the fact that it is used to preserve perishable items pending when the time when needed, it also used to chill drinks and some fruits to required temperatures, either at home, inns (domestic, offices). The ability of this system to be able to meet these variant number of requirements is a measure of its performance The performance of refrigeration cycle can be improved by lowering the compressor power consumption, increasing the condenser heat rejection capacity or reducing the difference between condenser and evaporator pressures (Cengel, 2007); Therefore, The performance of refrigeration system depends on the performance of all the components of the system in other words, it is joint function of the performance of its components.

Various studies have been done towards the improvement of refrigeration system, by improving the performance of its components. In 2016, a numerical investigation of the performance of segmentally baffle shell-and-tube evaporator with and without liquid overfeeding was conducted. In the study, refrigerant R134a flows inside the tube whilst water flows on the shell side. it was observed that at a constant heat duty and fixed exit water conditions, the inlet evaporator temperature increases using liquid recirculation when compared to the case of dry expansion (Bortolin *et al*., 2016). In 2004, an experimental study of the effects of variations in refrigerant charge and ambient temperatures on refrigeration system performance was studied with a household refrigerator and food freezer that utilized 340 W rated hermetic compressor and worked with R404A refrigerant as the working fluid. The unsteady analysis was carried out with and without a thermostat switch; the results showed that a great superheating effect of the refrigerant occurred at reduced/low refrigerant charge but as the charge increases, greater than 260 g, no improvement occurred with respect to the evaporator temperatures (Tochio and Anglesio, 2004). Sencan (2006) presented computer-based first law and exergy analyzes of the influence of subcooling and superheating of refrigerants (R134a, R477a and RR410a) on vapour compression refrigeration system performance. They observed that subcooling and superheating temperatures (which are measures of Coefficient of performance, COP) directly influence the system performance and depend on condenser and evaporator temperatures. It was also observed that R134a and R407a displayed similar trends in subcooling and superheating whilst R410a deviated; they found that R134a has the highest efficiency rate with R410a the least. In 2016, Hisamudin *et al*., experimentally evaluated effect of variation in ambient temperatures (30, 35 and 40 $^{o}$C), internal heat loads (0, 500, 700 and 1000 W) and engine speeds (1000, 1500 and 2000 rpm) on the performance of automotive air-conditioning system; they found that the performance of the automotive air-conditioning system decreases when the ambient temperature, internal heat load and compressor speed increase. Also, Vali *et al.,* in 2011 experimentally investigated effect of varying the condenser fins spacing on vapour compression refrigeration system performance with sole objective of increasing the heat dissipating rate in the condenser. From the study; they found that heat dissipating of the condenser depends on the fins spacing (pitch) and that the performance of the refrigeration system decreases as the fins spacing increases but maximum at 2 mm

From the literature survey, it was found that series of studies have been conducted on how, means of and why their needs to improve the performance of refrigeration system or its components but researches on statistical relevance of properties used for analyzing these performance characteristics are limited and this justified reason for delving into this study ,

**2. Materials and Method**

Six sets of hybrid-finned and three sets of single-finned air-cooled condensers developed from a mix of three fin geometrical shaped fins (circular, triangular and square) were used for the experimentation. Condenser tubes (of 6.35 mm diameter) were obtained from pieces of air-cooled condensers, whose attached array of 1.5 mm diameter circular fins had been definned with a plier and file, were procured from king Adesida market, Ondo State. The three sets of fin geometries (square, triangular and circular), used for developing the single-finned and hybrid-finned air-cooled condensers were produced by extrusion process at The Foundry Workshop of Federal Institute of Industrial Research, Oshodi (FIIRO. Lagos State Table 1 shows the elemental composition of the fin materials. Each of the geometries selected were made to specifications as detailed in Table 2

Table 1: Elemental Composition of Fins

| Element | Wt (%) | Element | Wt (%) | Element | Wt (%) |
|---|---|---|---|---|---|
| Al | 99.10 | Cr | 0.0302 | Cu | 0.004 |
| Si | 0.2842 | Zn | 0.0202 | Sn | 0.0021 |
| Fe | 0.2201 | Ti | 0.0125 | Ca | 0.0015 |
| Mg | 0.1961 | Mn | 0.0109 | Pb | 0.0011 |
| V | 0.0927 | Ni | 0.0101 | Na | 0.0009 |
| Cd | 0.0003 | | | | |

**Source:** Extracts from Specimen Laboratory Results, (Foundry Workshop, FIIRO)

Table 2: Specifications of developed Fins

| S/N | Geometry | Dimension (mm) |
|---|---|---|
| 1 | Circular | Ø 3.0 x 418 |
| 2 | Square | 2.67 x 418 |
| 3 | Triangular | 4.04 x 418 |

**Source:** Extract from the Fin Geometries Design Calculation

The developed finned air-cooled condensers were mounted and tested, in succession, on upright shelf-type freezer that utilizes 80 W rated hermetic reciprocating compressor (Model: TLES5KK2) manufactured by Danfoss to refrigerate 2.0 kg of water sample for period of 8 hours. The sample was maintained at four (4) range of thermal loads (prevailing ambient, 30 °C, 40 °C and 50 °C temperatures). The term 'thermal load' is used to refer to the heat content of the refrigerating (water) sample, expressed in terms of temperature, the sample is raised to by heating, prior to the commencement of the experimentation. Each of these thermal conditions was achieved by boiling the sample on 1000 W electric hot-plate (Model: MasterChelf, MC-2001HP). Data were recorded with the attached instrumentations as shown in Figure 1. Condenser exit, compressor suction and discharge pressures taken with Pressure Gauges (Model: GT-536G), Box temperature and sample temperature with Micro-computer temperature controller and Multi-thermometer (Models: Elitech, RT-200+, and H-9283) and ambient temperatures with digital Thermo-Hygrometer. Data obtained on pressures were used to compute the associated temperatures with the aid of Tables of properties of

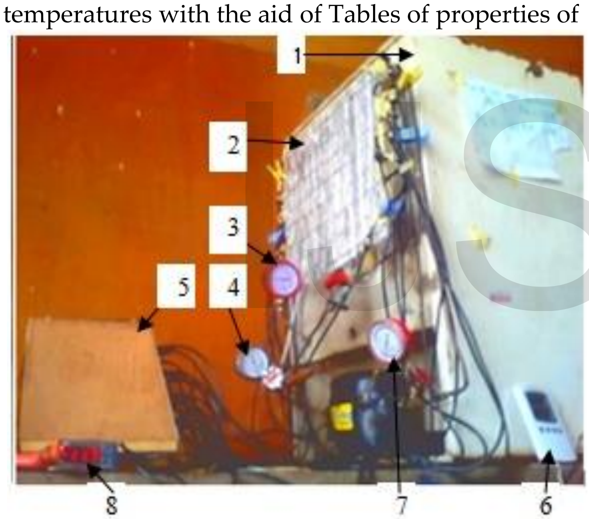


**Figure 1:** Experimental Test Rig

**Legend:** Refrigerating system (1); developed condenser (2);

Low pressure Gauge, compressor suction (3);

High pressure Gauge, Compressor discharge (4);

Temperature-data-logger (5); Thermo-Hygrometer (6);

High pressure Gauge, Condenser exit pressure (7);

Micro-computer temperature controller (8**)**

### 3. Results and Discussions

Tables 1 - 5 show the statistical results of the experimental data obtained in the course of evaluating relevance of the variations of measured thermodynamic parameters (refrigeration system's box, compressor suction, compressor discharge, condenser exit and prevailing ambient

the refrigerant (R600a). These measured values, after being expressed in unit of temperature, were then statistically analyzed with Analysis of Variance (ANOVA) technique to evaluate effects of condenser heat augmenting surface geometries and refrigerating sample thermal loads on the performance of the components of the refrigerating system. The statistical relations were evaluated based on two hypotheses (null and alternate):

**(i) Null Hypothesis ($H_o$):**

No statistically significant difference exists among measured parameters, condenser's augmentation and thermal loads

**(ii) Alternate Hypothesis ($H_a$):**

Statistically significant difference exists among measured parameters, condenser's augmentation and thermal loads

The statistical interpretation of these relations was done using P-value. The pictorial description of the set-up is presented in figure. 1.

temperatures) with condenser heat enhancing surface geometries and heat contents (thermal loads) of the refrigerating items to refrigerating system performance when used to refrigerate 2.0 kg of water sample for period of 8 hours per experimentation. These data presents quantitative analyses of the thermal responses of each of the measured thermal parameters to the

Table 1 shows the responses of the Box (cabinet) temperatures with fins configurations and thermal loads. From this Table, it was observed that at start-up the computed p-value is greater than the critical value ($\sigma > 0.05$); the Null hypothesis ($H_{o1}$) is accepted. This uniformity in the thermal response of the fins arose because at this period, the system has not been excited, no heat is available in the condenser to be rejected; thus, as at this stage, the evaporator functioned in isolation. The heat is only retained within the target box, no noticeable quantity of this heat is transferred in the system. For the first-three hours of the experimentation, the computed value is less than the critical value ($\sigma < 0.05$); the alternative hypothesis ($H_{a1}$) is therefore accepted. This difference is due to the fact that the refrigeration system (Test rig) is hot-pulled and considered to be working outside its design conditions; thus, more heat load is available in the target box to be absorbed by the refrigerant and rejected in the condenser, this accounted for the tremendous (high) increase in the temperature difference recorded during this period of refrigeration and hence, each of the heat enhancing configurations behaved differently in achieving the objective. At duration of 4 hours and above, the computed p-value is greater than the alpha critical ($\sigma > 0.05$) again; this indicated that there is no statistical significant difference in the thermal responses. Thus, the Null hypothesis ($H_{o1}$) is accepted. The uniformity in the thermal responses of these two factors occurred because at this period the system is working close to its design conditions; this could be traced to reduction in the superheat, temperature difference and gradual increase in the subcool

**Table 1**: Box temperatures with fin configurations and thermal Loads

| S/N | Period (hr) | ANOVA | | | F-critical value (8.85) | | | |
|---|---|---|---|---|---|---|---|---|
| 1 | Cold start | **Source** | **DF** | **Seq SS** | **Adj SS** | **Adj MS** | **F-Value** | **P-Value** |
| | | A | 8 | 16.52 | 16.52 | 2.0644 | 2.15 | 0.070 |
| | | B | 3 | 37.05 | 37.05 | 12.3485 | 12.86 | 0.000 |
| 2 | 1 | **Source** | **DF** | **Seq SS** | **Adj SS** | **Adj MS** | **F-Value** | **P-Value** |
| | | A | 8 | 143.5 | 143.5 | 17.940 | 2.71 | 0.028 |
| | | B | 3 | 253.9 | 253.9 | 84.640 | 12.78 | 0.000 |
| 3 | 2 | **Source** | **DF** | **Seq SS** | **Adj SS** | **Adj MS** | **F-Value** | **P-Value** |
| | | A | 8 | 143.5 | 143.5 | 17.940 | 2.71 | 0.028 |
| | | B | 3 | 253.9 | 253.9 | 84.640 | 12.78 | 0.000 |
| 4 | 2 | **Source** | **DF** | **Seq SS** | **Adj SS** | **Adj MS** | **F-Value** | **P-Value** |
| | | A | 8 | 65.00 | 65.00 | 8.125 | 2.70 | 0.029 |
| | | B | 3 | 36.66 | 36.66 | 12.221 | 4.06 | 0.018 |
| 5 | 4 | **Source** | **DF** | **Seq SS** | **Adj SS** | **Adj MS** | **F-Value** | **P-Value** |
| | | A | 8 | 131.69 | 131.69 | 16.46 | 1.23 | 0.324 |
| | | B | 3 | 30.00 | 30.00 | 10.00 | 0.75 | 0.534 |
| 6 | 5 | **Source** | **DF** | **Seq SS** | **Adj SS** | **Adj MS** | **F-Value** | **P-Value** |
| | | A | 8 | 60.210 | 60.210 | 7.526 | 1.91 | 0.105 |
| | | B | 3 | 3.712 | 3.712 | 1.237 | 0.31 | 0.815 |
| 7 | 6 | **Source** | **DF** | **Seq SS** | **Adj SS** | **Adj MS** | **F-Value** | **P-Value** |
| | | A | 8 | 73.44 | 73.44 | 9.180 | 1.65 | 0.162 |
| | | B | 3 | 14.98 | 14.98 | 4.993 | 0.90 | 0.456 |
| 8 | 7 | **Source** | **DF** | **Seq SS** | **Adj SS** | **Adj MS** | **F-Value** | **P-Value** |
| | | A | 8 | 70.14 | 70.14 | 8.768 | 1.78 | 0.130 |
| | | B | 3 | 47.51 | 47.51 | 15.835 | 3.22 | 0.040 |
| 9 | 8 | **Source** | **DF** | **Seq SS** | **Adj SS** | **Adj MS** | **F-Value** | **P-Value** |
| | | A | 8 | 69.052 | 69.052 | 8.632 | 2.05 | 0.083 |
| | | B | 3 | 5.059 | 5.059 | 1.686 | 0.40 | 0.754 |

Table 2 illustrates responses of the condenser exit temperatures with fins configurations and thermal loads. In case of effect of fins configurations, it was found that at cold-start the condenser tended to be in equilibrium with the surroundings temperature as no heat is available to be transferred by the system since it has not been switched on. The heat load is restricted to the evaporator. Hence, the $\sigma > 0.05$, and the Null hypothesis ($H_{o2}$) is accepted because the condenser responded equally at all the cases of the fin configuration. However, at excitation and throughout almost the experimentation period, the $\sigma < 0.05$ and hence the null hypothesis is rejected because there is a statistical significant difference in the performance of the condenser with respect to the arrangement of its extended surfaces. This difference arose as a result of the fact that the extended surfaces offered different surface areas to heat flow or effectiveness.

In case of the effect of thermal load on the thermal behaviours of the condensers, it was observed that: (i) at cold-start the heat load is within the target box (refrigerated space), no heat is transferred to other part of the system and no thermal relationship existed between the heat load together with the working fluid and the condenser, hence $\sigma > 0.05$, and the Null hypothesis ($H_{o2}$) is accepted; (ii) for the first-three hours after start-up, tremendous quantity of heat was absorbed in the evaporator and rejected

ISSN 2229-5518

in condenser to the condensing medium. This effect accounted for high condensing temperature and condenser split and the reduced degrees of refrigerant subcooling because the system is excited and worked under its intended design conditions, thus, $\sigma < 0.05$. the Null hypothesis $(H_{o2})$ is rejected, this observation aligned with the findings of Rani and Balachander (2008): that cumulated heat rejection rate increases with rise in condensing temperature and when condensation temperature increases latent heat to be removed reduces; More so this rise in condensing temperature as a result hot-pulling down condition gives rise to excessive heat load that tends to create abnormality in system performance according to Dick (2010), and (iii) at period of 4 hours and above, $\sigma > 0.05$, the Null hypothesis $(H_{o2})$ is accepted, as at this stage the system works close or within its design conditions

**Table 2**: Condenser exit temperatures with fins configurations and thermal loads

| S/N | Period (hr) | ANOVA | | | F-critical value (8.85) | | | |
|---|---|---|---|---|---|---|---|---|
| 1 | Cold start | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 72.44 | 72.44 | 9.055 | 0.97 | 0.480 |
| | | B | 3 | 18.40 | 18.40 | 9.132 | 0.66 | 0.580 |
| 2 | 1 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 311.5 | 311.5 | 38.935 | 5.65 | 0.000 |
| | | B | 3 | 375.7 | 375.7 | 125.219 | 18.16 | 0.000 |
| 3 | 2 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 311.5 | 311.5 | 38.935 | 5.65 | 0.000 |
| | | B | 3 | 375.7 | 375.7 | 125.219 | 18.16 | 0.000 |
| 4 | 3 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 507.6 | 507.6 | 63.45 | 3.51 | 0.008 |
| | | B | 3 | 150.8 | 150.8 | 50.27 | 2.78 | 0.063 |
| 5 | 4 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 317.72 | 317.72 | 39.71 | 3.93 | 0.004 |
| | | B | 3 | 41.14 | 41.14 | 13.71 | 1.36 | 0.280 |
| 6 | 5 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 192.715 | 192.715 | 24.089 | 2.10 | 0.076 |
| | | B | 3 | 7.907 | 7.907 | 2.636 | 0.23 | 0.874 |
| 7 | 6 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 181.01 | 181.01 | 22.626 | 1.79 | 0.129 |
| | | B | 3 | 19.46 | 19.46 | 6.485 | 0.51 | 0.678 |
| 8 | 7 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 402.7 | 402.7 | 50.33 | 2.49 | 0.040 |
| | | B | 3 | 194.4 | 194.4 | 64.78 | 3.21 | 0.041 |
| 9 | 8 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 266.23 | 266.23 | 33.279 | 2.95 | 0.019 |
| | | B | 3 | 15.60 | 15.60 | 5.199 | 0.46 | 0.712 |



Table 4 depicts responses of the compressor discharge temperatures with fins configurations and thermal loads. In the course of evaluating the contribution of the extended surfaces to the condenser discharge temperatures. It was observed that throughout the period of experimentation $\sigma > 0.05$ and as such no statistical difference was noticed, therefore, the Null hypothesis $(H_{o2})$ is accepted. This uniformity in the thermal response is due to the fact that compressor is a work utilizing devices, not a heat exchanger.

However, in case of the response of the compressor discharge temperatures to the thermal loads, it was observed that: (1) at cold start $\sigma > 0.05$ no statistical significant difference occurred as at this moment each of the component acted in isolation, no excitation occurred. Therefore, the Null hypothesis $(H_{o2})$ is accepted; (2) for the first-two hours start-up time $\sigma < 0.05$ the null hypothesis is rejected as statistical difference existed. The reason for the difference is that the heat load is high, the expansion valve needs to supply larger volume of refrigerant per cycle to remove the heat. Owing to the heat laden-nature of the refrigerant, the refrigerant boils off rapidly and causes both the suction pressure and temperature to rise. More so, there would be increase in the discharge pressure and temperature

ISSN 2229-5518

as more hot refrigerant would be circulated per cycle over the period to bring the system back to design conditions, and (3) after 3 hours from the start-up time $\sigma > 0.05$; no statistical significant difference occurred as at this moment the compressor discharge temperature, pressure ratio is proportionately reduced because the compressor is at its intended design conditions, working at low discharge pressure (temperature) due to corresponding falls in the box temperatures.

**Table 3**: Compressor discharge temperatures with Fins configurations and Thermal loads

| S/N | Period (hrs) | ANOVA | | | | | | |
|---|---|---|---|---|---|---|---|---|
| 1 | Cold start | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 109.63 | 109.63 | 13.70 | 0.91 | 0.524 |
| | | B | 3 | 35.89 | 35.89 | 11.96 | 0.80 | 0.509 |
| 2 | 1 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 567.2 | 567.2 | 70.90 | 2.10 | 0.076 |
| | | B | 3 | 373.6 | 373.6 | 124.55 | 3.69 | 0.026 |
| 3 | 2 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 567.2 | 567.2 | 70.90 | 2.10 | 0.076 |
| | | B | 3 | 373.6 | 373.6 | 124.55 | 3.69 | 0.026 |
| 4 | 3 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 328.35 | 328.35 | 41.043 | 1.62 | 0.171 |
| | | B | 3 | 13.13 | 13.13 | 4.378 | 0.17 | 0.914 |
| 5 | 4 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 246.39 | 246.39 | 30.80 | 1.17 | 0.354 |
| | | B | 3 | 61.88 | 61.88 | 20.63 | 0.79 | 0.513 |
| 6 | 5 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 162.0 | 162.0 | 20.25 | 0.86 | 0.564 |
| | | B | 3 | 163.3 | 163.3 | 54.44 | 2.30 | 0.102 |
| 7 | 6 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 2100.3 | 2100.3 | 262.5 | 0.84 | 0.575 |
| | | B | 3 | 806.3 | 806.3 | 268.8 | 0.86 | 0.474 |
| 8 | 7 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 457.26 | 457.26 | 57.157 | 1.34 | 0.270 |
| | | B | 3 | 26.32 | 26.32 | 8.774 | 0.21 | 0.891 |
| 9 | 8 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 173.4 | 173.4 | 21.68 | 0.53 | 0.823 |
| | | B | 3 | 169.4 | 169.4 | 56.47 | 1.38 | 0.273 |

Table 4 shows the thermal responses of the compressor suction temperatures with condenser extended surfaces configurations and the applied thermal loads of the refrigerated items. It was observed that:

(1) over the entire fins configurations $\sigma > 0.05$, the null hypothesis is accepted as there was no statistical difference between the contributions of each of the configurations of the condenser's extended surfaces to the compressor's suction temperatures this statistical uniformity arose due to the fact that compressor is a work utilizing device

(2) in case of relevance of the thermal loads, (i) at start-up $\sigma > 0.05$ and the null hypothesis is accepted because the compressor acted in isolation; (ii) for the first-three hours after start-up time $\sigma < 0.05$ and the Null hypothesis is rejected because at this period the system is hot-pulled which makes it to be working outside the design conditions as it has more heat load to be absorbed from the evaporator, more refrigerant to be metered by the expansion valve to achieve the refrigeration process, and increase in suction pressure (temperature) or work of compression is required to achieve the pumping action of the refrigerant, and (iii) at after four hours $\sigma > 0.05$, and the null hypothesis is accepted because there was no statistical difference

**Table 4:** Compressor suction temperatures with Fins configurations and Thermal loads

| S/N | Period (hrs) | ANOVA | | | | | | |
|---|---|---|---|---|---|---|---|---|
| 1 | Cold start | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 109.50 | 109.50 | 13.69 | 0.91 | 0.523 |
| | | B | 3 | 35.75 | 35.75 | 11.92 | 0.79 | 0.509 |
| 2 | 1 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 415.4 | 415.4 | 51.92 | 1.75 | 0.138 |
| | | B | 3 | 883.2 | 883.2 | 294.41 | 9.92 | 0.000 |
| 3 | 2 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 415.4 | 415.4 | 51.92 | 1.75 | 0.138 |
| | | B | 3 | 883.2 | 883.2 | 294.41 | 9.92 | 0.000 |
| 4 | 3 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 6068 | 6068 | 758.6 | 3.02 | 0.017 |
| | | B | 3 | 2183 | 2183 | 727.6 | 2.89 | 0.056 |
| 5 | 4 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 4948 | 4948 | 618.6 | 1.55 | 0.193 |
| | | B | 3 | 3163 | 3163 | 1054.3 | 2.64 | 0.073 |
| 6 | 5 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 1876.5 | 1876.5 | 234.6 | 0.67 | 0.711 |
| | | B | 3 | 370.4 | 370.4 | 123.5 | 0.35 | 0.787 |
| 7 | 6 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 1838.7 | 1838.7 | 229.8 | 1.27 | 0.303 |
| | | B | 3 | 704.0 | 704.0 | 234.7 | 1.30 | 0.297 |
| 8 | 7 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 1710.5 | 1710.5 | 213.8 | 0.35 | 0.938 |
| | | B | 3 | 803.9 | 803.9 | 268.0 | 0.43 | 0.730 |
| 9 | 8 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 1882.2 | 1882.2 | 235.3 | 1.39 | 0.252 |
| | | B | 3 | 615.8 | 615.8 | 205.3 | 1.21 | 0.327 |

From Table 5, it was observed that after the start-up, the variation of the prevailing temperature with condenser's heat enhanced surfaces became statistically different for the first-two hours of the experimentation and non-significant different for second-six hours of it. It was also observed that statistical difference occurred with respect to the influence of prevailing temperature and thermal loads for the first-hours of the experimentation. The statistical significance in the variation of the performance of the refrigeration system due to these factors is due to the fact that as the heat load increases the heat rejection rate increases and more heat is ejected to the condensing fluid which then accounted for the significance difference in the prevailing temperatures

**Table 5** Prevailing ambient temperature with Fins configurations and Thermal loads

| S/N | Period (hr) | ANOVA | | | F-critical value (8.85) | | | |
|---|---|---|---|---|---|---|---|---|
| 1 | Cold start | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 1.845 | 1.845 | 0.2306 | 0.62 | 0.753 |
| | | B | 3 | 22.216 | 22.216 | 7.4055 | 19.91 | 0.000 |
| 2 | 1 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 47.72 | 47.72 | 5.965 | 4.02 | 0.004 |
| | | B | 3 | 41.89 | 41.89 | 13.963 | 9.41 | 0.000 |
| 3 | 2 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 47.72 | 47.72 | 5.965 | 4.02 | 0.004 |
| | | B | 3 | 41.89 | 41.89 | 13.963 | 9.41 | 0.000 |
| 4 | 3 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 36.56 | 36.56 | 4.569 | 2.15 | 0.070 |
| | | B | 3 | 69.00 | 69.00 | 23.000 | 10.82 | 0.000 |
| 5 | 4 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 55.56 | 55.56 | 6.944 | 2.23 | 0.061 |
| | | B | 3 | 56.08 | 56.08 | 18.694 | 6.01 | 0.003 |
| 6 | 5 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 42.50 | 42.50 | 5.313 | 1.16 | 0.360 |
| | | B | 3 | 66.75 | 66.75 | 22.250 | 4.88 | 0.009 |
| 7 | 6 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | A | 8 | 44.50 | 44.50 | 5.563 | 1.14 | 0.374 |
| | | B | 3 | 47.22 | 47.22 | 15.741 | 3.22 | 0.041 |
| 8 | 7 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 27.22 | 27.22 | 3.403 | 0.47 | 0.867 |
| | | B | 3 | 15.44 | 15.44 | 5.148 | 0.71 | 0.557 |
| 9 | 8 | Source | DF | Seq SS | Adj SS | Adj MS | F-Value | P-Value |
| | | A | 8 | 26.06 | 26.06 | 3.257 | 0.81 | 0.601 |
| | | B | 3 | 17.64 | 17.64 | 5.880 | 1.46 | 0.250 |

## 4. Conclusion

The study focused on evaluating the statistical relevance of variation in configurations of extended surfaces geometries used to enhance the heat transfer rate of condenser and inherent thermal contents/ loads of refrigerating item on the thermal properties for estimating the performance of components of vapour compression refrigeration systems. On the basis of this research the following can be concluded:

(i) the cooling behaviour , a function of the box temperature, changes with configuration of the condenser heat augmenting surfaces,
(ii) condenser performance is strongly dependent of configurations of its extended surfaces geometries,
(iii) compressor performance (suction and discharge) is independent of configuration of condenser, but appreciably dependent of thermal loads before the system reaches its design working conditions
(iv) prevailing ambient temperatures of refrigerating system vary with both the configuration of the condenser and inherent thermal loads of the refrigerating sample

It is clear that configurations of extended surfaces of condenser and inherent heat loads of refrigerating item have different influence on the performance of the various components of refrigerating systems

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