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Original Research Article

Development of an Integrated Refrigeration and Air – Conditioning System

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ABSTRACT

In this study, a vapour compression refrigeration and an air conditioning system were designed separately using relevant thermodynamic equations and were integrated into a single unit. The refrigerating effect (R_e), theoretical work done per cycle (W_d) and the refrigerant mass flow rate (M_r) for the refrigerating system were determined using heat and energy analysis based on a T-S diagram. The compressor power required was estimated based on the T-S diagram for the system. The cooling capacity, Energy Efficiency Ratio (EER) and actual work done per cycle of the integrated unit were estimated. The actual work done per cycle of the integrated unit was also compared with the power input and their Coefficient of Performance (COP) evaluated. The cost of development of the integrated unit was then estimated. The R_e, W_d and M_r for the refrigerating system were 136.13 kJ/kg, 1.215 kW and 0.0286 kg/s, respectively. The total sensible and latent heat were 2876.10 and 614.5 W, respectively. The grand total heat generated by the hypothetical space was 3.5 kW. The estimated required compressor power was 1.5 HP. The cooling capacity and EER ranged from 3.707 to 2.63 kW and 4.572 to 3.241, respectively. The actual work done per cycle for the power input of 0.574, 0.794, 0.958, 1.052 and 1.144 kW were 1.959, 2.709, 3.267, 3.587 and 3.902 kW, respectively. The COP of air-conditioner and refrigerator ranged from 0.95 to 1.34 and from 4.63 to 20.63, respectively. The estimated cost of the developed integrated unit was ₦55,080.

1. INTRODUCTION

Globally, the air around us is a mixture of gases, mainly nitrogen and oxygen. It also contains smaller amounts of water vapor, argon, carbon dioxide and very small amount of other gases. Air is a precious resource, as it supplies us oxygen, which is essential for our bodies to live. Without it, we would die within minutes (Shakhashiri, 2007). It may be divided into natural and treated air. Natural air is supplied by nature and absorbed by human body system without any treatment while treated air is subjected to a conditioning system. Air-conditioning is the process of treating air so as to control simultaneously its temperature, humidity, cleanliness and distribution to meet the requirements of the conditioned space (Laxman et al., 2017). The air in a room needs to be cooled or heated, humidified or dehumidified, purified and circulated depending on the prevailing ambient conditions in order to keep the space conducive for occupants (Obanor and Egware, 2013). Maintaining this standard of thermal comfort for occupants of buildings or other enclosures is one of the important goals of HVAC (heating, ventilation and air conditioning) engineers (Varkute et al., 2016). Refrigeration works by removing heat from a product and transferring the same to the outside air (Scienceabc, 2015).

Wang and Lavan (1999) classified air conditioning system into individual, space-conditioning, packaged and central system. The major parts of refrigeration and air conditioning system that can be used to manage refrigerant and move air in two directions (indoors and outdoors) are the compressor (pump that pressurizes the refrigerant), expansion valve (to regulate refrigerant flow into the evaporator), evaporator (to receive the liquid refrigerant) and condenser (to facilitate heat transfer) (Satyam et al., 2016). Danfoss (2007) showed that the temperature and pressure at the compressor outlet are 60 ^{o}C and 7.6 bars and -10 ^{o}C and 1 bar at evaporator unit. In buildings, there are seven basic cooling loads, five of which are related to the envelope: roof, walls, floor, windows and infiltration. The other two loads are duct gains and internally generated heat. Internal heat gain can be attributed to the heat and moisture produced directly by the occupants as well as the appliance activities associated with daily living (Chasar, 2004).

In the last three decades, there has been a surge in energy demands due to increase in demand for refrigeration and air conditioning systems (Naikodi et al., 2014). In the 21st century, refrigeration and air conditioning system consume 30% of electricity worldwide (Markad et al., 2018). This has led to increase in pollution and energy cost that cannot be afforded by low income earners. Saving energy reduces production cost and also improves the living standard of the people (He, 2010). In an attempt to address the problem of power consumption, procurement and maintenance cost, developing an integrated cooling system is a step in the right direction. Therefore, in this study, an air-conditioning and a refrigeration system were designed, integrated into a fabricated single unit cuboidal frame and their performance was evaluated.

2. MATERIALS AND METHODS

2.1. Description of Designed System

The whole assembly is made up of two units which are; refrigerator unit and air conditioning unit. The refrigerant R – 134a in the system is compressed by the compressor, leaves as high-pressure refrigerant and travels under pressure through coils or tubes that make up the condenser, where it changes to liquid. The stream of liquid refrigerant is then divided into two through two metering or throttling devices to evaporator A (air conditioning unit) and evaporator B (refrigeration unit). The refrigerant from both units converges at the accumulator and is then delivered to the compressor at low pressure for another cycle, as shown in Figure 1. The temperature – entropy diagram for this cooling system (integrated single unit refrigeration and air-conditioning system) is shown in Figure 2.

2.2. Heat and Energy Analysis of the Designed Integrated Cooling System

The temperature of the refrigerant was determined at stages 1, 2, 3 and 4 before transferring the values to the temperature-entropy (t-s) diagram shown in Figure 2. Using steam table, the values of temperatures obtained at these stages were shown in Table 1. These values were used to calculate the refrigerating effect, theoretical work done per cycle, coefficient of performance (COP), mass flow rate of refrigerant, refrigerating capacity, compressor power, the speed of the electric motor and tonnages of the refrigerant used for the design.

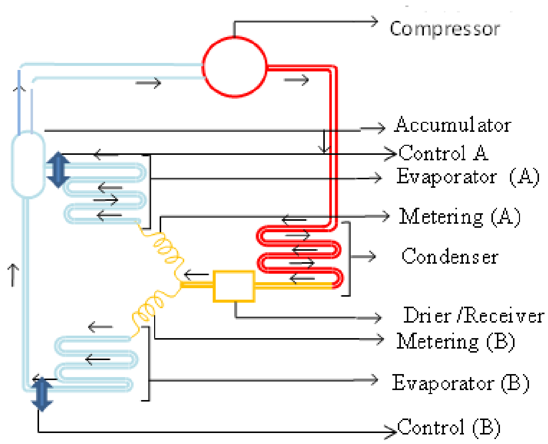

Figure 1: Plant-flow diagram for the integrated cooling system

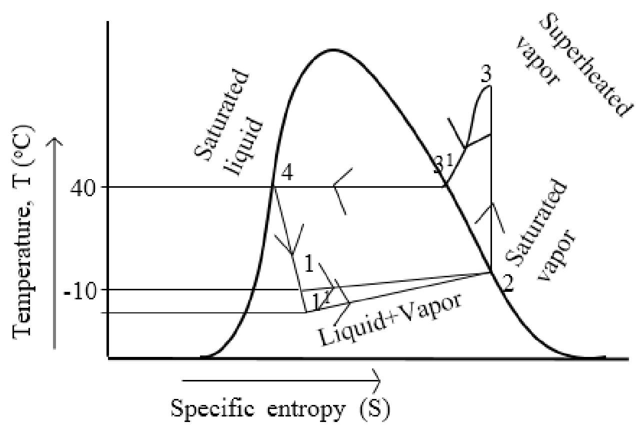

Figure 2: Temperature-entropy (t-s) diagram for integrated cooling system

2 to 3 = Compression of vapor

3 to3^1 = Vapor superheat removed in condenser

3^1 to 4 = Vapor converted to liquid in condenser

4 to 1 = Liquid flashes into liquid + vapor across expansion valve

1 to 2 = Liquid + vapor converted to vapor in evaporator (refrigerator unit)

1^1 to 2 = Liquid + vapor converted to vapor in evaporator (air conditioning unit)

Table 1: Values obtained from the steam table (Haywood, 1968)

Temperature T (°C)	Pressure p (MPa)	Specific volume V_g (m^3/kg)	Enthalpy (kJ/kg)		Entropy (kJ/kg)	
			h_f	h_g	s_f	s_g
-10	0.2006	0.0995	186.71	392.51	0.9506	1.7327
40	0.0163	0.0200	256.38	419.41	1.1903	1.7109

The specific heat of R134a at constant pressure is 0.90 kJ/kgK (ASHRAE, 2009).

From Figure 2 and Table 1:

$h_1 = h_{f4}$ = 256.38 kJ/kg, $h_2 = h_{g2}$ = 392.51 kJ/kg and $s_3 = s_2$ = 1.7327 kJ/kg

T_{3^1} = 273 + 40 = 313 K

$$s_3 = s_{3^1} + cp_s \ln\left(\frac{T_3}{T_{3^1}}\right)$$ (Chauragade et al., 2016) (1)

Since $s_3 = s_2$:

$$s_2 = s_{3^1} + cp_s \ln\left(\frac{T_3}{T_{3^1}}\right)$$ (2)

Using:

$$h_3 = h_{3^1} + cp_s(T_3 - T_{3^1})$$ (3)

Refrigerating effect (Re) is the difference between the heat at compression stage and evaporation stage in the cycle. It was estimated by:

$$R_e = h_2 - h_1$$ (Sahoo and Rout, 2016) (4)

Where h = enthalpy, h_f = enthalpy of liquid, h_g = enthalpy of gas, s = entropy, s_f = entropy of liquid, s_g = entropy of gas, T = refrigerant temperature, cp_s = specific heat capacity of the refrigerant at saturation.

Work done (Wd) per cycle was expressed as:

$$W_d = h_3 - h_2$$ (Arora, 2007) (5)

Coefficient of performance (*COP*) was calculated as

$$COP = \frac{Refrigerating\ effect\ (Re)}{Work\ done\ per\ cycle}$$ (Gonnade et al., 2018) (6)

Mass of refrigerant (*Mr*) circulated was given as:

$$Mr = \frac{14000}{(h_2 - h_1) \times 3600}$$ (Rajput, 2012) (7)

Refrigerating capacity (*Rc*) was calculated as:

$$R_c = mass\ flow\ rate\ \times refrigerating\ effect$$ (Akusu et al., 2018) (8)

$$= Mr \times (h_2 - h_1)$$

Compressor power *(Cp)* was given as:

$$Cp = mass\ flow\ rate\ \times work\ done\ per\ cycle$$ (Rajput, 2012) (9)

i.e. $Cp = m \times (h_3 - h_2)$

Capacity of the plant (Pc) was calculated as:

$$Pc = Number\ of\ tonnes\ of\ refrigrant\ \times 14000\ kJ/h \quad (10)$$

Which implies that:

$$Number\ of\ tonnes\ of\ refrigerant = \frac{Pc}{14000\ kJ/h}$$

Rating of the compressor electric motor (N) was expressed as:

$$N = \frac{60P}{2\pi T} \quad \text{(Khurmi and Gupta, 2006)} \quad (11)$$

$$T = Force\ \times radius \quad \text{(Daniel, 2015)} \quad (12)$$

Where p and T are power and torque, respectively

2.3. Cooling Load Calculation for the Hypothetical Space

In air-conditioning, the cooling load can be classified as room load (load that fall on the room directly) and grand total load (load that fall on the air conditioning apparatus). Grand total load / heat may be represented mathematically as:

$$GTH = TSH + TLH \quad \text{(Gaadhe et al., 2019)} \quad (13)$$

Where GTH, TSH and TLH are Grand Total Heat, Total Sensible Heat and Total Latent Heat, respectively.

2.4. Determination of the Thermophysical Properties of the Hypothetical Space

The hypothetical space was divided into wall, floor, door and window in order to determine the effective thermophysical properties of the space.

2.4.1. Wall

Figure 3 shows the wall structure of the hypothetical space and its overall heat transfer coefficient (U_{wall}) was estimated by:

$$\frac{1}{U_{wall}} = \frac{1}{f_0} + \frac{\Delta x_o}{k_o} + \frac{1}{c_b} + \frac{\Delta x_i}{k_i} + \frac{1}{f_i} \quad \text{(Rajput, 2012)} \quad (14)$$

Where: k_o = thermal conductivity of outside plaster, Δx_0 = thickness of the outside plaster, k_i= thermal conductivity of the inside plaster, Δx_i= thickness of the inside plaster, c_b = Conductance of the building block, f_0 = film co-efficient on outside wall, f_i = film co-efficient on inside wall.

2.4.2. Roof

Figure 4 shows the roof structure of the hypothetical space and its overall heat transfer coefficient (U_r) was estimated by:

$$\frac{1}{U_r} = \frac{1}{f_0} + \frac{1}{C_{ar}} + \frac{1}{C_w} + \frac{1}{C_a} + \frac{1}{f_i} \quad \text{(Rajput, 2012)} \quad (15)$$

Where C_{ar}= Conductance of abestores roofing sheet, C_w= Conductance of roofing wood, C_a= Conductance of asbestos.

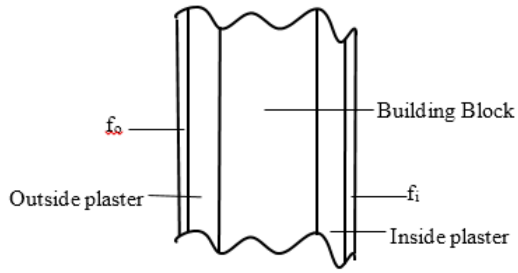

Figure 3: Hypothetical space wall structure

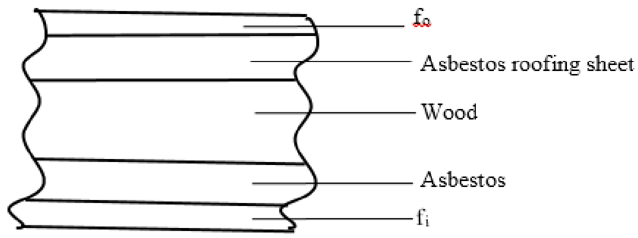

Figure 4: Hypothetical space roof structure

2.4.3. Floor

Figure 5 shows the floor structure of the hypothetical space and its overall heat transfer coefficient (U_f) was calculated by

$$\frac{1}{U_f} = \frac{1}{c_t} + \frac{1}{c_c} + \frac{1}{f_i} \text{ (Rajput, 2012)} \tag{16}$$

Where c_t = Conductance of the floor tile, c_c = Conductance of the floor concrete

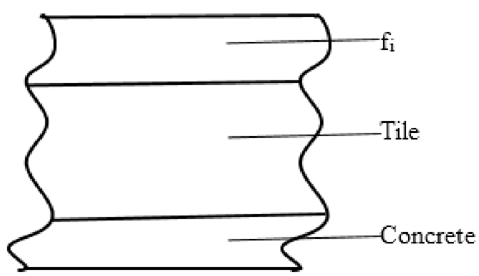

Figure 5: Hypothetical space floor structure

2.4.4. Door

Figure 6 shows the door structure of the hypothetical space and its overall heat transfer coefficient (U_d) was estimated by:

$$\frac{1}{U_d} = \frac{1}{f_0} + \frac{\Delta_{xoa}}{k_a} + \frac{\Delta_{xia}}{k_a} + \frac{1}{f_i} \text{ (Arora, 2007)} \tag{17}$$

Where k_a = thermal conductivity of aluminum plate, Δ_{xoa} = outside aluminum plate thickness, Δ_{xia} = inside aluminum plate thickness

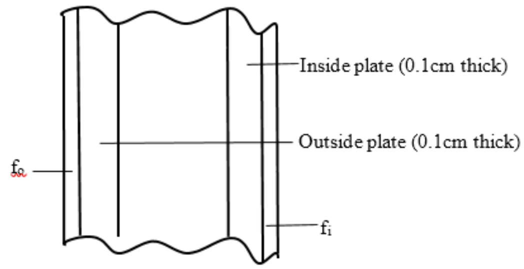

Figure 6: Hypothetical space door structure

2.4.5. Window

Figure 7 shows the window structure of the hypothetical space and its overall heat transfer coefficient of the window (U_w) was estimated by:

$$\frac{1}{U_w} = \frac{1}{f_o} + \frac{\Delta_{xg}}{k_g} + \frac{1}{f_i} \quad \text{(Rajput, 2012)} \quad (18)$$

Where k_g = thermal conductivity of glass, Δ_{xg} = thickness of the glass

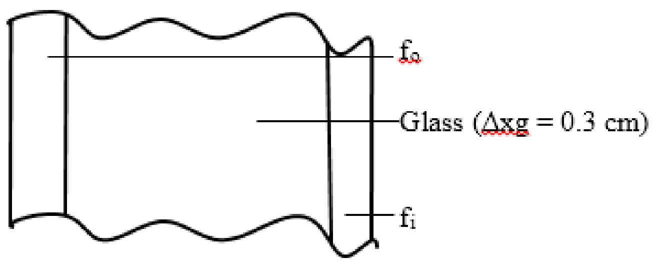

Figure 7: Hypothetical space window

2.5. Hypothetical Space Design Parameters

The hypothetical space dimensions used for this work were measured and are given as follows: Length (L) - 5.10 m, Breadth (B) -3.15 m, Height (H) - 3.06 m, Door size - 2.1 m × 0.9 m, Windows sizes were 1.8 m × 1.2 m and 1.2 m × 1.2 m. Solar Heat Gain Factors (SHGF) for south and north glasses are 104 and 44 W/m^2, respectively (Arora, 2007). Table 2 shows the values of the overall heat transfer coefficients for walls, roof, floor, door and windows calculated at outdoor conditions of 32 ^oC dry bulb temperature (dbt) and relative humidity of 0.026 kg/kg and indoor conditions of 24 ^oC dbt and relative humidity of 0.009 kg/kg, Sensible and latent heat loads per person are 59 and 73 W, respectively (Rajput, 2012), Ventilation requirements per person = 0.071 m^3/s (Rajput, 2012). According to Varkute et al. (2016), the minimum lighting load for office is 20 W/m^2. From Table 2, the total sensible heat gain through the wall was also estimated as 2080.45 W. Equations 19-29 were used to calculate the various parameters that helped in estimating the total cooling load.

Table 2: Estimation of sensible heat gain

Source	Overall heat transfer coefficient, U (W/m^{2}K)	Area -A (m^2)	Equivalent temperature difference (t_e) ^{o}C	Sensible heat gain = UAt_e (W)
North wall	2.93	15.61	8	365.90
South wall	2.93	12.28	8	287.84
East wall	2.93	9.639	8	225.94
West wall	2.93	9.639	8	225.94
Roof	3.18	16.065	8	408.69
Floor	2.11	16.065	8	271.18
Door	6.803	1.89	8	102.86
Window at the North wall	6.67	2.16	8	115.26
Window at the South wall	6.67	1.44	8	76.84
Total sensible heat gain			8	2080.45

$$Sensible\ Heat\ Gain = U \times A \times t_e \quad \text{(Tin, 2009)} \tag{19}$$

Solar Heat Gain through South Glass (SHGSG) was given as:

$$SHGSG = Area\ of\ window \times SHGF\ for\ South\ Glass\ (SG) \quad \text{(Rajput, 2012)} \tag{20}$$

Where *SHGF* is the Solar heat gain factor

Total sensible heat gain per persons (*Qs/p*) was given as:

$$Q_s/p = Sensible\ heat\ gain\ per\ person\ \times Number\ of\ persons \quad \text{(Rajput, 2012)} \tag{21}$$

Total latent heat gain per persons (*Ql/p*) was calculated as:

$$Ql/p = Latent\ heat\ gain\ per\ person \times Number\ of\ persons \quad \text{(Yonas, 2014)} \tag{22}$$

Amount of infiltrated air (V_{inf}) is calculated as:

$$V_{inf} = \frac{\text{L} \times \text{B} \times \text{H} \times \text{Number of air changes}}{60} \quad \text{(Rajput, 2012)} \tag{23}$$

Sensible heat gain due to infiltration air (Q_sV_{inf}) was estimated as:

$$Q_sV_{inf} = 0.02044\ V_{inf}(T_{outdoor} - T_{indoor}) \quad \text{(Rajput, 2012)} \tag{24}$$

Latent heat gain due to infiltration air (QlV_{inf}) was expressed as:

$$QlV_{inf} = 50\ V_{inf}(W_{outdoor} - W_{indoor}) \quad \text{(Rajput, 2012)} \tag{25}$$

Volume of ventilation (V_{ent}) was calculated as:

$$V_{ent} = 0.071\ m^3/s\ per\ person \times Number\ of\ persons \quad \text{(Rajput, 2012)} \tag{26}$$

Outside air sensible heat (*OASH*) is given as:

$$OASH = 0.02044\, V_{ent}(T_{outdoor} - T_{indoor})$$ (Rajput, 2012) (27)

Outside air latent heat (*OALH*) was express as:

$$OALH = 50\, V_{ent}(W_{outdoor} - W_{indoor})$$ (Rajput, 2012) (28)

Sensible heat gain due to lighting / Total Wattage (*TW*) was given as:

$$TW = Lighting\ load \times Total\ floor\ area$$ (Arora, 2007) (29)

The total cooling load, GTH (TSH + TLH), on the air conditioning unit was 3.5 kW.

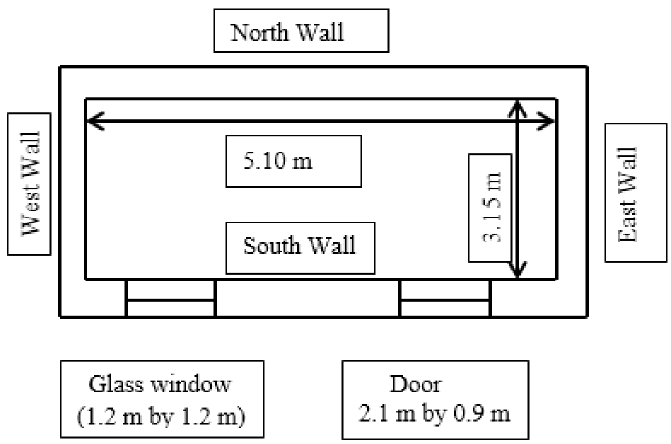

Figure 8: Plan of an hypothetical space (office)

The plan for the selected hypothetical space to be cooled is shown in Figure 8 which is of dimension 5.10 × 3.15 × 3.06 m and having the capacity to contain six to eight people.

2.6. Performance Evaluation of the Developed Integrated System

The performance of the developed integrated refrigeration and air-conditioning system was evaluated by placing the integrated unit inside the hypothetical space (5.10 × 3.15 × 3.06 m) with door and window closed. The heating units (compressor and condenser) were placed outside. The system was allowed to work for 30 mins and Humidity Temperature Clock (HTC) was placed at the middle of the hypothetical space. The initial temperature of the hypothetical space was measured and after the system worked for 30 mins, the hypothetical space temperature was measured. The evaluation was then repeated for 60, 90, 120, 150 and 180 mins. The office temperature was measured and recorded in each case. To evaluate the performance of the refrigerating unit, the initial reading of HTC in the refrigerator was taken. The refrigerating unit door was closed and was allowed to work for 10 mins and the refrigerating unit door was open and HTC reading was obtained. The procedure was then repeated for 20, 30, 40, 50, 60, 70 and 80 mins. Equations 30 to 35 were used to carry out the performance evaluation of the integrated cooling system. Figure 9 shows the exploded view of the developed integrated cooling unit.

$$Q = mcp\Delta T$$ (30)

$$COP\ for\ air\ conditioning\ unit = \frac{12}{cooling\ load \times 3.412} \quad \text{(ThemePacific, 2013)} \tag{31}$$

$$COP\ for\ refrigerating\ unit = \frac{h_2 - h_1}{h_3 - h_2} = \frac{T_2}{T_1 - T_2} \quad \text{(Rajput, 2012)} \tag{32}$$

$$COP = \frac{Cooling\ capacity}{Work\ done\ per\ cycle} = \frac{Cooling\ effect\ in\ watt}{Work\ input\ in\ watt} \quad \text{(Khaled, 2011)} \tag{33}$$

$$EER = COP \times 3.142 \quad \text{(ThemePacific, 2013)} \tag{34}$$

$$EER = \frac{System\ Cooling\ Capacity}{Electrical\ Power\ rating\ of\ the\ cooling\ system} \quad \text{(Mavrides, 2017)} \tag{35}$$

No	Qty	Description
1	1	Back Insulator
2	1	Back Cover
3	1	Top Insulator
4	1	Top Cover
5	1	Fan
6	1	Front Door
7	1	Ground Cover
8	1	Ground Insulator
9	1	Front Insulator
10	1	Front Cover
11	1	Compressor
12	2	Side Insulator
13	1	Refr Evaporator
14	2	Side Cover
15	1	Separator
16	1	A/C Evaporator
17	2	Frame
Title		Exploded View

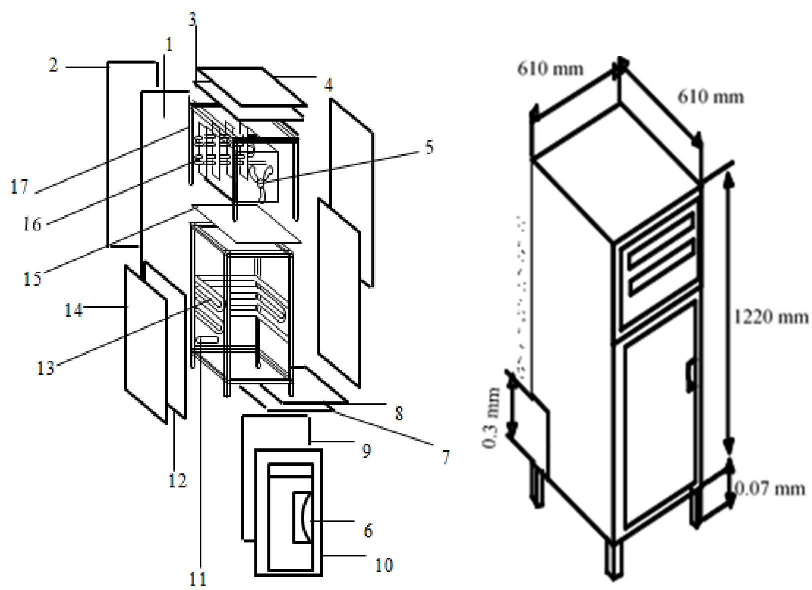

Figure 9: Exploded view of the integrated cooling system

3. RESULTS AND DISCUSSION

3.1. Integrated Refrigeration and Air-conditioning System

Plate 1 shows the fabricated integrated refrigeration and air-conditioning system of dimension 0.5 × 0.54 × 1.8 m. It was divided into two parts. The upper part is for the air-conditioning unit with dimension 0.5 × 0.54 × 1 m. It has a rectangular air net opening (0.35 × 0.28 m) for discharging cold air into the hypothetical space. The lower part of dimension 0.5 × 0.54 × 0.8 m is the refrigerating unit with a door of dimension 0.7 × 0.45 m. The system designed has a capacity of 1.5 Hp.

3.2. Evaluation of the Performance, *COP* and *EER* of Air-conditioning Unit

Figure 10 shows the cooling capacity of the air-conditioning system. The cooling capacity of air conditioning unit between 30 to 60 minutes is in agreement with the normal cooling capacity (3000 W) of a standard air-conditioning system with 1.5 Hp. Therefore, the effective cooling capacity of the air-conditioning unit was attained in 55 minutes. Figure 11 shows changes in *COP* and *EER* of the cooling system with time.

Plate 1: The developed single unit air-conditioning and refrigerating system

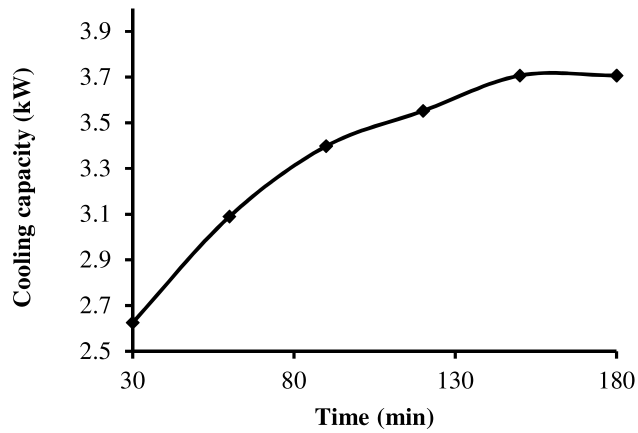

Figure 10: Cooling capacity of air conditioning system at different times

3.3 Evaluation of the Performance and *COP* for Refrigerating unit

Figure 12 shows the temperature drop with time in the refrigerating unit. The continuous decrease in temperature with time as the system worked shows that the refrigerating unit has the potential to keep any food item placed in it refrigerated.

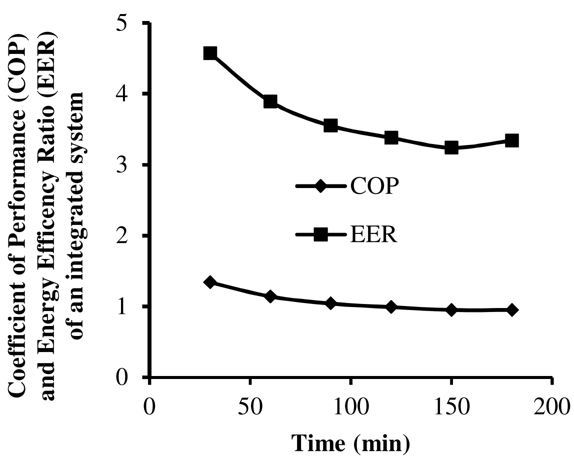

Figure 11: Comparison of COP and EER of air conditioning system at different times

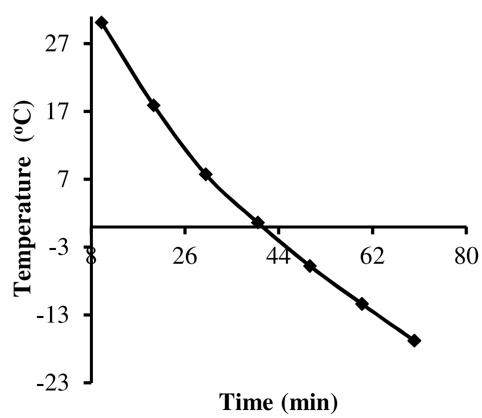

Figure 12: Temperature drop in the refrigerating unit

3.4. Evaluation of Power Consumption Rate of the Integrated Cooling System

Figure 13 shows the variation in the power consumed with work done per cycle. The power consumed varies from 0.57 to 1.14 kW and work done per cycle varies from 1.96 to 3.90 kW. This variation shows that the energy consumption rate of the integrated cooling system is smaller than the work done per cycle.

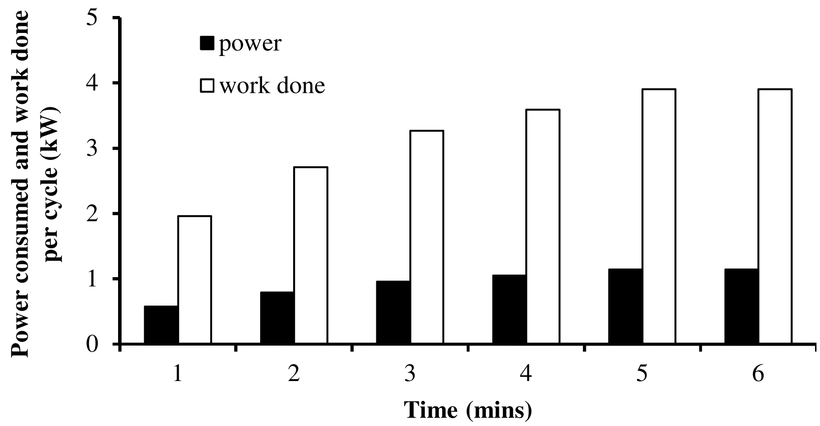

Figure 13: Comparison of the power consumed and work done per cycle

4. CONCLUSION

An integrated air-conditioning and refrigeration cooling unit has been designed and fabricated. The cooling capacity, *EER*, power consumption, *COP* and the work done per cycle of the integrated unit are within acceptable limits, although subject to improvement. With this design, energy consumption is reduced compared to having the air-conditioning and refrigeration systems as separate units. The production cost of this unit affords many low-income earners have access to it. For effective cooling, the compressor and condenser were placed outside of the space being conditioned.

5. ACKNOWLEDGMENT

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6. CONFLICT OF INTEREST

There is no conflict of interest associated with this work.

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