EXPERIMENTAL EVALUATION OF AIR-CONDITIONING SYSTEM

Tunji John Erinle*, Adebisi Olayinka Akinola, Olanrewaju Moses Adesusi, Idonorewrene Amudo Agbamu

Department of Mechanical Engineering, The Federal Polytechnic, Ado-Ekiti, Nigeria.
Department of Mechanical Engineering, The Federal University of Technology, Akure, Nigeria.
Department of Mechanical Engineering, The Federal University of Agriculture, Abeokuta, Nigeria.
*Corresponding Author E-mail: authenticfaithful@gmail.com,
E-mail (s): aoakinola@futa.edu.ng, adesusiolanrewajumoses@gmail.com, Idonorewreneagbamu@gmail.com

Abstract: This study focused on the experimental evaluation of air-conditioning system of its coefficient of performance. Performance of any vapour compression system is evaluated by the operational coefficient of performance (COP). The aim was to evaluate the effect of parameters that affect COP of refrigeration system. Measurements of important operational parameters such as the evaporator and condenser temperatures were studied and measured from the system. Other relevant parameters such as humidity of both the supply air and that of the space to be conditioned were also studied and measured. Experimental data were generated for 30 days to determine the performance of air-conditioning system. Performance evaluation of the system was determined in term of refrigerating capacity and COP. In this present study, load was estimated, refrigerating effect, work input and COP were estimated. The estimated results from the experimentation showed that the operational COP of the air-conditioning system range from 3.41 ± 0.02 to 4.98 ± 0.02. Efficiency requirement in Energy Efficiency Ratio (EER) for the air- conditioning system was estimated as 9.2. The coefficient of determination, R^2 value also showed that the variables being compared have strong correlation.

Keywords: Air-conditioning, COP, Effectiveness, Evaluation, Performance.

1. INTRODUCTION

The performance or effectiveness of air-conditioning system is determined by the coefficient of performance (COP) of the system. This experimental study was conducted to determine the performance of an air-conditioning system based on the operational COP. Data were collected from a typical office setting window unit air-conditioning system for some consecutive hot and humid period from ending of April to beginning of June for 30 working days. The experimental study involves measuring the temperature of important heat exchanger components such as evaporator and condenser.

Air-conditioning is used to control and maintain the temperature, humidity, air movement, air cleanliness, sound level and pressure differential in a space within the predetermined limits for the comfort and health of the occupants of the conditioned space and for the purpose of product processing (Wang, 2000; Jones, 2001; Richard & Gayle, 2007; Xiaohua, 2013; CCSW, 2013). The main factors that affect thermal comfort are temperature, relative humidity and air velocity. Thermal comfort is also an important consideration in evaluating the performance of air conditioning system (Harris, 1983).

Air-conditioning and refrigeration systems consume significant amount of energy in buildings and in process industries. The energy consumed in air conditioning and refrigeration systems is sensitive to load changes, seasonal variations, operation, ambient conditions and maintenance. Hence, the earlier mentioned factors should be taken into consideration in the performance evaluation of the system. The purpose of performance assessment is to verify the effectiveness and efficiency of a refrigeration system using field measurement parameters (Fulin et al., 2009; Othman et al., 2013; Vahid et al., 2014).

1.1 Coefficient of Performance (COP)

COP is highly dependent on operating conditions, especially absolute and relative temperatures between heat sink and system. For complete systems, COP should include energy consumption of all auxiliaries. COP of a refrigerating system has an expression as shown in equation (1) to (6) given by (Khurmi & Gupta, 2006).

$$\mathrm{COP_R} = \frac{\mathrm{Q_e}}{\mathrm{Q_c} - \mathrm{Q_e}} \quad (1)$$

$$\mathrm{W} = \mathrm{Q}_c - \mathrm{Q}_e \quad (2)$$

$$\mathrm{Q_e} = \mathrm{h_1} - \mathrm{h_4} \quad (3)$$

$$\mathrm{Q}_c = \mathrm{h_2} - \mathrm{h_3} \quad (4)$$

$$\mathrm{W} = \mathrm{h_2} - \mathrm{h_1} \quad (5)$$

$$\mathrm{COP_R} = \frac{\mathrm{Q_e}}{\mathrm{W}} \quad (6)$$

where**:** Q_c is the heat rejected by the condenser to the surrounding, (kJ)**,** Q_e is the heat absorbed from the evaporator, (kJ)**,** W is work input to the compressor by the system, (kJ), Enthalpy at the evaporator exit, $\mathrm{h_g} = \mathrm{h_1}$ (kJ/kg), Enthalpy at the compressor exit, $\mathrm{h_g} = \mathrm{h_2}$ (kJ/kg), Enthalpy at the condenser exit, $\mathrm{h_f} = \mathrm{h_3}$ (kJ/kg), Enthalpy at the evaporator entrance, $\mathrm{h_f} = \mathrm{h_4}$ (kJ/kg), Throttling process, $\mathrm{h_3} = \mathrm{h_4}$.

Qi et al. (2012) reported that air-conditioning system which plays an important role in creating the comforts of occupants were among of the largest energy consuming utilities in buildings. Furthermore, as most people spend more than 90 % of their time inside. Normally in buildings, more than 50 % of the energy is consumed by air-conditioning system (Yau, 2008; Enteria & Kunio, 2011). COP is one of the parameters used to monitor and evaluate the performance of air-conditioning systems in order to avoid unnecessary energy wastage due to its usage. Coefficient of Performance is the ratio of the refrigerating effect, which is amount of heat removed from the cooled space to the work input by the compressor of the system (Hosoi et al., 2006; Peng & Qungui, 2015; Sopian et al., 2013).

Yumoto et al. (2006) presented that the actual performance and energy consumption is quite different from the test performance by the experiments; in which the actual performance is changed by the load ratio of the system and outside temperature. In the Berman et al, (2017) study, the use of T-junction mode in air-conditioning system could generate an increase in the refrigerating effect and COP at a percentage rate which led to decrease in heat of compression with little percentage proportion. Due to the researchers' context, it had an implications for the lighter compressor work and also saving the power consumption of the system.
Moo-Yeon *et al*. (2012) studied the characteristic evaluation on the cooling performance of an electrical air-conditioning system using R744 for a fuel cell electric vehicle compared with conventional air-conditioning system using R-134a. The researchers affirmed that the cooling capacity and coefficient of performance for cooling of the former air-conditioning system tested were up to 6.4 kW and 2.5 better performances than the later respectively. Ali (2018a & 2018b) declared from its experimental investigation and performance enhancements on window type air-conditioning system with three capillary tubes with using of diffuser. It was affirmed that system with three capillary tubes had the highest COP from all other tested capillary tubes number with 14% increasing and overall pressure increased by 1.4 bar and 29 % COP increase from the base case of two tubes respectively. Berman *et al*. (2017) declared

that using T-junction installation on the air-conditioning system generates an increase in the refrigerating effect and coefficient of performance by 4 % and 10 % respectively.

The compressor to work easier would save electric power needed by the air-conditioning system. This has a significant implication for the lighter work of the compressor and low electrical power needed by the air conditioning systems (Berman et al., 2016). Abdul *et al.* (2017) affirmed that increase in the ambient air temperature has a negative effect on COP due to the decrease in the overall heat rejected, yet has a positive effect on refrigerant side pressure drop. Akintunde and Erinle (2015) studied the effects of temperature and humidity on coefficient of performance of air-conditioning system. It was revealed that the derived COP base on conditioned space temperature and relative humidity. It was affirmed that the derived COP from the researchers experiment gave the optimal coefficient of performance of 5.44.
This research centred on the experimental evaluation of an air-conditioning system with the capacity of 2 horse power (hp). The aim was to evaluate the effect of parameters that affect COP of refrigeration system. The type of air-conditioning system used for this study for the evaluation is window-type unit as shown in Fig. 1.

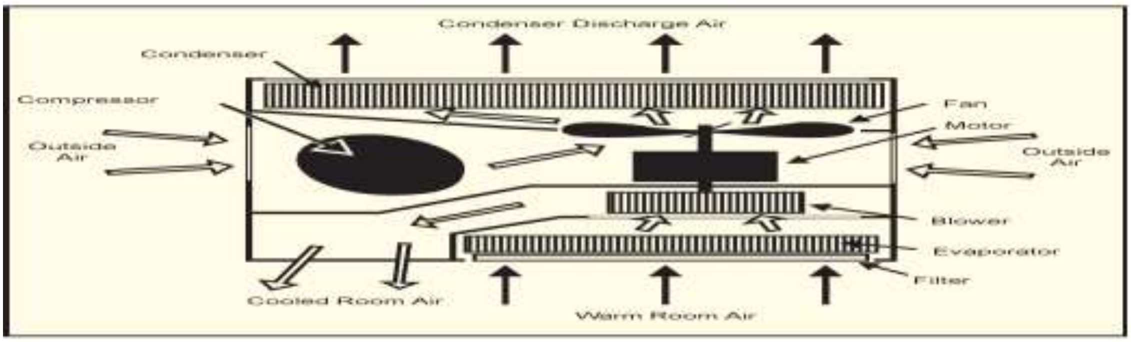

Fig. 1: Schematic top-view of typical air conditioning window type (Rosenquist, 1999).

2. MATERIALS AND METHODS

The exprerimental evaluation of the air-conditioning system was estimated using data collected in typical office setting at Mechanical Engineering Department of the Federal University of Technology, Akure, Ondo State, Nigeria. The area of the office used is 11.7 m^2 and volume is 46.8 m^3. The plan view of the office space used as shown in Fig. 2.

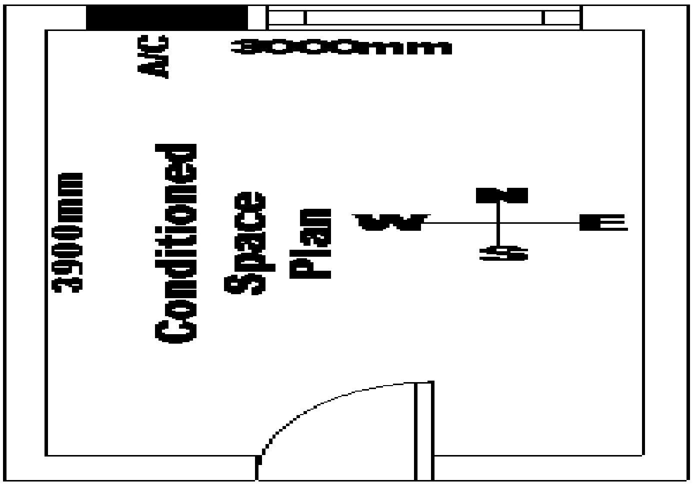

Fig. 2: Plan view of the office space used

2.1 Experimental Set-up and Procedures

The set up for the experimental study consists of window unit air-conditioning system which comprises of reciprocating compressor, the capacity of the compressor of the air-conditioning system used is 2 horse power, air cooled condenser, expansion device and coiled evaporator for cooling. The data collections for the experiment were done with the use of temperature and relative humidity measuring instruments. During the experimentation, the following parameters were measured: initial temperature of the conditioned room before conditioned, corresponding temperature of air-conditioning systems such as evaporator and condenser temperatures, conditioned space temperature, cooling coil and conditioned space relative humidity. The measured parameters were used to estimate psychometric properties of air in the room. Also, two digital hygrometers and thermometers with resolution of 0.1 ^{0}C and 1 % RH, measuring range of -10 ^{0}C to 70 ^{0}C, 20 % to 90 % RH (model-HH439) and -40 ^{0}C to 70 ^{0}C, 20 % to 90 %RH (model-ETH-1), uncertainty and expected % error of ± 1. One of these digital hygrometers was fitted to the air conditioning system to measure cooling coil relative humidity, while the other was mounted in the conditioned space in order to measure conditioned space relative humidity. Digital dual thermometers were fitted at the inlet and outlet pipe of the compressor to measure the evaporator temperature and with the help of sensors attached to the instrument to also measure the condenser temperature. Analogue thermometers were used to measure both the temperatures of the ambient and conditioned space. The air conditioning system was allowed to run for four hours and then the readings were recorded from the system for 4 hours at the interval of two hours were recorded for analysis. An experimental set-up for the performance evaluation of the air conditioning system, which consists of indoor and outdoor view as shown in Plate 1. Both the indoor and outdoor part composed of compressor, condenser, expansion valve, evaporator and other accessories.

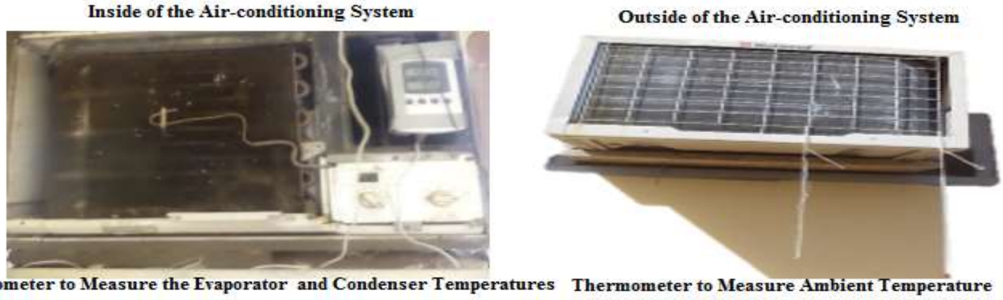

Plate 1: Experimental

2.2 Determination of the Cooling Load of the Air-Conditioning System

Air conditioning load was classified into two groups: these are sensible and latent heat load. Sensible heat load was assumed to have occurred due to any or all the following sources of heat transfers: heat flowing into the building by conduction through exterior walls, floors, ceilings, doors and windows due to the temperature difference on their two sides; heat received from solar radiation directly through glass windows, ventilators or doors and the heat absorbed by walls and roofs; heat given off by lights; heat liberated by the occupants; and heat carried by the outside air as result of infiltration and ventilation through the cracks in doors, windows and through their frequent openings. Latent heat load was assumed to have occurred due to any or all of the following sources of heat transfers: heat gain due to moisture in the outside air entering by infiltration and ventilation; and heat gain due to condensation of moisture from occupants. Equations (7) to (18) were used to estimate the cooling load by following (ASHRAE, 2001; Desai, 2012). The cooling load of the office was determined by using these parameters.

Space (Office) Data:
The Space dimension: Length of office = 3.9 m, Width of office = 3.0 m and Height of office = 4.0 m
Area of the Office Wall in the North and South Direction= $4.0 \times 3.0 = 12 \text{ m}^2$
Wall is made of bricks of equivalent thickness 0.225 m, 1.25 cm plaster on both sides
There is one window, 1.6 m x 0.6 m on the North side of the room.
There is a door 2.1 m x 0.9 m made of wood on the South Direction of the office.
1 bulb lamp rated 100 watts with 4 space hours operational time.

2.2.1 Cooling load from heat gain (sensible) through building structure (wall, door, floor, ceiling/roof)

$$Q = U A (t_a - t_r) \quad (7)$$

2.2.2 Cooling Load from heat gain due to infiltration and ventilation air (sensible and latent)

$$V_i = \frac{L \times W \times H \times G}{60} + \frac{\text{Door openings /hr} \times \text{Factor}}{60} \quad (8)$$

Total Infiltration, $V_i = V_c + V_d$ (9)
Load due to Outside Air by Infiltration
Outside air sensible heat, OASH= $20.43 V_i (t_a - t_r)$ (10)
Outside air latent heat, OALH = $50 V_i (W_a - W_r)$ (11)
Load due to Outside Air by Ventilation
Outside air sensible heat, OASH = $20.43 V (t_a - t_r)$ (12)
Outside air latent heat, OALH = $50 V (W_a - W_r)$ (13)

2.2.3 Cooling Load from heat gain due to conduction and solar radiation through glass window areas (Fenestration)

$$Q_{glass} = U_g A_g (t_a - t_r)_{convective} + A.GLF)_{radiative} \quad (14)$$

Cooling Load from Heat Gain from Occupants

Sensible heat gain, Q_S =Sensible heat gain factor, $Q_{SF} \times$ Number of occupants (15)
Latent heat gain, Q_L = Latent heat gain factor, $Q_{LF} \times$ Number of occupants (16)

2.2.4 Cooling Load from Heat Gain from Lighting Equipment, Q_{Light}

$$Q_{Light} = \text{Total wattage of light} \times \text{Use factor} \times \text{Allowance factor} \quad (17)$$

2.2.5 Cooling Load from Heat Gain due to Appliance

Sensible heat gain, Q_S =Recommended heat gain $\times$ Number of Appliance (18)

2.3 Sensible Heat Factor of the Room (Office) (RSHF)

The room sensible heat factor expresses the ratio between sensible heat load and total heat load in a room as mathematically shown in equation (19)

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{RSH}{RTH} \quad (19)$$

3. RESULTS AND DISCUSSION

3.1 Experimental Results

The observed values of evaporator and condenser temperatures which were taken for 30 working days at 4 hours per day in two hours intervals for morning and afternoon were averagely determined per week as presented in Tables 1 and 2.

Table 1: Evaporator and Condenser Temperature for Morning

Evaporator Temperature for Morning						
S/N	**t_e (°C) 1**	**t_e (°C) 2**	**t_e (°C) 3**	**t_e (°C) 4**	**t_e (°C) 5**	**t_e (°C) 6**
1	8	8	9	7	8	6
2	9	8	8	8	8	8
3	10	8	9	10	8	7
4	7	9	10	9	7	7
5	9	7	8	8	8	7
Average	**8.6**	**8**	**8.8**	**8.4**	**7.8**	**7**
Std. Error	**0.46**	**0.28**	**0.33**	**0.46**	**0.18**	**0.28**
Condenser Temperature for Morning						
S/N	**t_c (°C) 1**	**t_c (°C) 2**	**t_c (°C) 3**	**t_c (°C) 4**	**t_c (°C) 5**	**t_c (°C) 6**
1	42.7	42.7	44.3	46.3	40.4	42.4
2	46.1	42.7	44.3	48.9	46.3	42.4
3	46.1	39.6	48.3	46.3	46.3	42.4
4	46.1	53.1	53.1	48.9	40.9	42.4
5	46.1	44.3	44.3	39.1	42.4	42.4
Average	**45.42**	**44.48**	**46.86**	**45.90**	**43.26**	**42.40**
Std. Error	**0.61**	**2.04**	**1.56**	**1.61**	**1.15**	**0.00**

Table 2: Evaporator and Condenser Temperature for Afternoon

Evaporator Temperature for Afternoon						
S/N	**t_e (°C) 1**	**t_e (°C) 2**	**t_e (°C) 3**	**t_e (°C) 4**	**t_e (°C) 5**	**t_e (°C) 6**
1	8	8	8	8	8	6
2	8	9	9	8	8	8
3	10	9	9	10	9	7
4	7	9	10	10	7	8
5	7	8	8	8	7	8
Average	**8**	**8.6**	**8.8**	**8.8**	**7.8**	**7.4**
Std. Error	**0.49**	**0.22**	**0.33**	**0.44**	**0.33**	**0.36**
Condenser Temperature for Afternoon						
S/N	**t_c (°C) 1**	**t_c (°C) 2**	**t_c (°C) 3**	**t_c (°C) 4**	**t_c (°C) 5**	**t_c (°C) 6**
1	46.1	46.1	44.3	46.3	44.3	42.4
2	46.1	46.1	44.3	44.3	44.3	42.4
3	46.1	39.6	48.3	46.3	51.1	42.4
4	42.7	48.3	48.3	44.3	42.4	42.4
5	44.3	44.3	46.3	46.3	46.3	42.4
Average	**45.06**	**44.88**	**46.30**	**45.50**	**45.68**	**42.40**
Std. Error	**0.61**	**1.31**	**0.80**	**0.44**	**1.33**	**0.00**

The summary of the cooling load estimation for the office used for the experimental evaluation as presented in Table 3. The sensible heat factor for private office should be 0.9 according to Khurmi and Gupta. The sensible heat load and latent heat load generated within the office were 1380.21 W and 170.80 W respectively. The total heat load was 1551.01 W. The sensible heat factor estimated for the office used was 0.9.

Table 3: Estimation of the Cooling Load for the Office

Source of Heat Load	**Total Sensible Heat Gain (W)**	**Total Latent Heat Gain (W)**	**Total Heat Gain (W)**
North Wall	110.95		110.95
South Wall	101.61		101.61
Door	24.66		24.66
Floor	135.14		135.14
Ceiling/Roof	135.14		135.14
Fenestration	152.21		152.21
Infiltration	89.89	52.8	142.69
Ventilation	76.61	45	121.61
Occupant	59	73	132
Lighting	120		120
Equipment			
Appliance	375		375
Total	**1380.21**	**170.80**	**1551.01**

3.1.1 Performance Evaluation of Air-conditioning System

The average values of the refrigerating effect, work input and coefficient of performance (COP) per week for both morning and afternoon of the air-conditioning system as presented in Tables 4, 5 and 6. The Pressure-enthalpy diagram of refrigeration cycle as shown in Fig. 3. The enthalpy condition of the refrigerant at the inlet and exit of the condenser and evaporator were obtained from (Mclinden, et al., 1998) thermodynamics properties of R-22 refrigerant ($CHClF_2$).

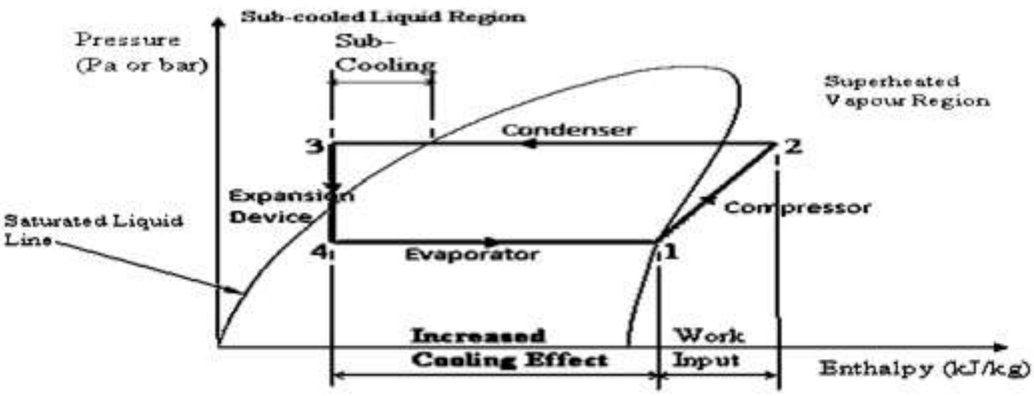

Fig. 3: P-h Diagram of Refrigeration (Mistral Associates, 2016)

Table 4: Refrigerating Effect and Work Input for Morning

Refrigerating Effect (kJ/kg) for Morning						
S/N	**Q_e 1**	**Q_e 2**	**Q_e 3**	**Q_e 4**	**Q_e 5**	**Q_e 6**
1	154.67	154.67	152.8	149.37	157.74	154.37

2	150.35	154.67	152.5	146.18	149.77	155.07
3	150.75	158.8	147.32	150.47	149.77	154.67
4	149.65	140.58	140.98	146.48	156.68	154.67
5	150.35	152.1	152.5	159.47	155.07	154.67
Average	**151.15**	**152.16**	**149.22**	**150.39**	**153.81**	**154.69**
Std. Error	**0.80**	**2.76**	**2.06**	**2.16**	**1.52**	**0.10**
Work Input (kJ/kg) for Morning						
S/N	**W 1**	**W 2**	**W 3**	**W 4**	**W 5**	**W 6**
1	34.48	34.48	35.25	37.29	32.92	34.97
2	36.45	34.48	35.55	38.63	36.89	34.27
3	36.05	32.37	37.93	36.19	36.89	34.67
4	37.15	41.2	40.8	38.33	33.66	34.67
5	36.45	35.95	35.55	32.03	34.27	34.67
Average	**36.12**	**35.70**	**37.02**	**36.49**	**34.93**	**34.65**
Std. Error	**0.40**	**1.33**	**0.95**	**1.07**	**0.74**	**0.10**

Table 5: Refrigerating Effect and Work Input for Afternoon

Refrigerating Effect (kJ/kg) Afternoon						
S/N	**Q_e 1**	**Q_e 2**	**Q_e 3**	**Q_e 4**	**Q_e 5**	**Q_e 6**
1	150.05	150.05	152.5	149.77	152.5	154.37
2	150.05	150.35	152.8	152.5	152.5	155.07
3	150.75	159.1	147.32	150.47	143.4	154.67
4	154.27	147.32	147.72	153.2	154.67	155.07
5	152.1	152.5	149.77	149.77	149.37	155.07
Average	**151.44**	**151.86**	**150.02**	**151.14**	**150.49**	**154.85**
Std. Error	**0.72**	**1.78**	**1.03**	**0.64**	**1.76**	**0.13**
Work Input (kJ/kg) Afternoon						
S/N	**W 1**	**W 2**	**W 3**	**W 4**	**W 5**	**W 6**
1	36.75	36.75	35.55	36.89	35.55	34.97
2	36.75	36.45	35.25	35.55	35.55	34.27
3	36.05	32.07	37.93	36.19	39.82	34.67
4	34.88	37.93	37.53	34.85	34.67	34.27
5	35.95	35.55	36.89	36.89	37.29	34.27
Average	**36.08**	**35.75**	**36.63**	**36.07**	**36.58**	**34.49**
Std. Error	**0.31**	**0.89**	**0.47**	**0.35**	**0.82**	**0.13**

Table 6: Coefficient of Performance (COP) for Morning and Afternoon

COP for Morning						
S/N	**COP 1**	**COP 2**	**COP 3**	**COP 4**	**COP 5**	**COP 6**
1	4.49	4.49	4.33	4.01	4.79	4.41
2	4.12	4.49	4.29	3.78	4.06	4.52

3	4.18	4.91	3.88	4.16	4.06	4.46
4	4.03	3.41	3.46	3.82	4.65	4.46
5	4.12	4.23	4.29	4.98	4.52	4.46
Average	**4.19**	**4.30**	**4.05**	**4.15**	**4.42**	**4.46**
Std. Error	**0.07**	**0.22**	**0.15**	**0.20**	**0.14**	**0.02**
COP for Afternoon						
S/N	**COP 1**	**COP 2**	**COP 3**	**COP 4**	**COP 5**	**COP 6**
1	4.08	4.08	4.29	4.06	4.29	4.41
2	4.08	4.12	4.33	4.29	4.29	4.52
3	4.18	4.96	3.88	4.16	3.60	4.46
4	4.42	3.88	3.94	4.40	4.46	4.52
5	4.23	4.29	4.06	4.06	4.01	4.52
Average	**4.20**	**4.27**	**4.10**	**4.19**	**4.13**	**4.49**
Std. Error	**0.06**	**0.17**	**0.08**	**0.06**	**0.13**	**0.02**

3.1.2 Efficiency of the Air Conditioning System

ASHRAE/IESNA Standard 90.1-1999 specifies that the minimum efficiency requirements for new installed packaged terminal air conditioners according to Wang (2000) should be range from 8.0 to 9.8 EER as given in equation (20)

The efficiency requirement in Energy Efficiency Ratio (EER) estimated for the air conditioning system is 9.2.

$$EER = 10.0 - \frac{0.16 \times \text{Refrigerating capacity (Btu/hr)}}{1000} = 9.2 \qquad (20$$

3.2 Discussion

The representations of the experimental study results in Tables 4 to 6 were presented in Figures 4 to 7. According to Arifianto et al. (2018) revealed that an increase in the refrigeration effect and coefficient of performance (COP) was due to ejector usage on air-conditioning system which had implications to better cooling capacity and lighter compressor work.

3.2.1 Effect of Evaporator Temperature on COP

The effect of evaporator temperature on COP as shown in Fig. 4 reveals that when evaporator temperature increase the work input will decrease and refrigeration effect will increase which leads to increase in the COP. The coefficient of determination, R^2 shows that there is strong correlation between the COP and evaporator temperature.

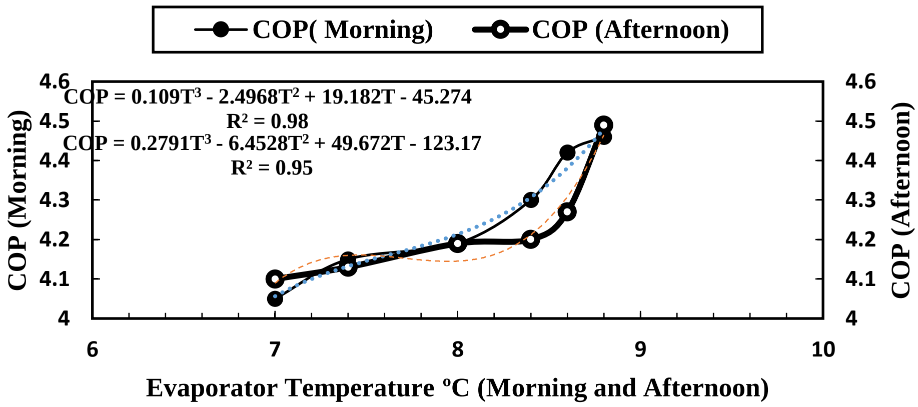

Fig. 4: Variation of COP versus Evaporator Temperature per Week

3.2.2 Effect of Condenser Temperature on COP

The effect of condenser temperature on COP as shown in Fig. 5 indicates that when condenser temperature decreases so the refrigeration effect and compressor input work will increase simultaneously which leads to increase in the COP. The coefficient of determination, R^2 shows that there is strong correlation between the COP and condenser temperature.

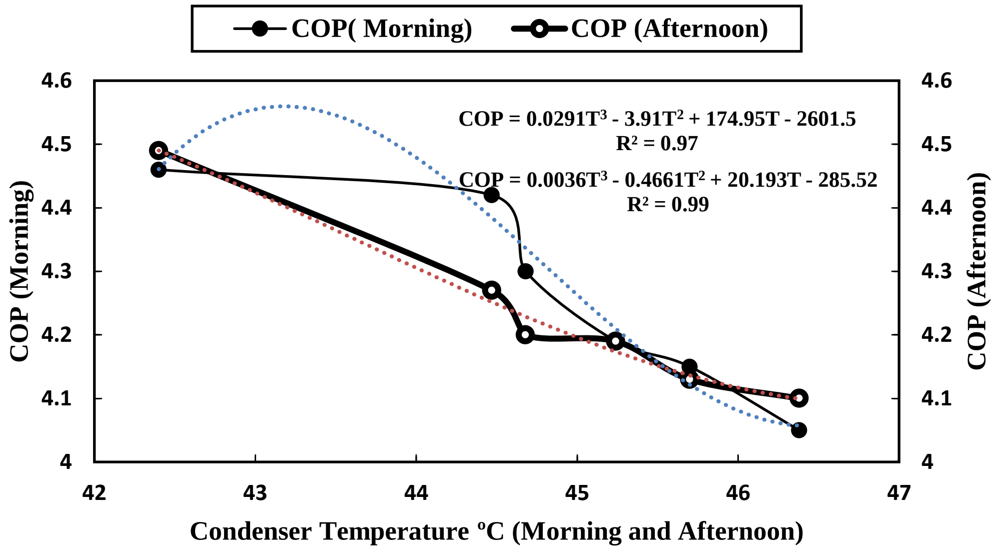

Fig. 5: Variation of COP versus Condenser Temperature per Week

3.2.3 Effect of Refrigerating Effect on COP

The effect of refrigerating effect on COP as shown in Fig. 6 reveals that when refrigerating effect increases, the COP will also increase. It means that the higher the refrigerating effect, the higher the COP. The coefficient of determination, R^2 shows that there is strong correlation between the COP and refrigerating effect.

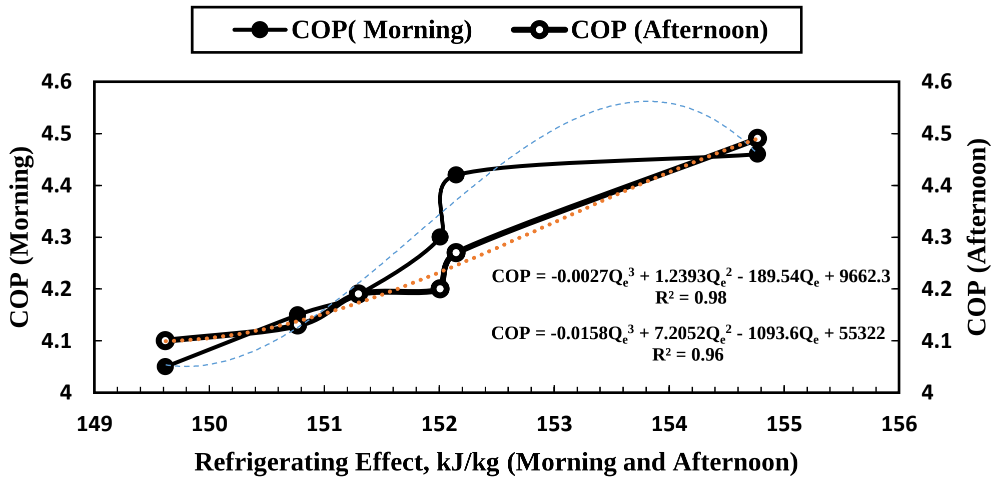

Fig. 6: Variation of COP versus Refrigerating Effect

3.2.4 Effect of Work Input on COP

The effect of work input on COP as shown in Fig. 7 indicates that when work input increases as the refrigerating effect increases simultaneously, the COP increases. The coefficient of determination, R^2 shows that there is strong correlation between the COP and work input.

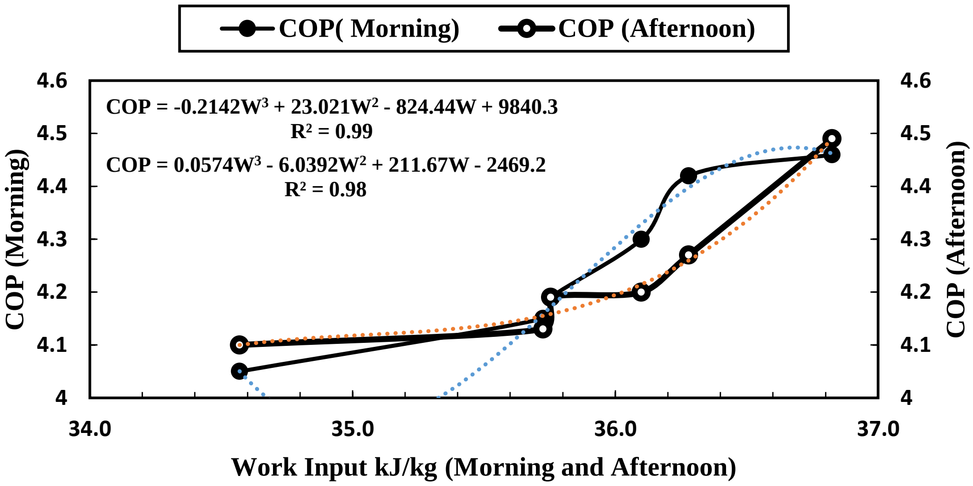

Fig. 7: Variation of COP versus Work Input

4. CONCLUSIONS

The Coefficient of Performance is served as a standard that can be employed to measure the performance and effectiveness of vapour compression refrigeration system which depends on two key parameters such as evaporator and condenser temperatures. This expression reveals that an increase in COP is achieved with increase in evaporator temperature and decrease in condenser temperature. Conclusively, an increase in COP is achieved also with increase in work input in accordance with simultaneous increase in refrigerating effect. The estimated results from the experimentation showed that the operational COP of the air-conditioning system range from 3.41 ± 0.02 to 4.98 ± 0.02. The estimated average value of the COP is 4.5 ± 0.02 for the experimental study and evaluation of air-conditioning system. The efficiency requirement in Energy Efficiency Ratio (EER) for the air-conditioning system used was estimated as 9.2. The coefficient of determination, R^2 value showed that the variable being compared has strong correlation; the closer the R^2 value is to one (1), the better the strength of the correlation between the COP and other variables being considered.

REFERENCES

Jones, W. P. (2001). Air Conditioning Engineering, 5th Edition, Elsevier Science & Technology Books Publisher. New York. ISBN: 0750650745.

Wang, S. K. (2000). Handbook of Air Conditioning and Refrigeration, 2nd Edition, McGraw-Hill, New York San Francisco Washington, D. C. ISBN 0-07-068167-8, pp.1195-1199.

Xiaohua, L., Yi, J. and Tao, Z. (2013). "Temperature and Humidity Independent Control (THIC) of Air-conditioning System", Springer Nature America, Inc.

Richard, C. J. and Gayle, B. P. (2007). Refrigeration and Air-Conditioning, Prentice Hall of India Pvt. Ltd. Cleaner Combustion and Sustainable World, CCSW (2013). Springer Nature America, Inc.

Harris, N. C. (1983). Modern Air Conditioning Practice.3rd Edition, McGraw-Hill, New York.

Fulin, W., Harunori, Y., Bo, L., Noriko, U., Satoshi, H., Takaaki, M. and Hideo, S. (2009). Evaluation and Optimization of Air-Conditioner Energy Saving Control Considering

Indoor Thermal Comfort. Eleventh International IBPSA Conference Glasgow, Scotland, pp. 88-95.

Othman, M. F., Abdullah, H., Sulaiman, N. A. and Hassan, M. Y. (2013). Performance Evaluation of anActual Building Air Conditioning System, 2nd International Conferenceon Mechanical Engineering Research, IOP Publishing, Malaysia. http://dx:doi:10.1088/1757 899X/50/1/012051.

Vahid, V., Bijan, S., Ahmad, F. and Kambiz, P. (2014). "A review of different strategies for HVAC energy saving", Energy Conversion and Management.

Qi, R. Lin, L. and Hongxing, Y. (2012). "Investigation on Air-Conditioning Load Profile and Energy Consumption of Desiccant Cooling System for Commercial Buildings in Hong Kong". Energy and Buildings, 49, 509-18.

Yau, Y. H. (2008). "The Use of a Double Heat Pipe Heat Exchanger System for Reducing Energy Consumption of Treating Ventilation Air in an Operating Theatre-A Full Year Energy Consumption Model Simulation" Energy and Buildings, 40 (5), 917-25.

Enteria, N. and Kunio, M. (2011). "The Role of the Thermally Activated Desiccant Cooling Technologies in the Issue of Energy and Environment" Renewable and Sustainable Energy Reviews, 15 (4), 2095-2122.

Hosoi, A. Sawachi, T. Miura, H. Sakamoto, Y. and Hori, Y. (2006). "Fluctuation Characteristics of Energy Efficiency Ratio of Residential Air Conditioner Depending on Load Ratio and Outdoor Temperature", Proceedings of the Society of Heating, Air-Conditioning and Sanitary Engineers, Nagano, Japan, pp. 1081-1084.

Peng, Q. and Qungui, D. (2015). "Performance Evaluation of a Variable Frequency Heat Pump Air-conditioning System for Electric Bus", International Journal of Fluid Machinery and Systems.

Sopian, K. Abdulateef, J. Alghoul, M. Yigit, K. S. (2013). "Evaluation of a Combined Solar-Assisted Ejector Absorption Chiller", Journal of Mechanical Engineering.

Yumoto, Y., Ichikawa, T., Nobe, T. and Kametani, S. (2006). "Study on Performance Evaluation of a Split Air Conditioning System under the Actual Conditions", Proceedings of the International Refrigeration and Air Conditioning Conference,
Purdue, America, R067, pp. 108-115.

Berman, E. T., Setiawan, A., Arifianto, E. S. and Mutaufiq, U. (2017). Evaluation of Performance an Air-conditioning Systems Using T-Junction Flash Gas Refrigerant. The 2nd Annual Applied Science and Engineering Conference (AASEC 2017) IOP Publishing IOP Conf. Series: Materials Science and Engineering 288: 012064, pp.1-6. doi:10.1088/1757- 899X/288/1/012064.

Moo-Yeon, L., Ho-Seong, L. and Hong-Phil, W. (2012). Characteristic Evaluation on the Cooling Performance of an Electrical Air Conditioning System Using R744 for a Fuel Cell Electric Vehicle, Energies, Switzerland. ISSN 1996-1073. www.mdpi.com/journal/energies, 5, 1371-1383. http://dx:doi:10.3390/en5051371.

Ali, K. S. A. (2018a). Experimental Investigation Effect of Capillary Tube Number on Window Type Air-conditioning Unit Performance. Journal University of Kerbala, Vol. 16 No.1 Scientific.

Ali. K. S. A. (2018b). Performance Enhancement of Window Type Air Conditioning Units. International Journal of Air-Conditioning and Refrigeration, 26 (2) 1850012. World Scientific Publishing Company. http://dx:doi: 10.1142/S2010132518500128.

Berman, E. T. Setiawan, A. Hasan, S. and Mutaufiq, U. (2017). Effect of Installation of a T-Junction on the Performance of an Air Conditioning System. Departemen Pendidikan Teknik Mesin, Universitas Pendidikan, Indonesia. Conference Paper. http://dx:doi: 10.1201/9781315166568-64.

Berman, E. T., Hasan, S. and Mutaufiq, U. (2016). "Enhancing the performance of the domestic refrigerator with hot gas injection to suction line", IOP Conference Series: Materials Science and Engineering, 28: 012028. doi:10.1088/1757-899X/128/1/012028.

Khalifa, A. H. N., Faraj, J. J. and Shaker, A. K. (2017). Performance Study on a Window Type Air-conditioner Condenser Using Alternative Refrigerant R407C. Engineering Journal Volume 21 (1), 235-243, ISSN 0125-8281. http://dx:doi: 10.4186/ej.2017.21.1.235.

Akintunde, M. A. and Erinle, T. J. (2015). Effects of Temperature and Humidity on Coefficient of Performance of Air-conditioning System, AU J.T., Thailand, 18 (4), 145-155.

Rosenquist, G. (1999). Window-Type Room Air-conditioner, ASHRAE Journal, No. 1, USA, 31-36.

Khurmi, R. S. and Gupta, J. K. (2006). Refrigeration and Air-conditioning, 3rd Edition, Eurasia Publishing house (Pvt.) Ltd. Ram Nagar, New Delhi – 110055, India. 526-568.

ASHRAE Handbook (2001). Refrigeration and Air-conditioning, 4 Volumes, McGraw-Hill, New York.

Desai, P. S. (2012). Modern Refrigeration and Air conditioning for Engineer’s, Khanna Publishers, Darya Ganj, New Delhi-110002, India.

McLinden, M. O., Klein, S. A., Lemmon, E. W. and Peskin, A. P. (1998). Tables of the Thermodynamic Properties of Freon 22 (R-22), REFPROP version 6.01, National Institute of Standards and Technology (NIST), USA.

Mistral Associates (2016). P-h Diagram of Refrigeration Cycle with Sub- Cooling. http://www.mistralassociates.com ph.html.

Arifianto, E. S., Berman, E. T. and Mutaufiq, M. (2018). Investigation on the Improvement of Car Air-conditioning system performance using an ejector. MATEC Web of Conferences 197, 08013, AASEC. https://doi.org/10.1051/matecconf/201819708013.