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Energy and exergy analysis of compact automotive air conditioning (AAC) system

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Abstract. Automotive Air conditioning (AAC) is a unit that uses a high measure of energy from a car total engine power. In equatorial climate country such as Malaysia, high usage of AAC is inevitable due to hot, humid and rainy weather throughout the year. An understanding about the energy and exergy losses is essential to find the potential improvement to maximise the efficiency in an AAC system. The main objective of this study is to study the performance of energy and the exergy of a compact automotive air conditioning system. This cycle uses R134a and PAG lubricant as the working fluid. The different ranges of initial refrigerant charge and compressor speed have been tested on the AAC to evaluate the effect of different major thermodynamic parameters in performance. A theoretical model is developed to work out the thermodynamic parameters such as coefficient of performance, exergy destruction ratio, component efficiency defect as well as the dimensionless exergy balance for the AAC system components. The results of this study have shown that most of the energy has been destructed in evaporator part. In order to maximize the efficiency and performance of AAC system, further optimization needs to be done in order to improve the evaporator component.

1. Introduction

Automotive air-conditioning (AAC) system is as important providing thermal comfort to the user in a vehicle passenger compartment. It has a large influence on our overall intellectual activity during travelling. AAC is one of the systems that helps us experience a comfortable ride as it provides the optimum temperature inside our vehicle and improving the indoor air quality inside the vehicle. It is very important to implement the AAC system in a country that experiencing summer and hot conditions, especially in equatorial or tropical countries like Malaysia that experiencing hot and humid climate throughout the year.

The AAC system work using the thermodynamic principle of vapour compression refrigeration cycle [1]. Çengel and Boles [2] found that in a vapour compression refrigeration system, large amounts of heat are being released and absorbed into and from the environment due to thermodynamic processes. The temperature in the designated area can be lower down by using the thermodynamic principle. In the vapour compression refrigeration system, various irreversibility occur which affect the performance of the system. Yumrutaş, Kunduz and Kanoğlu [3] shown that the major source of irreversibility in the vapour compression refrigeration cycle is due to heat transfer between the system and the surrounding environment which take place at a finite temperature difference. The performance of system degrades in the form of losses due to irreversibility. Dincer and Al-Muslim [4] found that the optimization of the cycle can be done by minimizing the irreversibility. Bejan [5] found that the evaluation of the losses in the vapour compression cycle can be done by considering individual thermodynamic processes that make up the cycle.

There is two methods used in the analysis of thermal systems which are energy analysis and exergy analysis. According to Dinçer and Rosen [6], energy analysis is based on the first law of thermodynamics while exergy analysis is based on both the first law and second law of thermodynamics. Exergy analysis is one of the technique for thermodynamic analysis. Bridges, Harshbarger and Bullard [7] found that the limitation of energy analysis is it does not provide any information on the location of inefficiencies in the system. A study from Dinçer [8] show that the exergy analysis can better and more accurately identify the causes and location of thermodynamic losses as well as performance degradation in the system. Saidur, Masjuki and Jamaluddin [9] found that exergy calculations on the process can provide deeper insight in the system as well as new improvements as a complement to the present materials and energy balances. Exergy analysis is a powerful tool in optimization, improving, designing and performance evaluation of energy systems. Exergy analysis is a powerful tool in optimization, improving, designing and performance evaluation of energy systems. Ahamed, Saidur and Masjuki [10] found that exergy is a better analysis as it considers the irreversibility and presents the actual performance of the system. The aim of exergy analysis is to determine the maximum performance of the system and to the sites of exergy destruction. Ozgener and Hepbasli [11] shown that exergy analysis can be performed on each of the components in the system and it can describe all the losses in the system components. The performance of the system can be optimized by using the result from exergy analysis. The direction for potential improvement in vapour compression refrigeration system can be shown by identifying the main sites of exergy destruction were studied by [10].

There are few studies in the current literature deal with energy and exergy analysis of automotive air conditioning system [12-14]. However, there are no experimental data for exergy efficiency are reported for small AAC system mostly used in compact cars. This work is a continuation of the previous study done by Redhwan et al. [15] where the study only stops at energy analysis of the same AAC system. In this paper, an energy and exergy analysis of an AAC system using R134a is carried out experimentally. The analysis takes into account the variation of mass charge of R134a, as well as different compressor speed.

2.0 Methodology

2.1 AAC Experimental Setup.

The AAC experimental setup was modified from the previous research [15, 16]. The original AAC system has been re-used which consist of four primary components: swash plate piston type compressor, a fan-cooled microchannel condenser, a thermostatic expansion valve and a microchannel evaporator. The system components are mounted on a frame by imitating the actual position as in the original car. The evaporator component is inserted into an insulated water tank containing circulating water inlet and outlet. The hot water goes into the tank and transfers its thermal energy to the evaporator. The difference of the water inlet and the outlet is used to calculate refrigeration mass flow rate and cooling capacity in accordance to the ASHRAE standard of Standard 41.9-2000: Calorimeter Test Methods for Mass Flow Measurements of Volatile Refrigerants.

The AAC setup in this experiment uses refrigerant R134a and PAG lubricant as the operating fluid. The measurement of temperature and pressure will be taken using the T-type thermocouple and pressure transducer. The temperature and pressure reading will be taken and stored in the PC using data acquisition hardware. The location of all sensors is shown in Figure 1. The experimental setup will be placed in a special room with constant ambient temperature, which 25 °C with an increment of ± 0.1 °C and constant humidity. The uncertainty for each measurement devices is summarised in table 1.

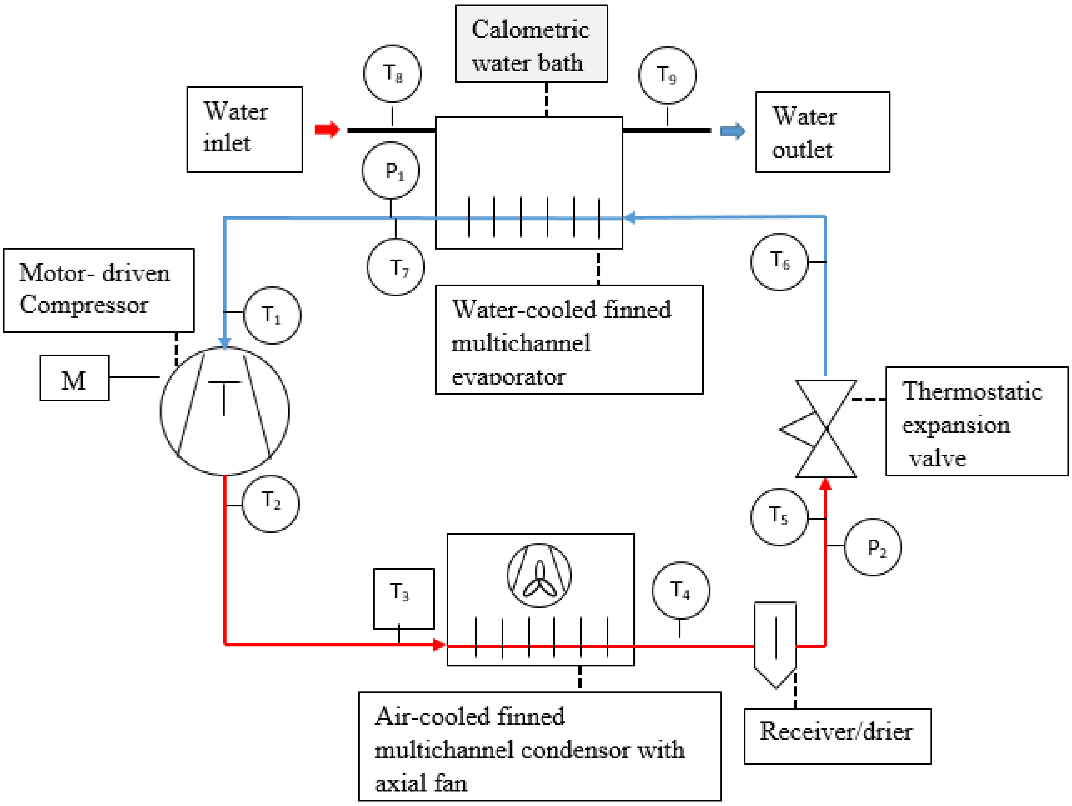

Figure 1. Schematic diagram of Automotive Air Conditioning experimental set-up.

Table 1. The summary for the uncertainties of the experimental parameters.

Parameters	Full scale	Uncertainty
pressure gauge, psi	0- 200	±0.1
K-type thermocouples, K	233.15 to 648.15	±1.5
water flow meter, LPM	0-100	±0.1
Weighing scale, kg	0-25	±0.001
Tachometer, rpm	0-20,000	±2

2.2 Performance analysis of the AAC system

2.2.1 Energy analysis of the AAC system. The model according to the first law of thermodynamics analysis were derived for AAC vapour compression refrigeration system setup. In order to evaluate the performance of energy for this setup, a steady-state process was assumed. The pressure drops in the pipelines and components, loses in the heat transfer, the kinetic, chemical and potential energy of the system were neglected.

Among the important energy parameter for refrigeration performance is the coefficient of performance (COP), which is the ratio of the cooling capacity and the compressor power input which is expressed by:

$$COP = \frac{\dot{m}_r(h_1 - h_5)}{\dot{m}_r(h_2 - h_1)} \tag{1}$$

Hence, the cooling capacity, $\dot{Q}_L$ are expressed as:

$$\dot{Q}_L = \dot{m}_r(h_1 - h_5) \tag{2}$$

The refrigerant mass flow rate, $\dot{m}_r$ are obtained by using the following relation:

$$\dot{m}_r = \frac{\dot{Q}_{water}}{(h_1 - h_5)} \tag{3}$$

$$\text{where } \dot{Q}_{water} = \dot{m}_{water}(h_{water,inlet} - h_{wateroutlet}) \tag{4}$$

And the compressor power, $\dot{W}_{in}$ are expressed as:

$$\dot{W}_{in} = \dot{m}_r(h_2 - h_5) \tag{5}$$

2.2.2 Exergy analysis of the AAC system. The model according to the second law of thermodynamics analysis were derived for AAC vapour compression refrigeration system setup. The limitation of the energy analysis is that it cannot find the nature of irreversibility in the refrigeration system. There are several irreversibility such as heat transfer, finite temperature difference and friction losses in all elements. Exergy analysis is a better tool to understand the magnitude, location and cause of the losses in a thermal system. The destroyed exergy that represents real losses in the quality of energy can be identified in exergy balance. The overall exergy balance to evaluate the irreversibility of system component is expressed as:

$$\dot{E}_{i+i} + \dot{E}_{recovered} = \dot{E}_i + I_i \tag{6}$$

The exergy rate, $\dot{E}$ are expressed by**:**

$$\dot{E}_i = \sum_i \dot{m}\Psi \tag{7}$$

The specific exergy, Ψ is expressed by**:**

$$\psi = (h - h_o) - T_o(s - s_o) \tag{8}$$

The differences of exergy rate are simplify to the following relation [10, 17]**:**

$$\dot{E}_{i+i} - \dot{E}_i = \dot{m}_r(\Psi_{i+i} - \Psi_i) \tag{9}$$

The respective exergy balance for each component of AAC system is written as:

Compressor:

$$\dot{E}_2 - \dot{E}_1 + \dot{E}_{comp} = I_{comp}$$
$$I_{comp} = \dot{m}_r[(h_2 - h_1) - T_o(s_2 - s_1)] + \dot{E}_{comp} \tag{10}$$

$$\dot{E}_{comp} = \frac{\dot{m}_r \left[h_2 - h_1\right]}{\eta_{mech}\eta_{el}} \tag{11}$$

Condenser:

$$\dot{E}_5 - \dot{E}_2 + \dot{E}_{cond} = I_{cond}$$
$$I_{cond} = \dot{m}_r[(h_2 - h_5) - T_o(s_2 - s_5)] + \dot{E}_H \tag{12}$$

$$\dot{E}_H = \dot{Q}_{cond}\left[1 - \frac{T_o}{T_H}\right] \tag{13}$$

Expansion valve:

$$\dot{E}_6 - \dot{E}_5 + \dot{E}_{recovered} = I_{THX}$$
$$I_{THX} = \dot{m}_r[(h_6 - h_5) - T_o(s_6 - s_5)] + \dot{E}_{THX} \tag{14}$$

Evaporator:

$$\dot{E}_1 - \dot{E}_6 + \dot{E}_L = I_{eva}$$
$$I_{eva} = \dot{m}_r[(h_1 - h_5) - T_o(s_1 - s_5)] + \dot{E}_L \tag{15}$$
$$\dot{E}_L = \left[\dot{Q}_L \frac{T_r - T_o}{T_r}]\right] \tag{16}$$

The total exergy destruction at all component were expressed by:

$$I_{Total} = I_{comp} + I_{cond} + I_{THX} + I_{eva} \tag{17}$$

Exergy destruction ratio:

$$EDR = \frac{I_{Total}}{\dot{E}_L} \tag{18}$$

Efficiency defects:

$$\delta_i = \frac{I_i}{\dot{W}_{in}} \tag{19}$$

2.3 Experimental procedures.

Before starting the experiment, all the conditions of the equipment used and conditions of the test setup must be checked and ensured that all safety precautions were followed before the experiment started. The experimental procedure of the AAC performance investigation will be conducted by following the regulations and recommendations from the standard of SAEJ2765 [18]. The AAC experimental test set-up was vacuum first by using a vacuum pump to remove the moisture and to determine if there are leaks in the system. The compressor lubricant is filled into the compressor with 110 mL. The charging machine will be utilized to charge the refrigerant R134a into the AAC system. The desired amount of refrigerant charge is charged into the system. The refrigerant tank will be weighed on a weighing scale to determine the amount of refrigerant charge. The water in the calorimetric water tank was heated up until the temperatures for the inlets and outlets were the same. Then, the experiment started by starting the induction motor with a speed of 900 rpm, which is adjusted by the frequency inverter. The experiment was kept running for 20 minutes and the data reading of temperatures, pressure, power analysers, and water mass flow rates was taken for 10 minutes after that. The data will be recorded and analysed. The variables of the following parameters were used to evaluate the response of the AAC system performance [15]:

1. Compressor speed: 900 rpm to 2100 rpm.
2. Initial refrigerant charge: 95 g to 160 g

3. Results and discussion

3.1. Energy performance of the AAC system.

The performance of the AAC cycle system is normally measured by COP of refrigeration obtained in the experiment. A high COP indicates that the VCRS system is well-functioned and running efficiently with the minimum energy required. Figure 2 shows the graph of COP for various initial refrigerant charges at different compressor speeds. This figure shows that the COP value tends to be higher with an increase in initial refrigerant charges. However, COPs decrease with the increase of compressor speed. The finding is consistent with the findings of past studies [16, 17]. It is noted that this AAC system is designed for a compact car which has a small engine. So, it is normal for the system to have a high value of COP because it needs to use the most minimum power from the engine to cool down the car cabin.

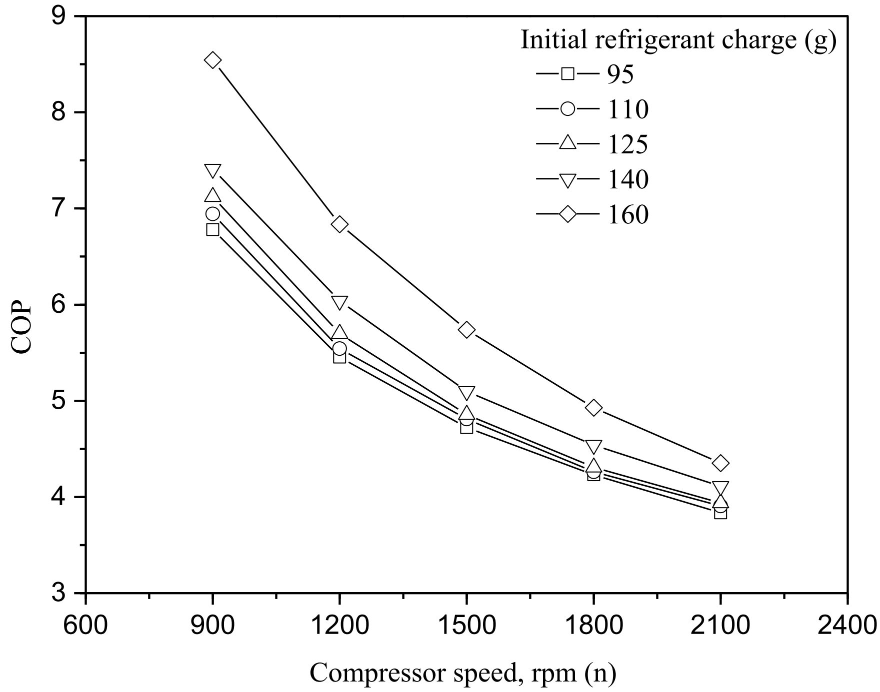

Figure 2. Variation of COP for various initial refrigerant charges at different compressor speeds.

Figure 3 shows the graph of cooling capacity against different compressor speeds for various initial refrigerant charges. It can be seen that the cooling capacity for the higher refrigerant charge is better than the lower refrigerant charge. The addition of the refrigerant will increase the pressure in the AAC system which also increases the enthalpy differences in the evaporator. It may be seen that cooling capacity of the system increase with the compressor speed increment. This is due to the increase of refrigerant mass flow rate thus the heat transfer at evaporator also increase. Figure 4 demonstrates the effect of compressor speed and initial refrigerant charge towards the compressor power. The figure show that the compressor power increases with the increasing amount of initial refrigerant charge and compressor speed. As the refrigerant charge and compressor speed increase, the compressor needs to work harder in order to pump the refrigerant throughout the AAC system.

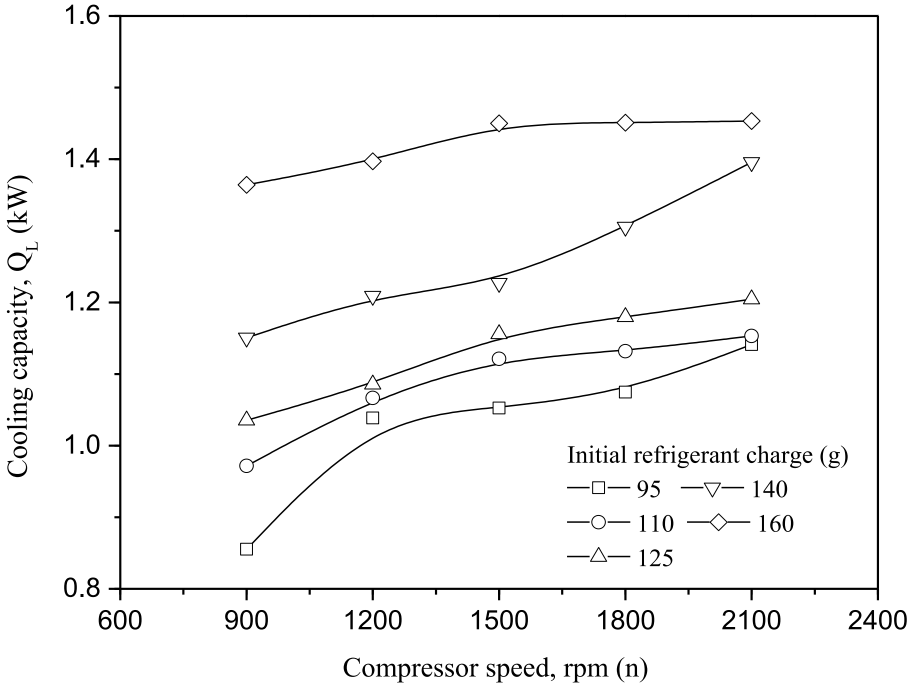

Figure 3. Variation of cooling capacity against different compressor speeds.

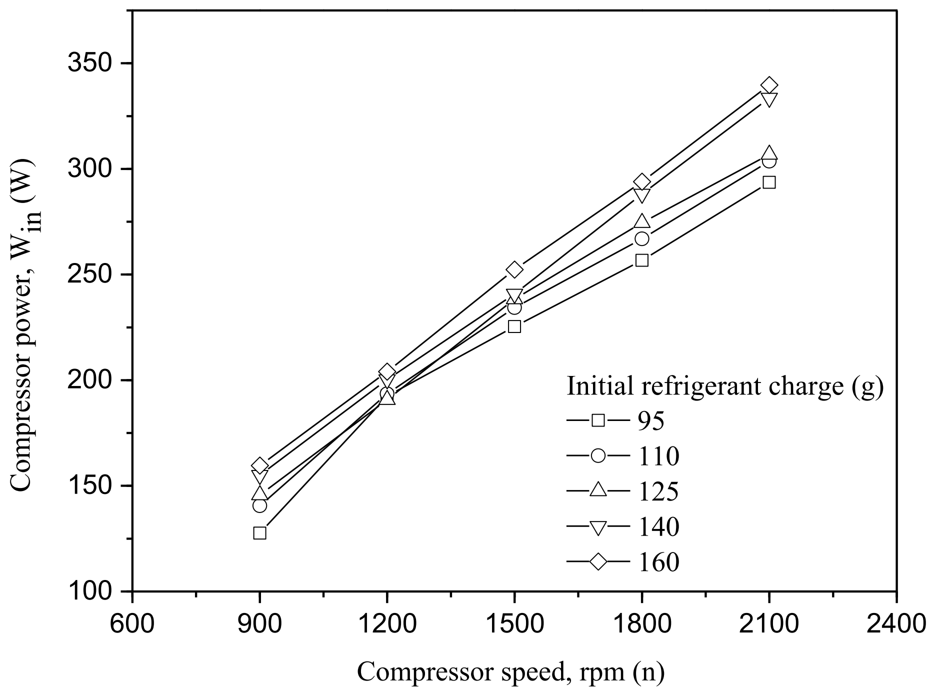

Figure 4. Variation of compressor speed and initial refrigerant charge against the compressor power.

3.2. Exergy performance of the AAC system.

Figure 5 shows the variation compressor speed of exergy destruction ratio at different initial refrigerant charges. According to the theory, the less EDR means that the system has lower irreversibility which is beneficial for the system. The increase of compressor speed has also increased the EDR. This is induced by the increase of the entropy flow throughout all the components in the system. The higher refrigerant charge also tends to have high EDR. These means that at a higher refrigerant charge, there is more irreversibility throughout the system. From the first stage analysis of the AAC system, the higher refrigerant charge gives out better performance in term of COP and cooling capability. Yet, further analysis also shows that the compressor work and EDR are better at a lower refrigerant charge. On the other hand, Figure 6 shows the average efficiency defects for all components in the AAC system. The lower value of efficiency defects means that less exergy was destructed at that particular component. From the figure, it can be concluded that the most exergy destructed at evaporator components. In order to maximize the efficiency and performance of AAC system, further optimization needs to be done in order to improve the evaporator component. In order to obtain a better energy and exergy performance of the AAC system, replacing the working fluid of the system with the nanolubricant is a recommendable option [16, 19-24].

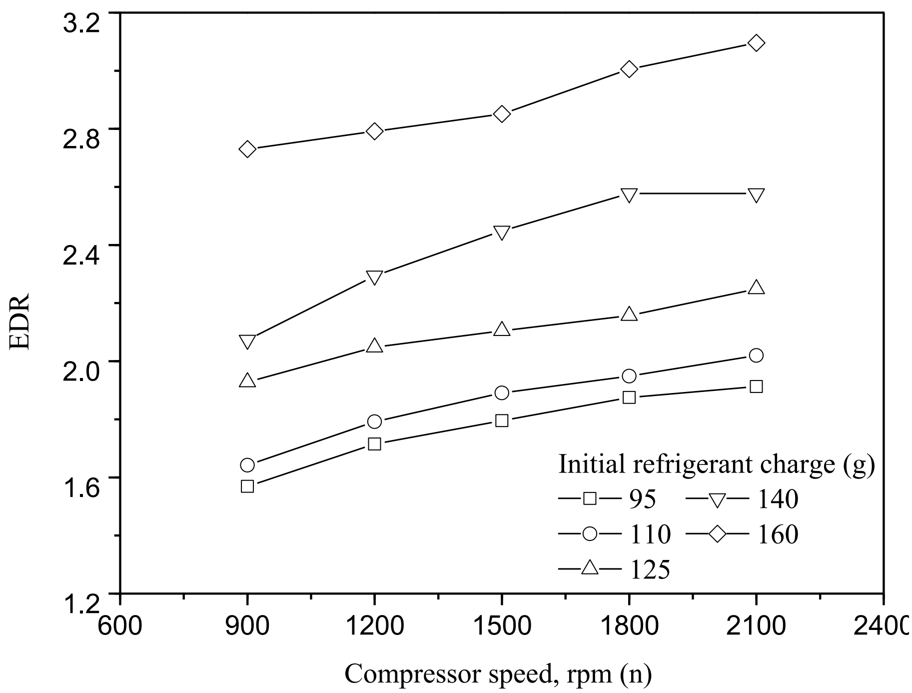

Figure 5. Variation of exergy destruction ratio at different compressor speed.

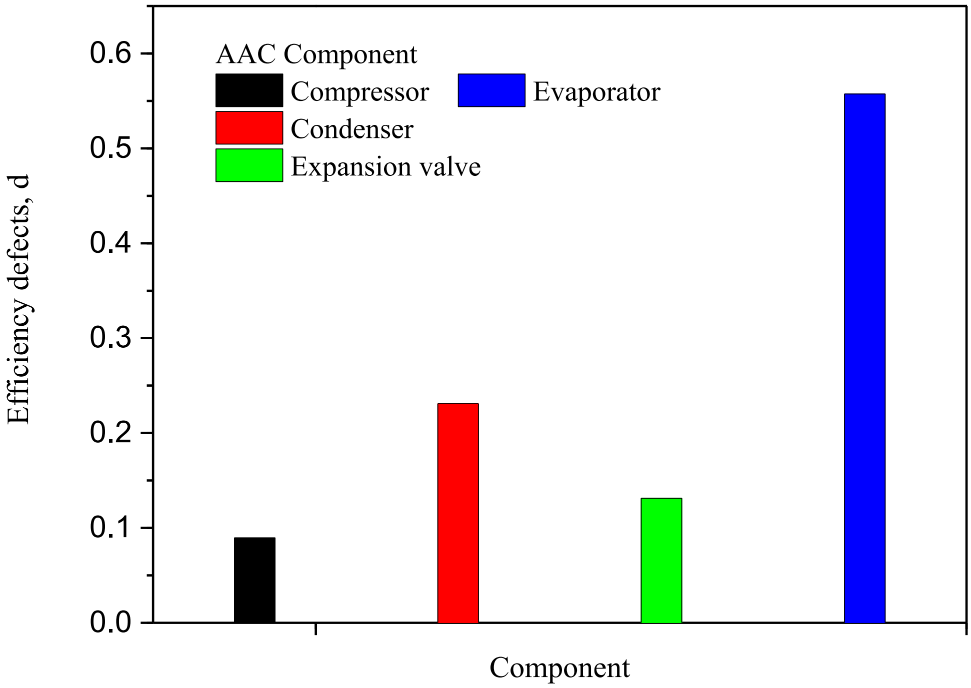

Figure 6. The average efficiency defects for all components in the AAC system.

4. Conclusions

The energy and exergy methods were utilized to analyse the AAC system with R134a refrigerant. The work was carried out by analysing the effect of the initial refrigerant charge and on the coefficient of performance, cooling capacity, compressor power, exergy destruction ratio, and efficiency defect of each component. The various compressor speed and mass of refrigerant charge of R13a show the different effect on the energy the exergy of the AAC system. This result shows that the COP value tends to be higher with an increase in initial refrigerant charges. However, COPs decrease with the increase of compressor speed. The cooling capacity of the AAC system increase with the increment of initial refrigerant charge and compressor speed. Also, the compressor power increases with the increasing amount of initial refrigerant charge and compressor speed. The optimum thermodynamic performances were found at higher refrigerant mass charges. The exergy analysis show that the losses for the system tend to be higher at the higher compressor speed and higher initial refrigerant charge. In addition, the evaporator component in the AAC system has the highest losses compare to compressor, condenser and the expansion valve of the system. Therefore, detailed optimization procedures should be made to minimize losses in all system components especially the evaporator component.

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