Performance of Small Commercial AC System with R-32 Applied for Educational Building in Tropical Country: A Thermodynamic Simulation

I Nyoman Suamir(✉), I Wayan Adi Subagia, I Dewa Made Susila, Ketut Bangse and Adi Winarta

Mechanical Engineering Department, Politeknik Negeri Bali, Badung 80364, Indonesia
nyomansuamir@pnb.ac.id

Abstract. A numerical evaluation on the performance of small commercial AC systems using R-32 as their refrigerant is presented. The evaluation is specifically applied for educational buildings in Bali Province, Indonesia. A digital model was developed. The model was established in Engineering Equation Solver (EES) program called U-CoolS V.1.12, which is a standalone computer program. The model has been validated using data specified by AC manufacturers and then it was used to perform digital evaluation of the AC system based on ambient temperature of the Bali Province, Indonesia. Effects of the condenser and evaporator temperatures on the compressor efficiency and AC system energy consumption were simulated. The results showed a strong effect of condenser and evaporator temperatures on the performance of the compressor and energy use of the AC system. The compressor power increases significantly for about 2.1%, system cooling capacity decreases by 1.8%, COP and EER decreases by 3.9% when the condenser temperature increases by 1 °C. While 1 °C increase of the evaporator temperature can increase power consumption of the compressor of 0.31%, improve the system cooling capacity by 4.4% as well as increase the system COP and EER of about 4.1%. The evaluation also showed that the performance of the split AC system was very good with COP range from 3.27 to 5.95. These finding implicate that R-32 has the potential to be a refrigerant for small commercial split-AC systems.

Keywords: Small Commercial AC System, R-32, Educational Building, Tropical Country, Thermodynamic Simulation

1 Introduction

Air conditioning (AC) system plays a very important role in providing thermal comfort in a conditioned room, specifically for regions with hot and humid climates. The energy used by AC systems in tropical regions could exceed 50% of the total energy use of buildings [1,2]. AC systems are also the utility with the largest energy-significant of commercial buildings particularly hotels. Therefore, there is a great potential to increase overall efficiency of AC systems in hotels for the purpose of conserving energy in buildings. Energy demand for AC system can increase quickly in the 21st century [3].

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Higher ambient temperatures impacting climate change influence cooling energy and more rigorous thermal comfort requirements. Additionally, the climate change impact of AC systems is expected to affect energy demand to rise up to 72%. The largest consumers of energy associated to AC systems originate from developing countries. A substantial upsurge is expected in the South Asian Region as energy demand for AC systems could increase by 50% as a result of climate change [4].

To anticipate these changes, increasing energy use efficiency in commercial buildings is presently the main target of energy policy at national level as well as international level. Among various utilities in buildings, there is a significant increase in energy use in ventilation and AC systems [5]. Several energy conservation solutions for buildings have been promoted including the development of building air conditioning systems equipped with heat recovery systems for the purpose of heating water [6-10]; maintain system performance by closely monitoring the key parameters of the chiller including the approach temperature of the evaporator and condenser [11]; and optimizing the distribution of the cold water circulation system [12]. This energy conservation solution for buildings can encourage wider application of energy efficiency, in support of the Government of Indonesia's commitments and programs to reduce emissions from Greenhouse Gases (GHG) by 26% by 2020 [13]. Therefore, action from the commercial building sector to mitigate and adapt to climate change early on becomes very important to reduce building energy use and greenhouse gas (GHG) emissions [14].

For buildings that use a split AC system, the installation method can also be a factor that can cause the AC system to be energy-intensive and can also cause premature damage to the AC compressor [15]. Premature damage can occur two or three years after installation. Although the service life of a split type air conditioner system under normal operating conditions can be up to 15 years. Practically replacement of split AC systems is recommended after every 10 year period [16].

This paper presents a thermodynamic study of energy through the vapor compression refrigeration cycle. The study was carried out by applying a computer program specifically designed to simulate the performance of a split AC system on various operational parameters. The main aspect that is studied and evaluated in this paper is energy performance including COP, EER, compressor efficiency and power consumption under various operating conditions in a commercial building split AC system using R-32A refrigerant. Refrigerant R-32A is a refrigerant type that is commonly chosen by various manufacturers because it has the advantages of not having the potential to cause ozone depletion (ODP = 0) and relatively low global warming potential compared to R-410A and R-407C [17].

2 Method

Thermodynamic simulation on the performance of refrigeration system for split AC systems using a standalone computer program is applied as a method in this study. The standalone computer program is called U-CoolS V.1.12 program. The program was developed in the Engineering Equation Solver (EES) Professional V11,334 program. A literature study method was also applied to obtain secondary data, especially from manufacturers of various split air conditioners. Secondary data is used to validate the split AC model created in the U-CoolS V.1.12 standalone computer program. The main operating parameters studied in this study include ambient air temperature, condensation temperature in the condenser, evaporation temperature in the evaporator. The effect of changes in these operating parameters on compressor performance, energy performance of air conditioning systems such as COP (Coefficient of Performance) and EER (Energy Efficiency Ratio) is comprehensively evaluated in this paper. To be able to conduct a comprehensive study, the Split AC model is also equipped with a refrigeration cycle based on the operational parameters of the split AC system in the main window diagram. The identification of each observation point of the refrigeration cycle also refers to the schematic of the AC system in the main window diagram.

The compressor performance studied and evaluated includes compressor efficiency, compressor temperature which can be estimated from the temperature of the refrigerant leaving the compressor (T_2), refrigerant mass rate and compressor power. In the thermodynamic simulation in the U-CoolS V.1.12 program, the compressor isentropic and volumetric efficiency is determined using the equation from reference [18] which has been adjusted to the correction factor through program validation based on the manufacturer's specification data for various types and brands of split air conditioners.

$$\eta_s = 1.32\ (0.00476.R_p^2 - 0.09238.R_p + 0.89810) \quad (1)$$

$$\eta_v = 1.18\ (0.00816.R_p^2 - 0.15293.R_p + 1.13413) \quad (2)$$

Where: s = isentropic efficiency; v = volumetric efficiency and R_p = ratio of high pressure and low pressure of the air conditioning system. The isentropic efficiency (η_s) can also be determined by dividing the compressor-specific isentropic compression work (h_{2s}-h_1) by its specific hydraulic compression work (h_2-h_1).

$$\eta_s = \frac{h_{2s}-h_1}{h_2-h_1} \quad (3)$$

Meanwhile, the compressor efficiency (η_{com}) can be determined based on the compressor hydraulic work ratio ($\dot{m}_r$ (h_2-h_1)) and compressor power (W_{com}) as can be seen in equation (4).

$$\eta_{com} = \frac{\dot{m}_r(h_2-h_1)}{W_{com}} \quad (4)$$

Where: r = mass rate of refrigerant (kg/s); h_1 = specific enthalpy of refrigerant at the compressor inlet (kJ/kg); h_2 = specific enthalpy of refrigerant at the outlet of the compressor (kJ/kg); h_{2s} = specific enthalpy of refrigerant at the compressor outlet which has the same entropy as the compressor inlet (kJ/kg); W_{com} = compressor electric power consumption (kW). The split AC system coefficient of performance (COP) is calculated from the cooling capacity of the AC system (Q_{eva}) in kW and compressor power (W_{com}) also in kW as shown in equation (5).

$$COP = \frac{Q_{eva}}{W_{com}} \quad (5)$$

Meanwhile, the energy efficiency ratio (EER) is calculated from the cooling capacity of the AC system (Qeva) and compressor power (Wkom). The Qeva unit is made in Btu/h while the Wkom unit becomes Watt and can be formulated into equation (6).

$$EER = \frac{Q_{eva}}{W_{com}} \quad (6)$$

The thermodynamic study of split AC performance in this study was applied to a 2 Pk split AC system with R-32 refrigerant which was applied to one of the computer-laboratory rooms at the Politeknik Negeri Bali. The evaluation is made based on the ambient air temperature in the Bali Region which is in the temperature range from 24 °C to 35 °C with data sources from the Bali Ngurah Rai Airport meteorological station which can be taken from references [19]. The complete variation of temperature and RH (Relative Humidity) of the ambient air used is data in the month of February as presented in Fig. 1.

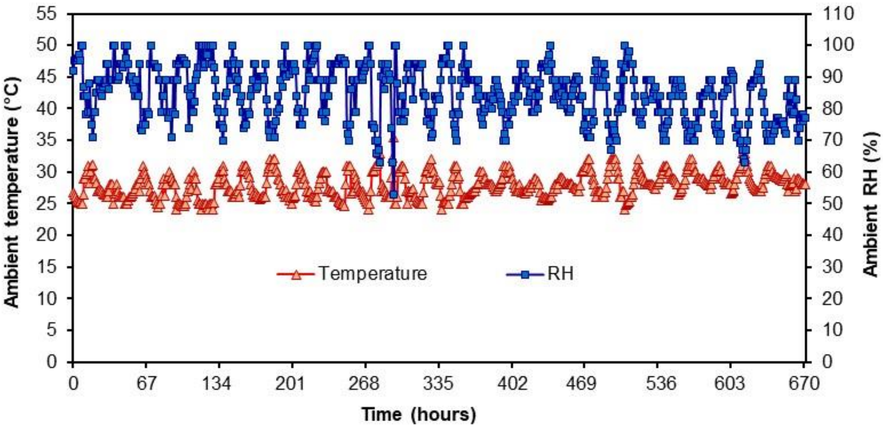

Fig. 1. Variation of ambient air temperature and RH in the Bali area per hour in February [19]

The thermodynamic study of the performance of the split AC system at various environmental temperature variations which directly affect the condenser temperature of the AC system was carried out on the AC system with various evaporation temperature conditions. However, the degree of sub-cooling of the condenser and the degree of superheat of the evaporator were kept constant at 3 °C and 7 °C, respectively. It is also assumed that the standard condenser TD (Temperature Difference) or condenser split is 17 °C and the evaporator has a standard split evaporator specification of 20 °C.

3 Results and Discussion

3.1 Condenser Temperature and Pressure

The results of simulations and studies on the effect of outdoor air temperature on the condensing temperature of the refrigerant in the condenser are presented in Fig. 2.

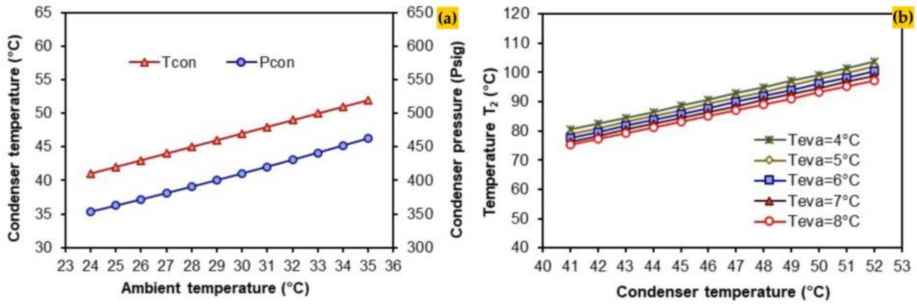

Fig. 2. (a) Effect of ambient air temperature on condenser temperature and pressure (T_{con} and P_{con}); (b) Compressor temperature that can be estimated from the compressor exit refrigerant temperature (T_2)

Evaluated based on the ambient air temperature from 24 °C to 35 °C there was a proportional increase in the condensing temperature of the split AC system from 41 °C to 52 °C. This happens because the condenser split or condenser TD from a constant condenser caused by the absence of changes in the rotation of the condenser fan and the condenser coil and fins are considered clean during the study. The condenser pressure increased significantly when the ambient temperature increased from 24 °C to 35 °C. The increase reached 109 psi, from the initial pressure of 354 Psig to 462 Psig. The full variation between condensing temperature and condenser pressure as a result of changes in ambient air temperature is presented in Fig. 2(a).

3.2 Effects of Condensation and Evaporation Temperatures on Compressor Performance and Cooling Capacity

The condensation temperature which is correlated with condenser pressure and evaporator temperature (T_{eva}) of the refrigerant in the evaporator is closely related to the evaporator pressure, which can have a significant effect on the energy and temperature performance of the compressor and the AC system as a whole. These two parameters can affect the compressor temperature, efficiency, compressor power and refrigerant mass rate that can be circulated by the compressor.

In Fig. 2(b), the temperature variation of the compressor is shown which is indicated from the temperature of the refrigerant leaving the compressor (T_2). It can be seen clearly that the compressor temperature increases as the condensing temperature increases. It was observed at the evaporation temperature (T_{eva} = 5 °C) 5 °C, when the condensing temperature increased from 41 °C to 52 °C, the compressor temperature increased by about 23 °C (from 79 °C to 102 °C). From Fig. 2(b) it can also be illustrated that the evaporation temperature (T_{eva}) can also affect the compressor temperature. Although the effect is not as strong as the effect of the condensation temperature and in fact the effect is the opposite of the effect of the condensation temperature. The compressor temperature increases higher when the AC system operates at a lower evaporation temperature.

The results of the evaluation also show the effect of condensation temperature on compressor efficiency. Compressor efficiency can decrease to 6% when the condensing temperature increases from 41 °C to 52 °C. Compressor efficiency on the other hand increases by 3% when the evaporation temperature increases from 4 °C to 8 °C (Fig. 3(a)). This variation is caused by changes in condensation and evaporation temperatures that can directly affect the compression ratio of the compressor. The compression ratio increases with an increase in the condensing temperature and/or a decrease in the evaporation temperature. Where the compressor efficiency is very dominantly influenced by the compression ratio of the compressor.

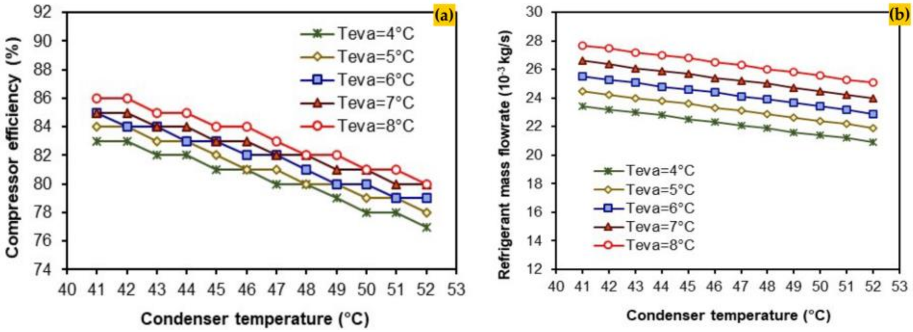

Fig. 3. (a) Compressor efficiency at various condensing and evaporation temperatures; (b) effect of condensation and evaporation temperatures on the refrigerant mass flowrate

The condensing temperature and the evaporation temperature can also affect the mass rate of refrigerant that the compressor is able to circulate as shown in Fig. 3(b). The refrigerant mass rate decreases by about 10.7% at an 11 °C increase in condensing temperature (from 41 °C to 52 °C) or decreased by about 1% per 1 °C increase in condensation temperature. The decrease in refrigerant mass rate due to the increase in condensing temperature is influenced by a decrease in the volumetric efficiency of the compressor when operating at a higher compression ratio. Increasing the condensing temperature at a constant evaporation temperature can increase the compression ratio of the compressor. The effect of the evaporation temperature shows the opposite of the effect of the condensation temperature. The mass of refrigerant that is able to flow by the compressor increases when the AC system operates at a higher evaporation temperature. The increase in refrigerant mass rate can occur about 4.3% per 1 °C increase in evaporation temperature. The increase in refrigerant mass rate is caused by the increase in the density of refrigerant gas at higher temperatures and pressures.

Fig. 4(a) illustrates the effect of condensation and evaporation temperatures on compressor power. It is very clear that the compressor power increases quite significantly by 0.3 kW when the condensing temperature increases from 41 °C to 52 °C. There is an increase in power consumption of about 2.1% per 1 °C increase in condensing temperature. This shows that the compression work of the compressor gets heavier when the condensing temperature of the AC system increases.

While the effect of evaporation temperature variations on compressor power consumption is not so significant. Changes in evaporation temperature in air conditioning systems operating at condensation temperatures below 44 °C do not result in significant changes in compressor power consumption. The effect of evaporation temperature appears to be relatively more significant in AC systems operating at higher condensing temperatures. Power consumption increases slightly by about 0.31% per 1 °C increase in evaporation temperature. The increase in power consumption occurs as a result of the mass rate of the refrigerant flowing which is greater when the evaporation temperature increases. Increasing the mass flow rate can increase the compression work of the compressor as a result of which there is an increase in the power required by the compressor.

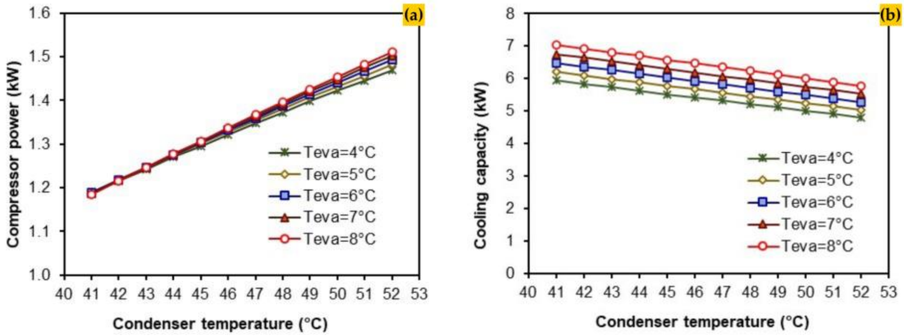

Fig. 4. (a) Effect of condensation and evaporation temperatures on compressor power; (b) Variation of cooling capacity at various condensing and evaporation temperatures

The results of a thermodynamic study on the effects of changes in condensing and evaporation temperatures on cooling capacity as one of the energy performance parameters of the AC system are presented in Fig. 4(b). The cooling capacity of the AC system with R-32 shows a decrease, although not significant if the condensing temperature increase. The magnitude of the decrease in cooling capacity is around 1.8% per 1 °C increase in condensing temperature. On the other hand, it can be seen from the graph in Fig. 4, that there is a significant increase in cooling capacity when the AC system operates at a higher evaporation temperature. For a 2 Pk split AC system with a cooling capacity of 5.37 kW at an evaporation temperature of 4 °C, there can be an increase in cooling capacity to reach 1.04 kW if the evaporation temperature increases from 4 to 8 °C. The increase in cooling capacity is calculated to reach 4.4% per 1 °C increase in evaporation temperature.

3.3 Effects of Condensation and Evaporation Temperatures on Energy Performance

The main performance parameter of the refrigeration system, namely COP or EER, can be ascertained also influenced by changes in condensation and evaporation temperatures as shown in Fig. 5(a) and 5(b). It can be seen from the graph that there is a decrease in COP and EER which is more significant than the decrease in cooling capacity when the AC system operate at elevated condensing temperatures. The amount of decrease in COP or EER per 1 °C increase in condensation temperature can reach 3.9%. This change is caused by in addition to a decrease in cooling capacity, there is also an increase in compressor power when the condensing temperature of the air conditioning system increases.

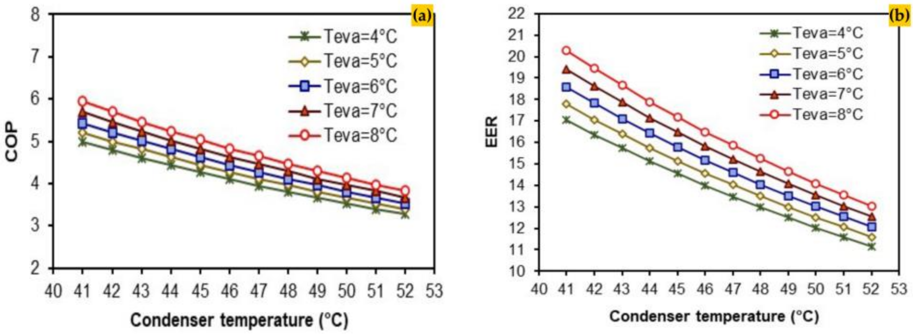

Fig. 5. (a) Variation of COP at various condensing and evaporation temperatures; (b) Variations in EER energy performance of split AC systems due to changes in condensation and evaporation temperatures

On the other hand, COP or EER was found to increase significantly when the evaporation temperature increased. For a 2 Pk AC system operating with an increase in evaporation temperature from 4 °C to 8 °C, there would be an improvement in COP or EER of 16.4% or about 4.1% per 1 °C increase in evaporation temperature. The increase in COP or EER is smaller than the increase in cooling capacity as a result of an increase in compressor power consumption when operating at a higher evaporation temperature.

The results indicate that to ensure the split AC system can operate efficiently with high performance, it is very important to maintain the split AC system so that it can operate at lower condensing temperatures and higher evaporation temperatures. The results also show that the performance of the split AC system with R-32 refrigerant has a good COP range from 3.27 to 5.95. These results can give an indication that R-32 has the potential to be a refrigerant for split AC system applications as an alternative for low GWP refrigerant.

4 Conclusion

An evaluation study on the energy performance of a split AC system using R-32 for application in educational buildings at the Bali State Polytechnic has been carried out by using U-CoolS V.1.12 independent program. The program was developed in the EES V11,334 program. The results of the study show the strong influence of the operational parameters (the condenser and evaporator temperatures) on the compressor performance and the overall energy performance of the split AC system. The results of this thermodynamic study also indicate that the split AC system can operate efficiently with high energy performance, so it is very important to maintain the split AC system so that it can operate at a relatively lower condenser temperature and a relatively higher evaporation temperature.

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