

Exergy and Energy Analyses of Dual-Temperature Evaporator Split AC System Incorporated A Capillary Tube and A Two-Phase Ejector

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ABSTRACT

This study was aimed to investigate performance of a split type air conditioning (SAC) system applying exergy destruction method. A numerical model was established based on exergy destruction analysis of a Condenser Outlet Split-Split Air Conditioning (COS-SAC) system integrated with dual-temperature evaporator and incorporated capillary tube and ejector as expansion devices. An experimental test system was also established to experimentally validate the model. Two type of refrigerants R-290 and R-22 were involved in the evaluations. The Coefficient of Performance (COP) of the ejector COS-SAC system, exergy destruction, and exergy efficiency were determined and compared with those in the SAC system utilizing capillary tube. The results showed there was a significant improvement in the overall exergy efficiency of the COS-SAC systems. The COP of the COS-SAC system was also found to be better than the COP of the SAC system.

Keywords:

Exergy analysis; COS-SAC system; SAC system; energy performance; dual-temperature evaporator

1. Introduction

One popular utility used to condition the indoor environment to meet the level of human comfort is split air conditioning (SAC) system. Indonesia, as a tropical country, has significant segment on the use of SAC system to bring down the temperature and humidity of the hot and humid air to a comfort zone. This makes the demand of SAC systems continue to increase along with the growth of population and the need for comfort. Most of the SAC systems currently use electricity as the first choice of their energy source. This leads to the intensification of electrical energy use.

Besides that, the SAC system commonly uses vapor compression refrigeration system incorporated capillary tube as the expansion device. The use of capillary tube can cause energy losses

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in the system due to isenthalpic expansion process. Therefore, investigation on the energy conservation technology for SAC system need to be carried out.

Implementation of ejectors to produce constant entropy in the expansion process of refrigeration cycle is one of the energy conservation technologies that can be also applied for SAC system. Some studies of the technologies on refrigeration applications have been reported. Starting with a single-phase ejector on the cooling system resulted in a 7% to 9% increase of Coefficient of Performance (COP) for air conditioning (AC) and heat pump uses [1]. Whereas the concept of a two-phase ejector first applied to a cooling system was reported by Gay [2]. Other advantages of two-phase ejectors include: low price, without moving parts, easy construction, and minimum maintenance constraints; these cause the ejectors become attractively prospective in the future [3, 4].

The two-phase ejectors applied for refrigeration system could be classified into two types: constant area ejector and constant pressure ejector [5-8]. Physical differences between constant area and constant pressure ejectors have been explained in Keenan *et al.*, [9]. A constant pressure ejector, investigated at the same operating temperature, was reported to have a lower COP values than a system using a constant area ejector [10]. While Ersoy and Bilir Sag [11] reported the results of their theoretical study under off-design conditions on constant area ejector. They found that the systems with a constant area ejector showed an increase in COP value of 22.3% when compared with the conventional systems. The increase in the COP was depending on the operating conditions. A thermodynamic analysis on refrigeration system incorporated two phase ejector showed that the increase in COP was above 20%, but for experimental study the increase was not more than 10% [12].

The effects of ejector dimensions, geometric, constant area, and diffuser are currently the most popular research topics. The nozzle dimensions, including convergent and divergent angles and other sizes, commonly refers to the standard from ASHRAE [13]. At the early stage of the application of two-phase ejector for AC units was experimentally investigated as discussed in Ünal [6]. Numerical and experimental investigation of a prototype ejector designed to reduce throttling losses has been reported by Elbel and Hrnjak [14]. The liquid mass could cause inefficiency of about 15% due to the two-phase flow was difficult to come out properly at each port. Eventually such inefficiency could reduce the system COP. The invention was identified to have limitations due to the use of fluid accumulators or gas separators. Such inefficiency might occur due to the refrigerant flows into the compressor was in a liquid phase. The liquid phase may damage compressors as well as the AC units. Besides that, the size of the accumulator is commonly quite large. When such system is applied for a split AC unit, the installation of the accumulator can become a problem.

This paper presents numerical and experimental investigation results on energy performance and exergy destruction analyses of a Condenser Outlet Split-Split Air Conditioning (COS-SAC) system incorporated dual temperature evaporator at downstream of expansion devices: capillary tube and ejector. Currently, very few researchers work on the dual-temperature evaporator AC ejector system incorporated capillary tube and ejector. Hassanain *et al.*, [15] investigated design and performance of dual-temperature evaporator ejector system for bus application. No researches are published as yet on the application of dual-temperature evaporator ejector on split type air conditioning (SAC) system. The utilization of dual-temperature evaporator can replace the use of liquid accumulator or gas separator. The arrangement without liquid accumulator or gas separator can prevent the emergence of inefficiencies in two-phase ejector refrigeration systems. The ejector, capillary tube and dual-temperature evaporator are found to be properly and nicely fitted in the indoor unit of SAC system to form a tempting COS-SAC system. The paper also presents experimental investigation through a test equipment of the COS-SAC system established for validating the numerical model. HCFC Refrigerant R-22 and hydrocarbon based refrigerant R-290 are considered in the analyses and evaluations. Refrigerant R-290 is substitute of R-22 for SAC applications due to environmental issues.

R-290, which HCR-22 is one of its trade name, is an environmentally friendly refrigerant and it could perform better than refrigerant R-22 as reported by Aziz *et al.*, [16].

2. Test Equipment and Methodology

2.1 The COS-SAC System and the Two-Phase Ejector

Figure 1(a) and (b) illustrate schematics of the investigated COS-SAC system and P-h diagram of the related vapor refrigeration processes. The COS-SAC system was developed from a conventional SAC system which initially consisted of evaporator, compressor, condenser, and capillary tube as expansion device.

A two-phase flow ejector was added in order to increase the compressor suction pressure and recover the loss of the expansion process in conventional system due to the use of the capillary tube. The original single-temperature evaporator was modified to fit both expansion devices (capillary tube and ejector). The modified evaporator has dual temperature and pressure levels. Thus, the evaporator of the COS-SAC system comprises primary coil at downstream of the ejector where heat is absorbed at higher temperature and pressure (process 6-1) and secondary coil at downstream of the capillary tube where heat is absorbed at lower temperature and pressure (process 7-8).

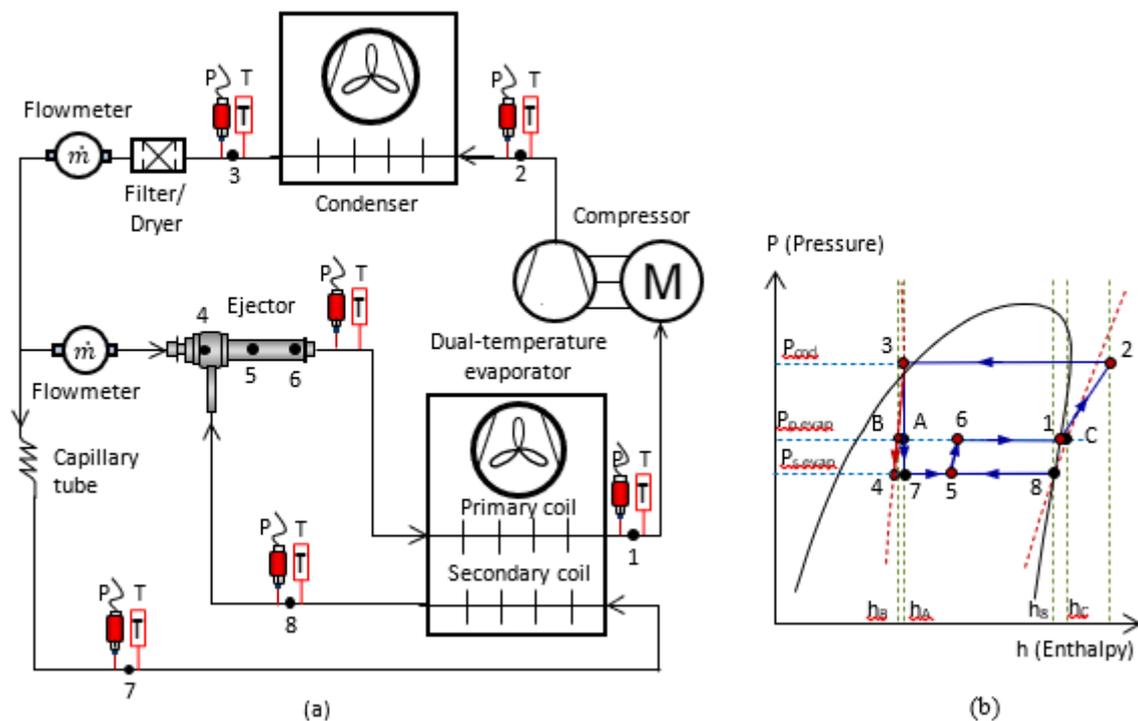


Fig. 1. The investigated COS-SAC system (a) schematic (b) pressure enthalpy diagram

In the investigated COS-SAC system, as per Figure 1, the compressor compresses the refrigerant from the primary evaporator (state 1) so that the pressure increases, which is then followed by the increase of the refrigerant temperature. The superheated refrigerant exits the compressor (state 2) and flows into the condenser where the process of rejecting heat to the ambient air and condensation occurs. The refrigerant, at high-pressure liquid conditions, exits the condenser (state 3) and the flow of refrigerant splits into two liquid streams. The first stream (points 3-4) is primary flow where liquid refrigerant (state 3) flows through the ejector and expands in the diverging-convergent nozzle to be a two-phase flow (state 4). The second stream (point 3-7) is secondary flow

where the liquid refrigerant (state 3) passes through the capillary tube and expands to become a two-phase mixture conditions (state 7). The two-phase refrigerant from the capillary tube enters the secondary coil of the evaporator and absorbs heat from the load; at the same time the evaporation process occurs. The two-phase primary flow (state 4) and secondary flow in slightly superheated vapor (state 8) mix in the constant area of ejector to become a mixture of state 5 which then flows through the ejector diffuser (state 6) to the primary coil of the evaporator. Heat absorption and evaporation also simultaneously occur in the primary coil evaporator which change the phase and state of the refrigerant from vapor-liquid mixture (state 6) to a superheated vapor (state 1). Then the process repeats to form a complete COS-SAC ejector cycle as shown in Figure 1(b).

2.2 Experimental Test Setup

Figure 2 shows the test equipment used for the experiment which comprises a COS-SAC system, instrumentation and data logging system. The equipment was placed in a room where its air temperature was maintained relatively constant at 27 °C. It was also equipped with ducting system incorporated one set electric heater and damper for possibly regulating temperature of the entering air to the condenser. The entering air was kept in the range from 27 °C up to 37 °C which represented variation of monthly design dry bulb air temperature in Indonesia [13]. The damper was used to regulate the cooling air flowrate and to simulate the worst case scenario when installation of the outdoor unit could not allow proper ambient air circulation around the unit. To achieve the test conditions, the damper was set at a position such that air flow rate across the condenser of 0.236 m³/s. It was obtained at average air velocity in the duct of 1.5 m/s. The average air velocity in the duct was determined using Log-T Method and a hot wire anemometer with ±2% reading accuracy was used to measure the air velocity. This arrangement could make the test system simulate condensing temperature in the range between 42 °C and 62 °C.

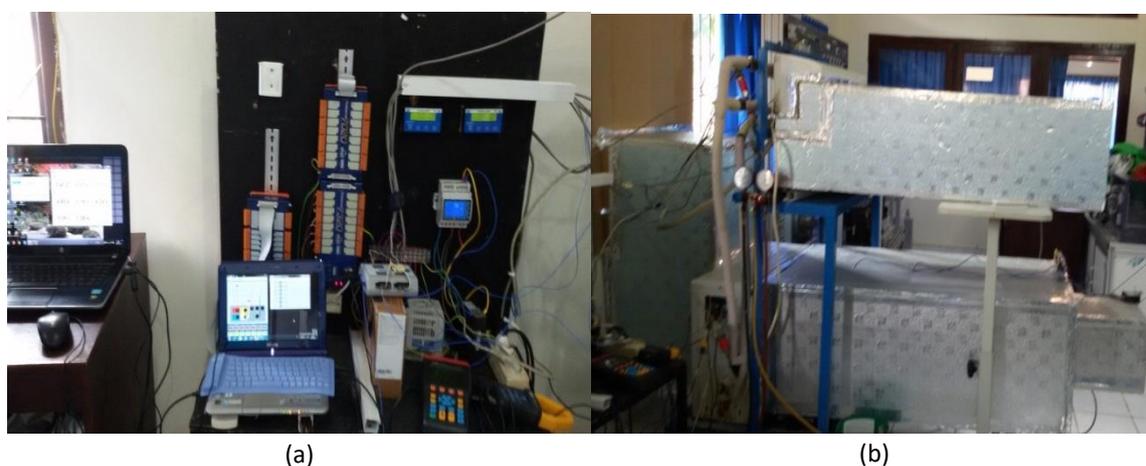


Fig. 2. The experimental test equipment (a) instrumentation and data logging systems (b) COS-SAC system

The instrumentation system used K type thermocouples and pressure transmitters as temperature and pressure sensors of accuracy ± 1.5 °C and $\pm 0.8\%$ respectively. The sensors were connected to a data logging system utilizing Datascan 7320 analog measurement processor and Datascan 7020 analog expansion modules. The data logging system was also equipped with a computer set. The compressor power recorded with a TES 3600 power analyzer of $\pm 1\%$ accuracy. Two oval gear flow meters were used to measure refrigerant mass flowrates which were installed in

the main liquid flow at downstream of the condenser and at the primary refrigerant flow before the ejector as shown in Figure 1(a). The flowrate of the refrigerant through the secondary stream could be calculated from the main and the primary stream flowrates. The flow meters have a measurement range from 0.07 - 5 liter per minute; operating pressure from -0.8 bar to 30 bar; and running temperature ranging from -40 °C to 90 °C. The accuracy of the flowmeters is $\pm 1.0\%$ with 0.5% repeatability of frequency response.

2.3 Methodology

Energy and exergy of the COS-SAC system were numerically and experimentally analyzed and evaluated. The experimental investigation was conducted through direct observation and measurement on the COS-SAC test equipment. Firstly, the experimental test was performed for the COS-SAC system using R-22 refrigerant and secondly the test was carried out for the system with R-290. Energy and temperature performance parameters were observed and recorded. Points of measurements refer to Figure 1(a). The data obtained from the measurement system were processed in a spreadsheet software and simulated in EES (Engineering Equation Solver) program. EES program was also used to process energy and exergy analyses. Energy was analyzed according to the thermodynamics first law (First-LT). Some assumptions were applied which included: negligible pressure loss in the system, isenthalpic expansion process in the capillary tube, mixing process in the constant mixing area of ejector, and steady-state one-dimensional flow.

Isentropic efficiency used to calculate the thermodynamic properties of the compressor output. Empirical expression of the isentropic efficiency of the compressor ($\eta_{\text{cmp, is}}$) is given in Eq. (1) [17,18].

$$\eta_{\text{cmp, is}} = 0.874 - (0.0135) P_i \quad (1)$$

P_i = the compressor pressure ratio determined by using Eq. (2)

$$P_i = \frac{P_2}{P_1} \quad (2)$$

The isentropic efficiency of the ejector nozzle was obtained from the calculations as reported in Hassanain *et al.*, [15]. The calculations were based on the energy equations at the inlet of the nozzle (point 3) and the outflow (point 4) as shown in Figure 1(a). Then, the equation was used to calculate the thermodynamic properties of the refrigerant on the exit side of the nozzle. The efficiency of the isentropic diffuser was calculated based on the energy equation at points (5) and (6) which was then used to determine the thermodynamic properties of the refrigerant at the diffuser outlet. At the diffuser inlet, the properties of the refrigerant were calculated using energy and efficiency of the mixing part equations (point 4). Mixing efficiency indicates the friction loss in the mixing section.

The entrainment ratio (ω) is the ratio between the secondary refrigerant mass flowrate entering the ejector (\dot{m}_8) and mass flowrate of the refrigerant at primary inlet ejector (\dot{m}_4). The mass flowrates (\dot{m}_3 and \dot{m}_8) were obtained from the measurement results (Figure 1(a)). The mass flowrate at primary inlet ejector (\dot{m}_4) was calculated by using equation $\dot{m}_3 = \dot{m}_4 + \dot{m}_8$. At the capillary tube, the refrigerant enters at state (3) conditions and exits at point (7), and because isenthalpic process occurs, the enthalpy at point (7) is the same as the enthalpy at point (3). Then the efficiency of the ejector (η_{ejr}) can be calculated based on Eq. (3) [6, 18].

$$\eta_{ejr} = \omega \frac{h_C - h_B}{h_A - h_B} \quad (3)$$

The specific enthalpy at state point A (h_A) can be determined by using the temperature and pressure of refrigerant at inlet of the nozzle. While the state point B (h_B) can be obtained from the specific entropy of refrigerant at inlet of the nozzle and the refrigerant pressure at diffuser outlet. The state point C (h_C) is determined by using the diffuser outlet pressure and the specific entropy of the refrigerant at secondary flow inlet (state 8).

Exergy, in general, is a quantitative measure of the quality or usefulness of energy, and especially exergy allows comparison based on the same amount (input and output) of various types. Exergy (Ex) is always destroyed in real processes, in part or whole, as explained in Lawrence and Elbel [7]. At each point in the COS-SAC cycle, the specific and physical exergy values are as follows:

$$Ex = (h - h_0) - T_0(s - s_0) \quad (4)$$

$$\dot{Ex} = \dot{m} [(h - h_0) - T_0(s - s_0)] \quad (5)$$

$$Ex_q = (1 - T_0/T)_q \quad (6)$$

For constant temperature process of T , the flow of exergy associated with the heat transfer Q is denoted as Ex_q , and it can be calculated from Eq. (6). Subscripts "0" indicates a reference parameter and represent environmental pressure and temperature. The environmental pressure and temperature were considered 100 kPa and 27 °C respectively. These were taken in accordance with Bilir Sag *et al.*, [19]. The equations used to calculate the destruction of exergy and efficiency of the COS-SAC components referred to Dincer and Rosen [20]. The formula used to compute the total amount of exergy destruction (\dot{Ex}_{tot}) is shown in Eq. (7) comprises: the compressor, condenser, evaporator and expansion device exergy destructions which are expressed respectively as \dot{Ex}_{cmp} , \dot{Ex}_{cnd} , \dot{Ex}_{evap} , and \dot{Ex}_{thv} .

$$\dot{Ex}_{tot} = \dot{Ex}_{cmp} + \dot{Ex}_{cnd} + \dot{Ex}_{evap} + \dot{Ex}_{thv} \quad (7)$$

The overall exergy efficiency (Ψ_{sys}) obtained from the ratio of total exergy damage (\dot{Ex}_{tot}) to total exergy input (\dot{Ex}_i) [20] which is given as Eq. (8). Where \dot{W}_{cmp} is power consumption of the compressor

$$\Psi_{sys} = \dot{Ex}_{tot} / \dot{Ex}_i = 1 - (\dot{Ex}_{tot} / \dot{W}_{cmp}) \quad (8)$$

3. Results and Discussion

3.1 Energy Analysis

Energy performance parameters considered in the investigation involve evaporation temperatures of dual-temperature evaporator, ejector efficiency, condensation temperature, cooling capacity, power consumption and COP. The evaporation and condensation temperatures as well as the power consumption were directly observed from the experimental test results.

3.1.1 Evaporation temperatures

Figure 3 shows temperature variations of the primary coil and secondary coil evaporators which were tested at room air temperature around 27 °C. The COS-SAC system was charged with R-290

refrigerant. The evaporation temperature at primary coil varies in the range between 9.3 °C and 13.0 °C (11.4 °C in average). Refrigerant leaves the primary coil at temperature ranging from 12.4 to 15.2 °C (average 13.8 °C) with degree of superheat at the inlet of the compressor for about 2.4 °C. This indicates that the compressor has worked safely.

The saturation pressure of the primary coil evaporator is in the range between 524 kPa(g) and 593 kPa(g) (76 psig and 86 psig). It is higher than refrigerant saturation pressure of the secondary coil which is ranging from 469 kPa(g) to 517 kPa(g) (68 psig to 75 psig). Evaporation temperature of the secondary coil fluctuates from 6.1 °C to 8.8 °C (at average of 7.8 °C). Degree of superheat of the refrigerant leaving the secondary coil is found to vary from 0 to 0.8 °C. This can make a very effective mixture with two phase primary flow exiting the ejector nozzle. The primary and secondary coil evaporation temperatures of 11.4 °C and 7.8 °C respectively show that overall temperature evaporation of the COS-SAC system is significantly higher than SAC system which commonly ranges from 4 to 6 °C. Higher evaporation temperature can lead to better energy performance.

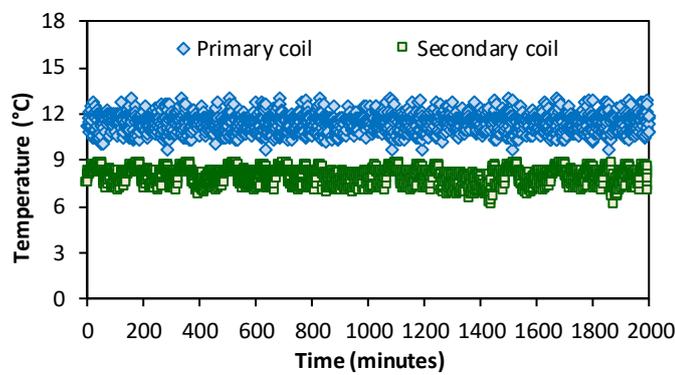


Fig. 3. Saturation temperature of dual-temperature evaporator in the COS-SAC system

3.1.2 Condensation temperature

Figure 4 illustrates the influence of condenser temperature on the entrainment ratio. The condensation temperature was varied from 42.3 to 61.6 °C. The entrainment ratio tends to decrease from 0.08 to 0.06 when the condensation temperature increases up to 61.6 °C. The negative correlation between condensation temperature and the entrainment ratio (ω) also proves the trend. The correlation is given in Eq. (9).

$$\omega = -0.0006 T_{cnd} + 0.0999 \quad (9)$$

Coefficients of determination of the correlation (R^2) is 0.388 which signifies that the correlation is moderate. This implies that the entrainment ratio in the COS-SAC system is acceptably affected by the condensation temperature. Figure 5 shows effects of condenser temperature on the COP and ejector efficiency. The COP of the system decreases from 4.2 to 2.9 when the condensation temperature increases from 42.3 to 61.6 °C but the ejector efficiency slightly increases from 0.60 to 0.64. The influence of condensation temperature can be observed through correlation equations as given in Eq. (10).

$$COP = -0.0517 T_{cnd} + 6.1203 \quad (10)$$

$$\eta_{ejr} = 0.0003 T_{cnd} + 0.6047$$

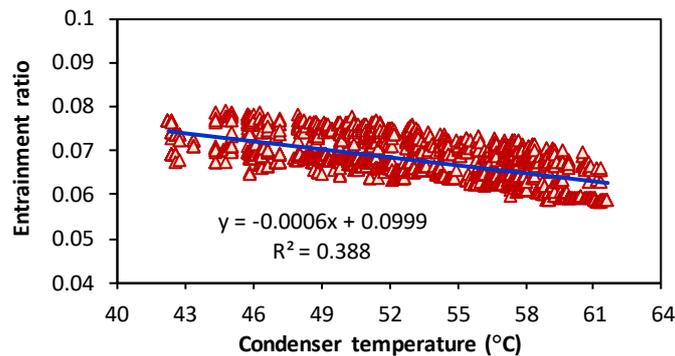


Fig. 4. Effect of condenser temperature on entrainment ratio

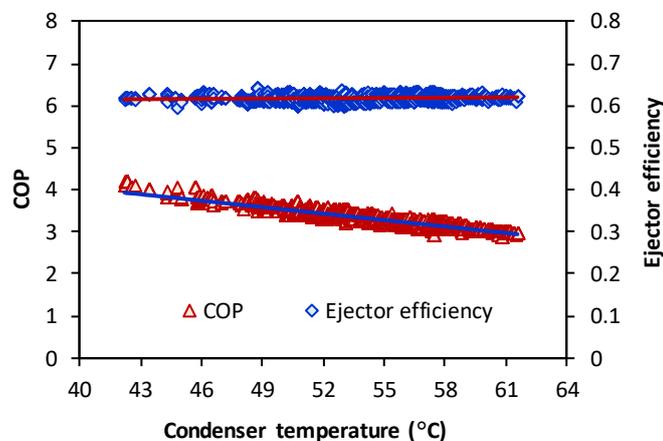


Fig. 5. Effect of condenser temperature on COP and ejector efficiency

The equations confirm a negative relationship between condensation temperature and the COP with strong correlation of determination coefficient (R^2) 0.908. Positive but weak correlation is found on the relationship between condensation temperature and ejector efficiency with coefficient of determination 0.027. The results signify that condensation temperature of the COS-SAC system considerably and negatively affects the COP but have negligible influence on the ejector efficiency.

The efficiency of the ejector can affect the refrigerant flow to the primary evaporator inlet and accordingly can also influence the compressor inlet pressure. Subsequently, the pressure ratio and compression work decrease. This leads to a reduction on input power required to drive the compressor and has a good effect on the COP value of the COS-SAC system.

3.1.3 Cooling capacity and COP

Figure 6 shows input power of the compressor in the COS-SAC system charged with R-290, COS-SAC with R-22 as well as SAC system with R-22. The investigation was conducted at condensing temperature 43 °C and room air temperature to be maintained at 27 °C. Power consumptions of the COS-SAC R-290, COS-SAC R-22 and SAC R-22 are respectively 461.9 W, 539.6 W and 540 W. When tested at condensing temperature of 50 °C, then the power consumption of the three systems are found to increase up to 491.9, 560 and 572 W respectively. Power consumption of the COS-SAC R-

290 system is 14.4% lower than COS-SAC charged with R-22 and of about 14.5% lower than the power of SAC R-22 system. On the other hand, Figure 7 shows that the cooling capacity of the COS-SAC R-290 of about 1847.6 W. It drops for about 10.4% and 2.2% from the cooling capacity of the COS-SAC R-22 and SAC R-22 respectively. Due to power consumption is much lower, however, the COS-SAC R-290 system is found to have a better COP of about 4.7% higher than the COP of COS-SAC R-22 and of about 14.3% better than the SAC R-22 system. The COP of the three systems investigated at 43 °C are found to be 4.0, 3.8, and 3.5 respectively for COS-SAC R-290, COS-SAC R-22 and SAC R-22 systems.

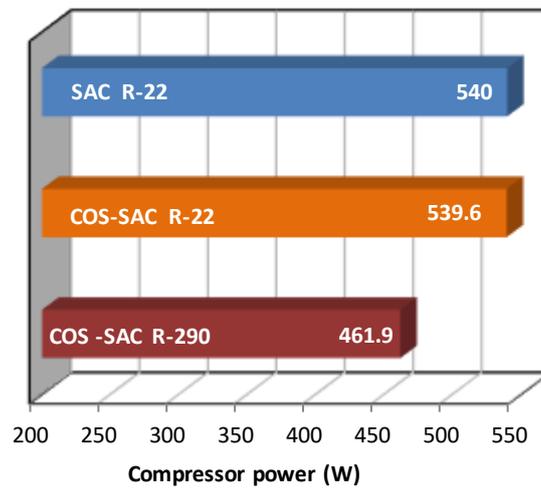


Fig. 6. Compressor power consumption of the COS-SAC R-290, COS-SAC R-22 systems in comparison with SAC R-22 system

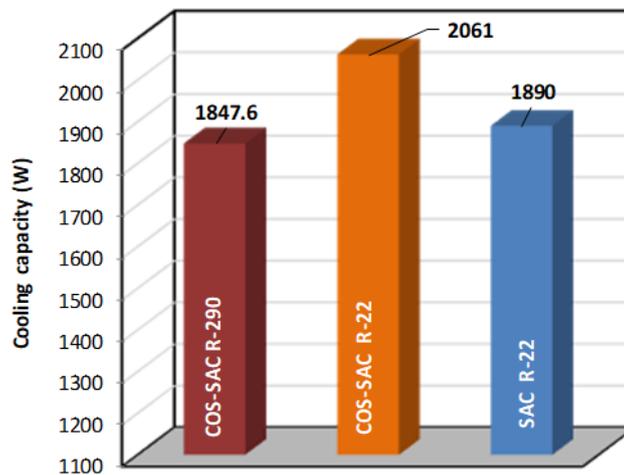


Fig. 7. Cooling capacity of the COS-SAC R-290, COS-SAC R-22 systems and compared with SAC R-22 system

These experimental results show that the COPs of the proposed COS-SAC arrangements (COS-SAC R-290 and COS-SAC R-22) have exceeded COP of SAC system recommended by ASHRAE of 3.5. The results are found in agreement with Zhang *et al.*, [21] that stated the ejector expander refrigeration cycle (EERC) system with R-32 and R134a could perform better than basic SAC system with COP improvement respectively by 5.22-13.77% and 6.63-17.83%. The results are also in line with

Bilir Sag *et al.*, [18] that reported the COP of an ejector expander refrigeration cycle (EERC) R-134a system incorporated accumulator was higher by 7.34-12, 87% than the COP of a basic SAC system and results.

3.2 Exergy Analysis

The number of exergy destruction in each component for COS-SAC has been calculated and the results are presented in Table 1. The values of exergy destruction were simulated at operating conditions: $T_{p, \text{evap}}$ ranging from 9.3-13 °C, $T_{s, \text{evap}}$ in the range between 6.1-8.8 °C and T_{cnd} from 42.3 to 61.6 °C. The exergy evaluations were conducted for COS-SAC charging with R-290 (COS-SAC R-290), COS-SAC with refrigerant R-22 (COS-SAC R-22) and SAC system using refrigerant R-22 (SAC R-22). Simulation results as can be seen in Table 1 show that the highest irreversibility or exergy destruction (ED) value of COS-SAC system occurs in the compressor and the lowest value happens in the capillary tube.

Table 1

Exergy destructions (ED) and exergy efficiency of the investigated cycles

Components	SAC R-22		COS-SAC R-22		COS-SAC R-290	
	ED (W)	Exergy efficiency	ED (W)	Exergy efficiency	ED (W)	Exergy efficiency
Compressor	290.8	0.5145	286.4	0.5335	238.3	0.5828
Condenser	18.59	0.8245	16.02	0.914	16.64	0.9131
Primary evaporator	93.86	0.2352	28.61	0.1586	21.67	0.1747
Secondary evaporator	0	0	57.46	0.1423	47.48	0.1713
Throttle device	39.09	0.8996	0.5315	0.9088	1.05	0.8932
Ejector	0	0	29.45	0.9305	28.59	0.9435

When the ED values of the investigated systems are compared, the exergy destruction that occurs in the capillary tube of the SAC R-22 system is 39.09 W and the total exergy destruction of the ejector and capillary tube are 29.64 W and 29.98 W correspondingly for COS-SAC R-290 and COS-SAC R-22. Therefore, the COS-SAC R-290 and R-22 systems can provide a reduction in exergy destruction by 24.17% and 23.3% respectively.

Table 1 also shows exergy efficiency of the investigated systems. It is found that the COS-SAC R-290 system can provide better exergy efficiency in almost all of the system components. This certainly leads to a better overall exergy efficiency. The overall exergy efficiencies for COS-SAC R-290 and COS-SAC R-22 systems are 23.4% and 22.5% which are better than that of the SAC R-22 system which is only 20.6%. The overall exergy efficiency improvements are accounted for 13.9% and 9.2% respectively for the COS-SAC R-290 and COS-SAC R-22 systems. These results are found slightly higher than those reported in [19, 21].

The improvement on the exergy efficiency of the COS-SAC system can also be observed from Figure 8 which illustrates exergy balance of the three investigated systems using the Grassmann diagram. The COS-SAC R-290 system can provide the highest percentage of exergy output which is accounted for 20.31%. This indicates that the system experiences the smallest total exergy destruction and certainly provide the highest overall exergy efficiency and better COP compared with the other two systems using R-22. This can also provide advantage for the COS-SAC R-290 system because refrigerant R-22 will be phased out in 2030 for Article 5 countries including Indonesia.

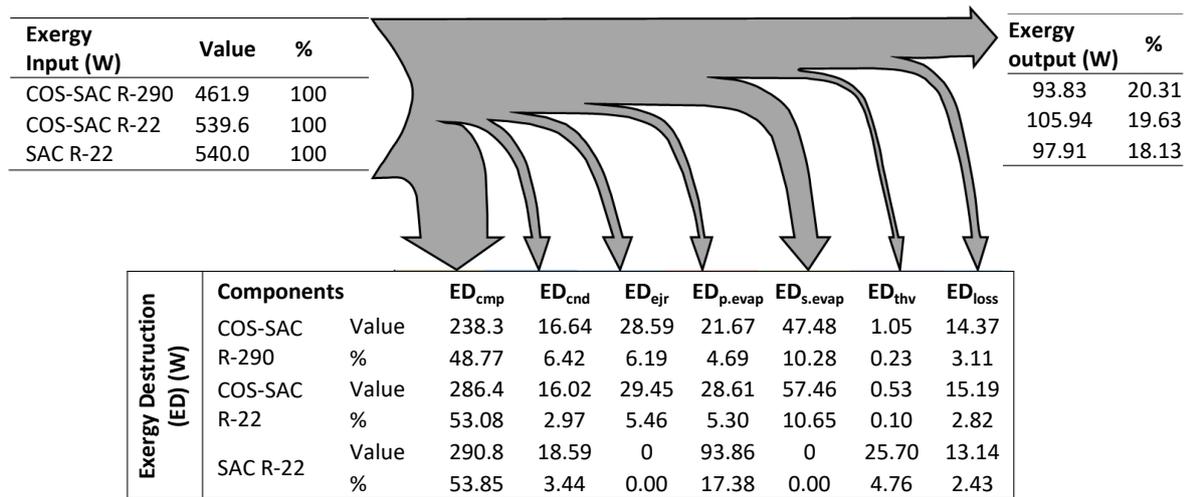


Fig. 8. The exergy balance of the COS-SAC systems in comparison with SAC system

The overall results show that the innovated COS-SAC dual evaporator temperature proposed in this study can eliminate the use of accumulator in the two-phase ejector refrigeration system as well as improve the energy performance of the system. Further investigation will be required on the noise level of the COS-SAC system because noise could reduce efficiency and performance of a split air conditioner system as reported in [22].

4. Conclusions

The Condenser Outlet Split-Split Air Conditioning (COS-SAC) system integrated with dual-temperature evaporator and incorporated capillary tube and ejector as expansion devices has been investigated. A numerical model was established to theoretically evaluate the performance COS-SAC system and an experimental test system was also built to experimentally validate the model. Utilization of dual-temperature evaporator eliminated the use of accumulator as it was in previous ejector technology and make the ejector be properly fitted in the indoor unit of the COS-SAC system. The study also found that the COS-SAC system could work at higher evaporation temperature and pressure than SAC system. Energy performance of the COS-SAC system using R-290 were found to be as high as 4.2 which was 4.7% higher than the COP of COS-SAC charged with R-22 and of about 14.3% better than the SAC R-22 system. The COS-SAC R-290 and COS-SAC R-22 systems could offer a significant overall exergy efficiency improvement from SAC R-22 system of about 13.9% and 9.2%.

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