Paper_1_BC_INSuamir_2021

by I Nyoman Suamir

Submission date: 11-May-2023 06:34AM (UTC+1000)
Submission ID: 2089783549
File name: Paper_1_BC_INSuamir_2021.pdf (814.76K)
Word count: 5630
Character count: 31124

Chapter 13
Print ISBN: 978-93-90888-66-5, eBook ISBN: 978-93-90888-71-9

Thermodynamic Analysis of Two-Phase Ejector as Expansion Device with Dual Evaporator Temperatures on Split Type Air Conditioning Systems: An Advance Study

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DOI: 10.9734/bpi/aaer/v7/7917D

ABSTRACT

This paper presents a numerical and experimental study of increasing the performance coefficient (COP) of split AC (SAC) by reducing compressor work and increasing cooling capacity. Two phase ejector as an expansion device with a new design of dual evaporator temperature used. numerical methods apply the mathematical model developed in the EES software that is applied. Thermodynamic analysis is carried out to achieve ASHRAE Standard requirements for a minimum SAC with COP application of 3.5. The SAC system is filled with R-290 as a thermal fluid medium. Based on the simulation results a numerical model of the ejector is then produced and installed in a modified SAC system of cooling capacity of at least 9000 BTU/hour. An experimental test was conducted to investigate the actual performance of the ejector and its effect on the performance of the SAC system. The results showed that the two phase ejector with the new design of the dual evaporator temperature system was successful. When compared to standard split air conditioners using capillary tube expansion devices the tested ejector system contributes around 35% of power savings. The COP of the system can reach 5.5 which accounts for 39% of the increase in performance. Use of the constant ejector area COS-SAC dual temperature evaporator system which is intended to be applied for split AC to replace accumulator in standard ejector system has been successful.

Keywords: Two-phase ejector; expansion device; air conditioners; coefficient of performance.

1. INTRODUCTION

Air conditioners (AC) are generally used to make air in a room achieve temperature and humidity corresponding to the required level of human comfort. AC systems also provide air circulation throughout the room, air cleanliness and fresh air. The use of AC systems becomes very popular especially in tropical countries such as Indonesia. Currently the use of AC systems is increasingly widespread not only for office buildings and hotels but also for household. Accordingly, electrical energy consumption of AC systems for home and commercial needs has increased significantly. It has been reported that the proportion of electrical energy use by AC systems in hotels could reach 65% of total energy use. This requires attention in the effort to apply energy conservation program in hotel industries especially for electrical energy [1]. Modern air-conditioners incorporate variable-speed compressors and variable-opening expansion valves as the actuators for improving cooling performance and energy efficiency [2-4].

Split air conditioning systems are well known to use vapor compression cycle technology with capillary tube as their expansion devices. Existing concepts about the expansion devices (capillary

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tube, thermostatic/electronic expansion valve (TXV/EXV), orifices) of refrigeration systems are functioned for regulating mass flow of the refrigerant as well as reducing the pressure from the condenser pressure to the evaporator pressure which occurs at constant enthalpy (an isenthalpic process). One thermodynamic disadvantage of the vapor compression cycle is its isenthalpic expansion process that occurs in the expansion device. Isenthalpic process can reduce cooling capacity in the evaporator due to energy losses at the throttling process in the expansion device. Compared with the Carnot cycle which has an isentropic expansion process, the Coefficient of Performance for subsequent abbreviation (COP) of the vapor compression cycle is decreased. This occurs as a result of a decrease in capacity and the loss of recovery of expansion work. Capillary tube are also known to have weaknesses due to friction of the refrigerant flow along the pipe wall as well as changes in velocity along the capillary pipe which cause considerable energy loss.

To overcome such energy losses isentropic process required. Ejectors can be used to generate constant entropy in the throttling process. Some researchers who have tried the appropriate ejector device on the cooling system also stated an increase in COP from the system such as: Single-phase ejectors in modified systems utilized for air conditioning and heat pumping applications resulting in 7% COP improvement up to 9% [5]. The concept of using a two-phase ejector to reduce losses due to the throttling process in the cooling system was first offered [6]. In addition, the two-phase flow ejector has no moving parts, low cost, simple construction and low maintenance requirements that make it a promising system modification [7,8]. Furthermore, the subject of research into applied research that is currently popular among researchers is by replacing the expansion valve of the capillary pipe with ejector device. Kornhauser [9] in different studies using ejector expansion as an expansion device in the cooling system and by using R-12 refrigerant, R-134a concluded the same by comparing them to conventional vapor compression refrigeration systems it was reported that there was an increase in COP values ranging from 3.8% to 21%.

Chunnanond and Aphornratana [10] called a refrigeration system that used two-phase flow ejector to be its expansion tool as Ejector Expansion Refrigeration System (EERS). The use of a two-phase flow ejector instead of the conventional expansion device is one of the efficient techniques to improve the system performance not only by recover the expansion process loss by generating isentropic expansion process but also by increase the system cooling capacity and decrease compressor power [11]. Based on position of motive nozzle, the two-phase flow, ejectors can be categorized into two types namely: constant pressure ejector and constant area ejector [12]. The ejector grouped into a constant-pressure ejector is an ejector with a motive nozzle whose exit plane located in the suction nozzle before to its constant area. Whereas the ejector with outlet field of the motive nozzle exit plane within a constant area ejector was categorized as a constant area projection [13-15]. At the same operating temperature using a constant area ejector COP and its EER is higher than of the system using a constant pressure ejector as reported in [16]. Bilir and Ersoy [17] also reported theoretical studies using R134a. In off-design conditions, the system showed higher COP values than conventional systems and by using COP constant area ejector could be increased by 22.3% depending on operating conditions.

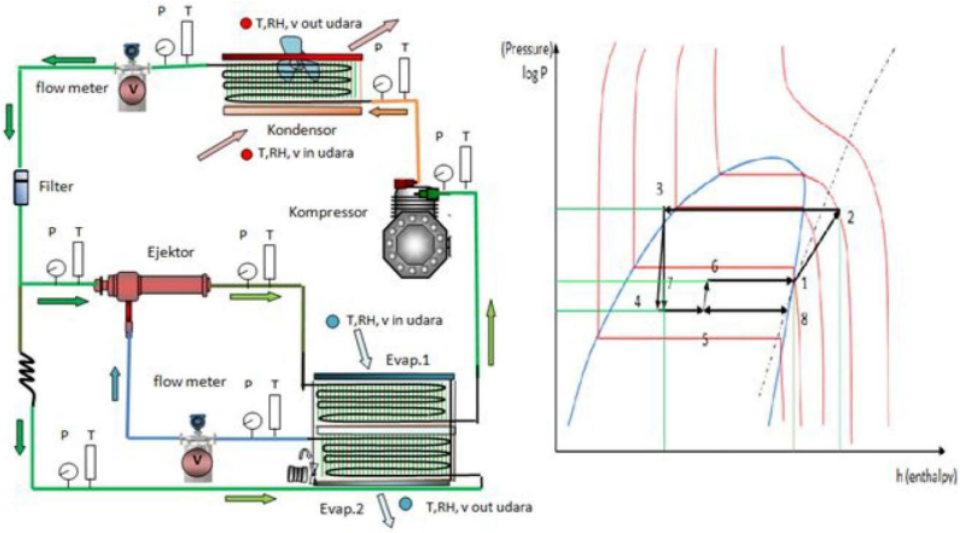
Various studies on numerical analysis and experimental results showed that using a two-phase ejector as an expansion device allowed for the improvement of COP in the steam compression cooling cycle. Thermodynamic analysis showed that COP improvement was above 20%, but no experimental method resulted in an increase of more than 10% [18]. This resulted in improved systems and effects of geometric ejector dimensions, such as the throat of the motive nozzle, the suction chamber, the constant area and the diffuser being an interesting research topic by the majority of researchers. The nozzle was designed according standard based on recommendations from the ASHRAE Handbook including the other dimensions, lengths of each section and the convergent and divergent angles [19]. According to the experimental results as reported in [20], the improvement of COP when using a two-phase ejector as an expansion device on the bus AC system was 8%. Thermodynamic simulation model in [21] stated that if the inefficiency that occurred about 15% of the liquid and vapor mass did not exit properly on the respective ports would decrease the COP of the standard two phase ejector cycle. It was also noted that the ejector cycle with Condenser Outlet Split (COS) ejector cycle could achieve a COP improvement above 10% compared to conventional cycles.

Ejector Cycle adds the throttle variable that is set the degree of exposure by the superheat temperature of the refrigerant at the evaporator output or the refrigerant entering the evaporator. The goal is to improve the efficiency nozzle and improve the range of larger load variations. Nevertheless, the above inventions still have disadvantages and limitations which include the use of fluid accumulator and gas separator where inefficiency occurs especially in the introduction of refrigerant into the compressor so that the liquid refrigerant may enter compressor. This can damage the compressor and the system cannot run properly. A large size accumulator complicates the design if it is used on a split AC system.

This study focuses on liquid and gas refrigerant separator accumulator, using a secondary evaporator to replace accumulator. Separation of the refrigerant flow out from the main liquid refrigerant line of the condenser is directed towards the ejector and the ejector exit is connected into the primary evaporator and then refrigerant gas out evaporator is connected to the compressor. The secondary flow of refrigerant is through a capillary tube and then enters a secondary evaporator. From the secondary evaporator, refrigerant gas flows into ejector as a secondary flow. So that there is refrigerant mixing between primary and secondary flows in ejector and simultaneously increases the refrigerant pressure that enters the compressor. This paper presents constant area ejector applied in a split air conditioner with a secondary evaporator to replace accumulator. Results of experimental tests at Indonesian environmental temperature conditions are also presented. Moreover, the ejector's internal efficiency has numerically been predicted to ensure the system can work with appropriate efficiency.

2. EJECTOR REFRIGERATION SYSTEM FOR AIR CONDITIONER

A schematic illustration of the ejector COS-SAC (Condenser Outlet Split - Split Air Conditioner) system and the corresponding p-h diagram is shown in Fig. 1a and 1b. The main components of this system are almost the same as conventional compression system. The system comprises compressor, condenser, expansion device, and evaporator. The system also incorporates a two-phase flow ejector with an additional evaporator. These two additional components are inserted to recover the expansion process losses and to increase the compressor suction pressure.



a. Schematic diagram **b. Pressure-enthalpy**

Fig. 1. The ejector COS-SAC system

From Fig. 1 can be seen that saturated vapor exits from primary evaporator (point 1) is sucked and compressed to a high pressure and temperature using the compressor. Then, the superheated vapor discharges from the compressor (point 2) is condensed by transfer its heat to the surrounding in the condenser (Q_c). The high pressure saturated liquid exits from condenser (point 3) and then separated into two liquid flows: first flow - primary flow liquid expanded through ejector (point 3-7); second flow - the liquid expanded in the expansion device (point 3-4). The primary flow passed through ejector under high pressure and expands through the converging-diverging nozzle. Mixing of the primary and secondary flows takes place in the constant area ejector (state 5) and a liquid and vapor mixture at an intermediate pressure leaves the diffuser (point 6) of the ejector to the primary evaporator inlet, whereas the vapor circulates back through compressor (point 1–2). The refrigerant after expanded in the expansion device (point 4), goes into the secondary evaporator, and finally enters the ejector as a secondary flow.

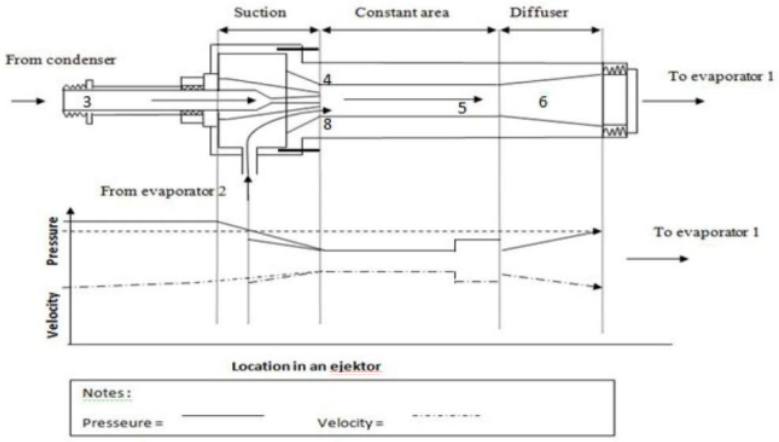


Fig. 2. Constant-area mixing ejector with its pressure and velocity profiles

Operating principle of a constant-area ejector is illustrated in Fig. 2. It consists mainly of two conversion diversion nozzles. Fig. 2 also illustrates pressure and velocity distributions along the axis of the ejector. The high pressure primary flow (state point 3) is expanded in the motive nozzle to a back pressure which is lower than the evaporator pressure (state point 4). Hence, the secondary flow is entrained and expanded through the secondary nozzle to a pressure at state point 8 and then both refrigerant gases are mixed together in the beginning of the constant area section and the mixed pressure (state point 5) is assumed to be higher than the back pressure. A shock wave takes place in the constant-area section and the pressure increases while the fluid velocity decreases. In the diffuser, the pressure increases and the fluid becomes mixture (state point 6) at pressure of the outlet of ejector (ejector back pressure). This pressure is an intermediate pressure between the primary and secondary gas pressures. Technical drawing of the ejector used in this study is developed from [20].

3. METHODS

This study used an experimental method, a direct observation on the AC split test ejector system carried out at ambient temperature which was kept constant at a temperature of 28°C ± 0.5°C. The ejector is designed and self-made (Fig. 2). Thermodynamic analysis and COP of the ejector COS-SAC system are processed and calculated using EES program. The COS-SAC system with two evaporator temperatures can be seen in Fig. 1.

3.1 Thermodynamic Analysis

The thermodynamic analysis is carried out based on mass, energy and momentum conservation equations. The following assumptions are made in the system analysis: Steady state one-dimensional flow; Condensation, evaporation, sub-cooling and superheating temperatures are known; Isentropic efficiencies of the nozzle and diffuser are known; Efficiency of the mixing section is known; The process in the mixing section takes place at constant pressure and constant cross-sectional area; The throttling process in expansion valve is isenthalpic; Pressure losses in the system are neglected. The ejector plays an important role in the COS-SAC as it recovers the expansion process losses and increases the compressor suction pressure. Three ejector parameters such as, entrainment ratio (secondary mass flow to primary mass flow), pressure lift ratio (diffuser exit pressure to secondary nozzle inlet pressure) and geometric area ratio (mixing chamber area to primary nozzle exit area) significantly influence the system performance with an optimum ratio. Based on the above assumptions, in the next section, the conservation equations of mass, energy and momentum are successively applied to each element of the primary, secondary and mixed fluids developed to determine its COS-SAC performance characteristics. The EES program uses to be calculated and simulated and compared with the experiment data was collected by data acquisition. The refrigerant enters the compressor at the point (1) as superheated vapor, and gets out from point (2). Then, the refrigerant enters into the condenser. All the thermodynamic properties at the compressor inlet can be determined if the primary evaporator and superheating temperatures are known. In order to calculate the thermodynamic properties at the compressor exit, compressor isentropic efficiency can be used. [22]. Compressor isentropic efficiency is expressed as follows:

$$\eta_{com.,is} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{1}$$

An empirical expression given in Eq. (2) is obtained for the isentropic efficiency of compressor by using experimental results given in [23,24].

$$\eta_{com,is} = 0.874 - (0.0135)\, P_i \tag{2}$$

P_i = the compressor pressure ratio and h_{2s} is determined by using Eq. (3) and (4):

$$P_i = \frac{P_2}{P_1} \tag{3}$$

$$h_{2s} = F\,(S_{2s}, P_{2s}) \tag{4}$$

Here, s_{2s} is equal to s_1. The enthalpy of the refrigerant at the compressor exit can be calculated from Eq. (1) by using the compressor isentropic efficiency given in Eq. (2). Since the condenser and sub-cooling temperatures are known, thermodynamic properties at the condenser exit are calculated As it is known, an ejector comprises three main sections that are nozzle, mixing, and diffusor. Schematic view of the ejector is given in Fig. 5. Point (3), point (4), point (5) and point (6) indicate nozzle inlet, mixing section inlet, diffuser inlet, and diffuser exit, respectively. Thermodynamic properties at the nozzle exit can be calculated by using the energy equation between points (3) and (4) given in Eq. (5) and nozzle isentropic efficiency given in Eq. (6). Due to conservation of mass principle, it should be considered that $\dot{m}3=\dot{m}4$ and velocity of the refrigerant at the nozzle inlet is neglected in Eq. (5).

$$\dot{m}_3 h_3 = \dot{m}_4 h_4 + \dot{m}_4 \frac{v_4^2}{2} \tag{5}$$

$$\eta_n = \frac{h_3 - h_4}{h_3 - h_{4s}} \tag{6}$$

Thermodynamic properties at the diffuser inlet can be calculated by using energy equation in Eq. (7) and mixing section efficiency in Eq. (8) as follows:

$$(h_4 - \frac{v_4^2}{2}) + \omega\,(h_8 - \frac{v_8^2}{2}) = (1+\omega)\,(h_5 - \frac{v_5^2}{2}) \tag{7}$$

$$\eta_m = \frac{(1+\omega)\frac{v_5^2}{2}}{\frac{v_4^2}{2} + \omega\frac{v_8^2}{2}} \tag{8}$$

Mixing efficiency denotes the frictional losses in the mixing section. ω is the entrainment ratio, which expresses the ratio of mass flow rates of primary and secondary fluids that enter the ejector, has been defined by Eq. (9).

$$\omega = \frac{\dot{m}_8}{\dot{m}_3} \tag{9}$$

For the determination of thermodynamic properties of the refrigerant at the diffuser exit, energy equation and diffuser isentropic efficiency can be used as given in Eq. (10) and Eq. (11), respectively. The minimum refrigerant velocity for the oil return was recommended as 5-7 m s^{-1} in the compressor suction line [25]. There is an evaporator between the compressor and diffuser outlet of the investigated two-phase ejector cooling system in this study. So, velocity of the refrigerant at the diffuser outlet is considered as v_6 = 15 m s^{-1} for the sake of safe oil return.

$$(h_5 - \frac{v_5^2}{2}) = (h_6 + \frac{v_6^2}{2}) \tag{10}$$

$$\eta_d = \frac{h_{6s} - h_5}{h_6 - h_5} \tag{11}$$

The refrigerant enters the expansion valve at the point (3) and gets out from point (7). Since expansion process in the expansion valve is constant enthalpy process, the enthalpy at the point (7) will be equal to the enthalpy at the point (3). Primary mass flow rate $\dot{m}_3 = \dot{m}_4$ can be calculated from the cooling capacity of the system which is given in Eq. (12):

$$Q = \dot{m}_4\,[\omega\,(h_8 - h_7) + (1+\omega)(h_1 - h_6)] \tag{12}$$

The work of compressor can be calculated by the equation given below:

$$W = \dot{m}_4\,(1+\omega)(h_2 - h_1) \tag{13}$$

and the coefficient of performance of the system is :

$$COP = \frac{\omega(h_8 - h_7) + (1+\omega)(h_1 - h_6)}{(1+\omega)(h_2 - h_1)} \tag{14}$$

COP of the two-phase ejector refrigeration system given in Eq. (14) is compared to the COP of the conventional air conditioning system (COP_{std}), and the COP increase rate (COP*) is determined as follows:

$$COP^* = \frac{COP - COP_{std}}{COP_{std}} \quad (15)$$

The coefficient of performance of the conventional refrigeration system can be calculated by the following equation:

$$COP_{std} = \frac{h_8 - h_7}{h_2 - h_8} \quad (16)$$

3.2 Experimental Set up

An experimental ejector COS-SAC test system was established. The test system assembly incorporated a two-phase ejector and dual evaporator-temperature according to the schematic presented in Fig. 1. The tests were carried out in two stages. The first stage, the system was charge with refrigerant R-22 and the second stage, refrigerant R290 was used. Data of test parameters included pressure, temperature were recorded with data logging system consisted DataScan 7200 modules, K type thermocouples and pressure transducers as well as a monitoring display system. Power consumption was recorded with the power analyzer. Two refrigerant micro flow meters at the main flow (at condenser outlet side) and primary flow at the ejector inlet were installed to measure mass flow rate of the refrigerant in the system and refrigerant flow ratio (ω). The test was carried out for 18 hours with 10 repetitions to observe the hysteresis and uncertainty in the measurement.

4. RESULTS AND DISCUSSION

4.1 Evaporation Temperature

Fig. 3a shows variation of evaporator temperature 1 tested at ambient temperature 28°C with two types of refrigerants. The average temperatures of the refrigerant out primary evaporator for R-22 and R-290 are 19.5°C and 18.0°C respectively. Pressures of the primary evaporator side are respectively 110 Psig and 84 Psig for R-22 and R-290. From the data can be estimated saturation temperature of the evaporator are 18°C (for R-22) and 13°C (for R-290) which provide super heat degree of the system 2 K and 5 K respectively for the system running with R-22 and R-290.

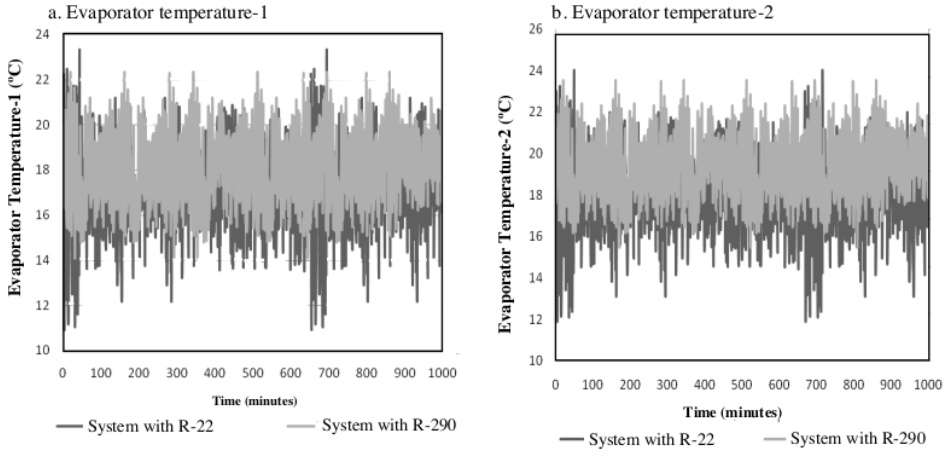


Fig. 3. Dual evaporator temperatures in the ejector COS-SAC system

Variation of evaporator temperature-2 tested at ambient temperature 28°C can be seen in Fig. 3b. Average refrigerant temperature out of secondary evaporator for R-22 and R-290 are 20.3°C and 19.3°C respectively. Pressures of the secondary evaporator are respectively 107 Psig and 80 Psig for R-22 and R-290. While saturation temperature of the secondary evaporator are 17°C for R-22 and 11°C for R-290 which provide degree of super heat 3 K and 7.3 K respectively for the system running with R-22 and R-290. This indicates that refrigerant entering the ejector is in superheat conditions.

4.2 Condensation Temperature

Temperatures of refrigerant entering and leaving the condenser are presented in Fig. 4a and 4b respectively. Average refrigerant temperatures at inlet of the condenser are 53.1°C and 52.7°C respectively for system charged with R-22 and R-290. At the outlet side of the condenser, average refrigerant temperatures for system with R-22 and R-290 were 30.8°C and 29.6°C. Pressures at the high side of the system are 177 Psig for system with R-22 and 150 Psig for system with R-290. Based on high side pressures obtained from measurement, saturation temperatures in condenser can be estimated which are 34°C and 32°C respectively for system when was charged with R-22 and R-290. The tested system was found to work with degree of sub-cooling of about 3.3 K (for system with R-22) and 2.5 K (for system with R-290). This provides indication that refrigerant entering the ejector (the primary flow) and capillary tube (the secondary flow) is indeed in liquid phase. The test results also ensure that the ejector has worked with two phase refrigerant as discussed in published articles especially for split air conditioning applications.

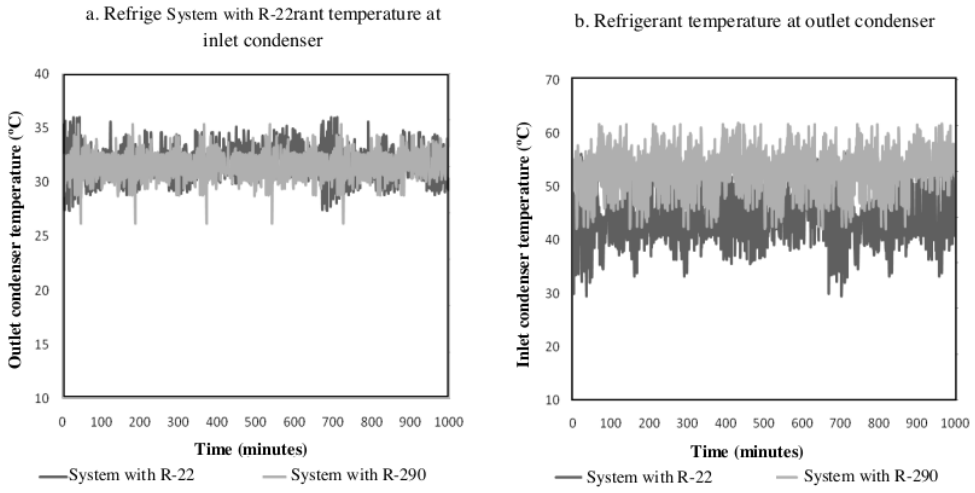


Fig. 4. Condenser inlet and outlet temperatures

4.3 Ejector Internal Efficiency and System Power Consumption

The internal efficiency of the ejector is shown in Fig. 5. The ejector that has a good pressure increase and a good refrigerant velocity d[illegible] at the diffuser outlet can provide a good effect on the primary evaporator. It can also increase refrigerant pressure at the suction side of the compressor. As the result the work of the compressor decreases. The ejector internal efficiency obtained from the test such as nozzle efficiency is 0.30 for the system charged with R-22 and 0.23 for the system charged with R-290. Mixing section efficiency for both systems is 1.36. However, due to small efficiency of the diffuser, the diffuser of the ejector can be considered failed with regards to the effort to increase the pressure and decrease the speed of the refrigerant from the mixing section area. This affects the work of the compressor. It can be seen from the power needed by the compressor for R-290 to be greater than the average power required in the system with R-22 as can be seen in Fig. 6.

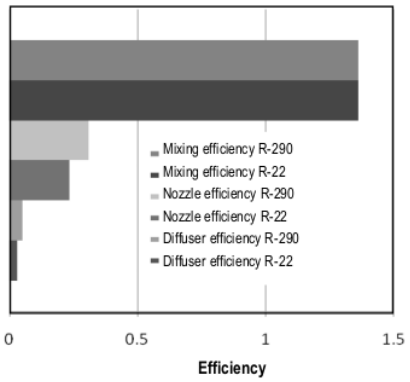


Fig. 5. Ejector efficiency of the COS-SAC system

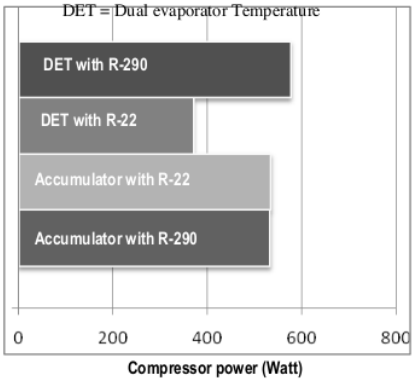


Fig. 6. Power consumption of the COS-SAC system

4.4 Cooling Capacity and COP

The cooling capacity of the ejector COS-SAC system with R-22 refrigerant is 2036 Watts or 8661 BTU hour^{-1} and for R-290 is 3144 Watts or 10737 BTU hour^{-1} (Fig. 7). This shows that the dual evaporator ejector system designed for split air conditioner (SAC) works efficiently. Power consumption of the test system is only 500 watts. When it is compared to standard SAC for the same capacity the average compressor power usage is 770 watts, so there is a 35% savings potential from the power input. In other words, for the arrangement it can be used a compressor with a smaller capacity to get the same capacity.

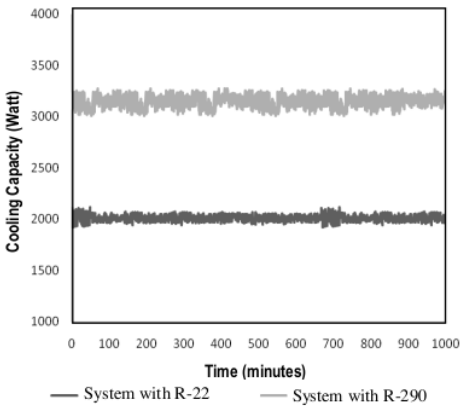


Fig. 7. Cooling capacity of the COS-SAC system

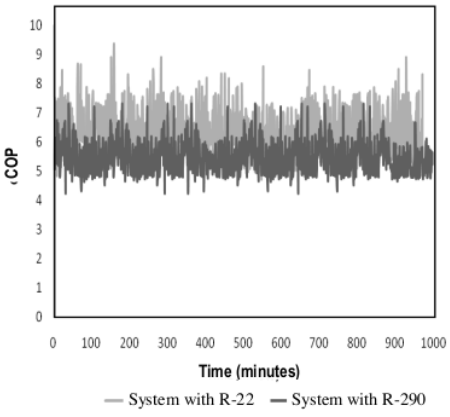


Fig. 8. COP of the COS-SAC system

Fig. 8 shows experimental results on the coefficient of performance (COP) of the test system using R-22 and R-290 refrigerants. It can be seen the test system with R-22 has a mean COP of 5.80, which is slightly better that the COP of test system using R-290 with a mean COP of 5.52. When compared with COP of a standard SAC system with a capillary pipe as the expansion valve, the COP is 3.5. Therefore by using dual evaporator ejector system, the COP improvement was found to be 36% for systems with R-290 refrigerant and 39% for system with R-22. This is in agreement with Lawrence

and Elbel [26] who reported that theoretical COP ejector cycle would not be below when compared with the system using the expansion valve. There are several theoretical advantages of the two phase ejector COS multi evaporator system such as improving the distribution of refrigerants, reducing pressure drop and increasing heat transfer coefficient of refrigerant. COS ejector cycles also have the advantage in terms of oil return.

5. CONCLUSIONS

Based on the results obtained from the study, it can be concluded that the use of the constant ejector area COS-SAC dual temperature evaporator system which is intended to be applied for split AC to replace accumulator in standard ejector system has been successful. The COP system, which uses R-22 refrigerant, has an average value of 5.80 which is 5.1% better than the COP system using the R-290 which is 5.52. If compared to standard split air conditioners using capillary tube expansion device, the tested ejector system contribute about 35% power savings. While the system COP can reach as high as 5.8 and 5.5 accounted for 36% and 39% increase in performance for the system charged with R-22 and R-290 respectively.

ACKNOWLEDGEMENTS

The authors acknowledge the financial support received from the Higher Education Directorate General of the Ministry of Research, Technology and Higher Education of the Republic of Indonesia.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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This chapter is an extended version of the article published by the same author(s) in the following journal.
International Conference on Mechanical Engineering Research and Application, 494: 1-11, 2019.

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