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Thermodynamic analysis with energy recovery comparison of transcritical CO_2 heat pump system using various expansion devices.

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Abstract. The high irreversibility caused by the expansion valve in the conventional transcritical CO_2 heat pump cycle has been reported as the major drawback on the overall system performance. To overcome this problem and recover some of the energy lost, different isentropic expansion devices such as turbine expander and two phase ejector have been proposed. This study aims to numerically compare the performance of the transcrtical CO_2 heat pump in terms of first and second law of thermodynamics. In addition, the energy recovered by the two phase ejector and the turbine expander cycles have been evaluated. The pressure recovery and entrainment ratio in the ejector device were investigated comprehensively. Two numerical models using MATLAB and ASPEN PLUS software have been developed, and REFPROP database was used to estimate the refrigerant thermophysical properties. The results showed that the heating coefficient of performance (COP_h) of the ejector cycle is higher than that of the turbine and valve cycles by 10.15 % and 20.84 % respectively. In addition, the ejector cycle has the highest second law efficiency (0.1) and the recovered energy is (0.63 kW) compared to (0.107 kW) gained by the turbine cycle. The ejector device has the least exergy destruction (0.2 kW) and can recover 0.7 Mpa of the pressure losses.

1. Introduction

The environmental friendly behaviour and the high efficiency performance of the working fluid are essential demands by the heating and cooling industries for any modern heating systems. The transcritical CO_2 heat pump system could provide such characteristics [11, 17]. However, the system performance of the conventional cycle which utilise an expansion valve is compromised by the high irreversibility (throttling losses) in the expansion process. During this process, the CO_2 velocity rises as a result of the gained kinetic energy. This will lead to high friction losses which would reduce the cooling capacity of the evaporator. As a consequence, the system performance will decrease [8, 14]. Moreover, in the transcritical CO_2 heat pump cycle, the heat rejection process take place in the supercritical region which requires higher pressure ratio. This will lead to higher throttling losses compared with the subcritical cycle [5, 17]. Recent studies have claimed that some of the energy lost can be recovered by using isentropic expansion process (constant entropy) instead of isenthalpic expansion process (constant enthalpy) [9, 14]. This can be achieved by replacing the conversional expansion valve by two phase ejector or turbine expander [8, 16]. The two phase ejector isentropically convert the potential energy of the high pressure working fluid into kinetic energy without consuming mechanical work [4, 6]. The

main parts of the ejector are primary (motive) nozzle, suction nozzle, mixing section, constant area section and diffuser nozzle [10], as shown in Figure 1. The high pressure refrigerant flow enters the ejector through the primary nozzle which is a coverage diverge nozzle. As a consequence, the CO_2 isentropically expands leading to a velocity rise from subsonic to supersonic speed. Then in the mixing zone, the high energy two phase refrigerant entrains the low pressure vapour refrigerant from the suction nozzle. Thus, the two flow are mixed and the momentum are exchanged. Finally, at the exit of the diffuser nozzle, further increase in the mixture pressure take place. This pressure value is greater than the suction pressure in the valve cycle [3, 6, 8].
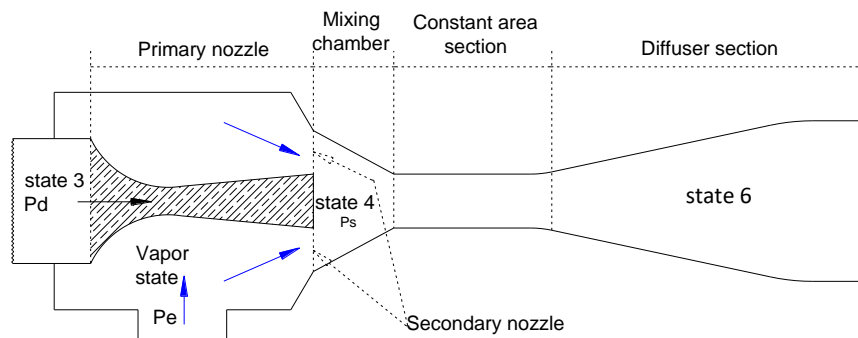
Figure 1. Schematic diagram of the two phase ejector [1].

Theoretical and experimental studies have investigated the performance of the two phase ejector cycle extensively. These studies revealed that using the ejector as an expansion device can reduce the work of the compressor and consequently increase the COP of the ejector cycle as well as reduce the evaporator size [6, 8, 12-15,17]. A theoretical investigation conducted by Li and Groll [9] showed around16% improvement of the COP for the ejector transcrtical CO_2 heat pump system. Whereas, two other studies reported an improvement in the theoretical COP by 22% and 21% respectively [3, 5]. Similarly, an experimental work by Elbel and Hrnjak [6] concluded that the COP for the ejector cycle has been improved by 18%, and 14.5% of the energy lost has been recovered. Whereas, a more recent study pointed out that the COP of the ejector cycle can be improved by up to 30% as well as the exergy lost could be reduce by 25% [7]. Furthermore, the ejector device has more advantages including simple structure, inexpensive and no maintenance required [3, 12].

On the other hand, the two phase turbine expander can recover some of the energy lost throughout the expansion process by converting the high kinetic energy into mechanical work. This recovered energy could be used to drive a compressor or an electric motor [16]. A theoretical and experimental study was conducted by Yang et al. [16] to study the transcritical CO_2 water-to-water heat pump system with turbine expander. The throttling valve was replaced by turbine expander to recover the energy lost in the expansion process. An experimental data was obtained from a test rig by varying the compressor output pressure between (7.5-9.5 MPa). In addition, a steady-state mathematical model was developed and the accuracy of the model was verified by comparing the simulation results with the experimental data. The results showed that the cooling COP of the system could significantly vary with changing the mass flow rate and temperature of the water entering the evaporator. It will also slightly increase the optimal high pressure.

The aim of the current study is to investigate and compare the overall efficiency performance of the transcritical CO_2 heat pump system in term of first and second law of thermodynamics using a valve, two phase ejector and a turbine expander as expansion devices. In addition, the energy recovered by the ejector and turbine cycles is evaluated and analyzed. Moreover, the effect of pressure recovery and entertainment ratio on the ejector design parameters were studies.

2. Modelling of the cycles

The schematic diagram of the two phase ejector heat pump system is shown in Figure 2.
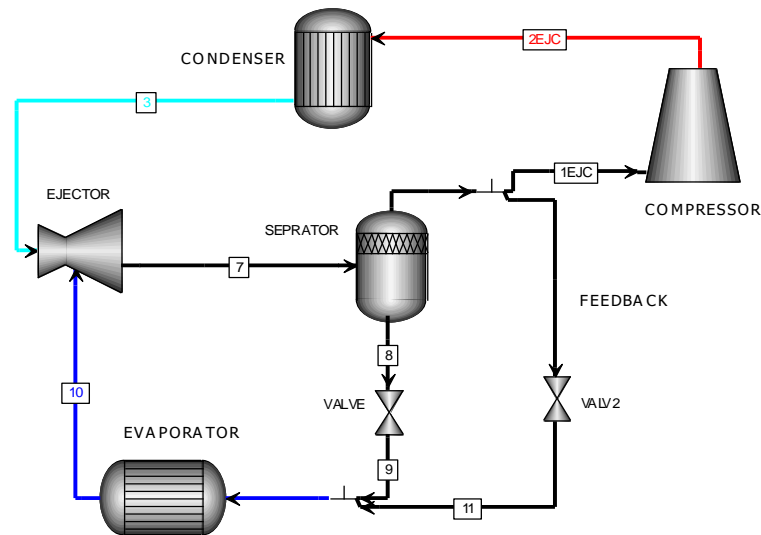
Figure 2. Schematic diagram of heat pump system with two phase ejectors.

The pressure enthalpy (P-H) diagram for the three cycles are represented in Figures 3. For simulation purposes, thermodynamic equilibrium model is assumed across the entire cycles components, i.e. the velocity, pressure and density of the refrigerant phases are equal. The compression process for all cycles is assumed to be an adiabatic process. As shown in Figure 3, for the ejector cycle, the CO_2 enters the compressor at a recovery pressure (P_c) state (1ejc) and exit at a discharged pressure (P_d) state (2ejc).
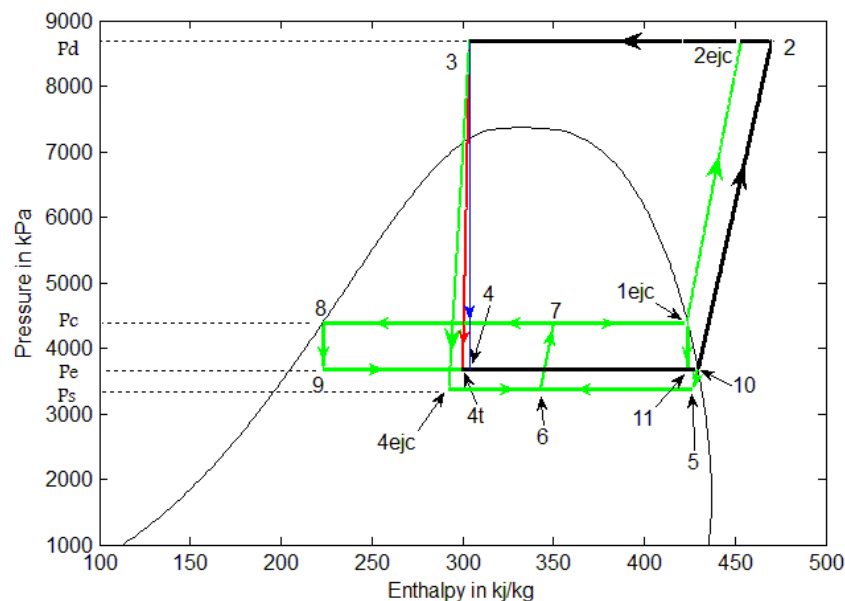
Figure 3. P-H diagram for the three CO_2 heat pump cycles

For the other two cycles, the compression process is slightly different. The refrigerant enters the compressor at an evaporator pressure (P_e) state (10) and compressed to reach a gas cooler pressure (P_d)

state (2). The isentropic efficiency of the compressor η_{comp} is adopted from Ahammed *et al*. [3]. After that, the heat rejection process take place in the gas cooler under the assumption of constant pressure, which is represented by state (2ejc-3) for the ejector cycle and state (2-3) for the valve and turbine cycles. The working fluid is assumed to leave the gas cooler at discharged pressure (Pd) state (3) with a temperature (T3) of 35 °C. Next, the CO_2 is adiabatically expanded in the expansion devices into two phase flows. In the ejector, the refrigerant is assumed to enter the primary nozzle at a stagnation condition, where it expands and exits at state (4ejc). A constant pressure model is adopted from Li and Groll [9]. This model state that the pressure will drop to a value less than the evaporator pressure by 0.3 bar which represent the mixing pressure (Ps) in the mixing section. While for the turbine and the valve cycles, the Carbon dioxide expands to evaporation pressure (Pe) at state (4t) and state (4) respectively. The isentropic efficiency correlations for the turbine (η_t) is adopted from a measured experimental data by Yang *et al*. [16]. While the isentropic efficiency for the ejector nozzles (η_{pn},η_{sn}) are assumed at 85% based on Ahammed *et al.* model [3].

Finally, for all cycles, the stream is assumed to be fully evaporated at the exit of the evaporator with evaporation temperature (Te) of 2 °C (state 10) to re-enter the compressor again. Furthermore, the refrigerant total mass flow rate ($\dot{m}_{tot}$) is assumed to be (0.05 kg/s). In the valve and turbine cycles, this mass will circulate through all parts of the system. Whereas for the ejector cycle, the $\dot{m}_{tot}$ in the mixing chamber is the sum of $\dot{m}_p$ (the mass rate running from the gas cooler through the motive nozzle) and $\dot{m}_s$ (the mass rate passing from the evaporator through the suction nozzle). In the diffuser section (diverge nozzle in Figure 1), the pressure of the refrigerant increases to reach the recovery pressure (Pc). After that, the mixture leaves the ejector at state (7) to enter the separator. In the separator, the wet refrigerant flow is separated into a saturated vapor (state 1ejc) and saturated liquid (state 8). The vapor will be re-compressed again, while the liquid CO_2 will expand in the valve of the ejector cycle at evaporator pressure (Pe) (state 9).

For the ejector cycle simulation, a feedback throttle valve adopted from Li and Groll model [9] was assumed. The purpose of this valve is to maintain the mass balance in the system by shifting excess vapor exiting the ejector back to the evaporator. This will keep the vapor fraction (CO_2 quality) equal to the mass ratio of the primary over the total refrigerant flow.

In order to evaluate the overall performance of the three cycles in terms of first and second law of thermodynamics, a steady state one dimensional mathematical model written by MATLAB software is developed. This code is linked to REFPROP database in order to obtain the thermophysical properties of the refrigerant in different states across these cycles including pressure, temperature, enthalpy, entropy and quality. In addition, the P-H diagram is obtained from this code. In this simulation, a scaler vector of discharged pressure ranged between (8 to 12 Mpa) is set in order to identify the optimum gas cooler discharged pressure that could produce the maximum value of COP_h.

The coefficient of performance (COP_h) for the three cycles is calculated from the following equations:

$$COP_{h\_Valve} = \frac{\dot{m}_{tot} \times (h_2 - h_3)}{\dot{m}_{tot} \times (h_{10} - h_2)} \tag{1}$$

$$COP_{h\_Turbine} = \frac{\dot{m}_{tot} \times (h_2 - h_3)}{\dot{m}_{tot} \times (h_{10} - h_2) - \dot{m}_{tot} \times (h_3 - h_{4t})} \tag{2}$$

$$COP_{h\_Ejector} = \frac{\dot{m}_p \times (h_{2ejc} - h_3)}{\dot{m}_p \times (h_{2ejc} - h_{1ejc})} \tag{3}$$

The exergy destruction for each thermodynamic process is carried out under the assumption that the environment temperature (Tw) is 27 °C and the reference temperature (To) is 35 °C.

$$I_{expan.\_Valve} = \dot{m}_{total} \times T_o \times (S_4 - S_3) \tag{4}$$

$$I_{expan.\_Turbine} = \dot{m}_{total} \times T_o \times (S_{4t} - S_3) \tag{5}$$

$$I_{expan.\_ejc} = \sum I_{pn} + I_{sn} + I_{dff.} + I_{mx} \tag{6}$$

Second law efficiency is defined for each cycle as follow:

$$\gamma_{exergy\_Valve} = 1 - \frac{\sum I_{\_Valve\_cycle}}{W_v} \tag{7}$$

$$\gamma_{exergy\_Turbine} = 1 - \frac{\sum I_{\_Turbine\,cycle}}{W_v + W_{t\_out}} \tag{8}$$

$$\gamma_{\text{exergy\_Ejector}} = 1 - \frac{\sum I_{\_Ejector\_cycle}}{W_{ejc}} \tag{9}$$

The energy recovered by the ejector and turbine cycles:

$$Energy\ recovered_{ejector\_cycle} = W_v - W_{ejc} \tag{10}$$

$$Energy\ recovered_{turbine\_cycle} = W_v - W_{t\_out} \tag{11}$$

The pressure recovery in the ejector cycle is determined by:

$$Pressure\ recovery_{ejectro\ cycle} = P_c - P_e \tag{12}$$

3. Results and discussion

The comparison between the steady state results for the three heat pump cycles conducted using MATLAB code and ASPEN software are shown in table 1. These values were obtained at a calculated optimum discharge pressure of around 8.7 Mpa.

Table 1. Steady state results of the three heat pump cycles employing three different expansion devices.

	Ejector cycle		Turbine cycle		Expansion valve	
	MATLAB	ASPEN PULSE	MATLAB	ASPEN PULSE	MATLAB	ASPEN PULSE
COP_h	4.9398	4.912	4.4848	4.425	4.0879	4
Exergy efficiency	0.1065	0.1	0.0929	0.091	0.0823	0.0815
Exergy destruction in the expansion device (watt)	0.2087	0.2	0.4871	0.41	0.6959	0.67
Cooling capacity (kW)	2.2280	2.2	3.6120	3.6	3.7809	3.6
Heating capacity (kW)	2.7648	2.6	4.9060	4.9	5.0054	5

Table1 shows that the ejector cycle has achieved the highest values in terms of first and second law of thermodynamics. For instant, from MATLAB results, the COP_h of the ejector cycle is higher by 10.15 % and 20.84 % than the turbine and expansion valve respectively. While, the second law efficiency of the ejector cycle is higher by 14.64 % and 29.40 % compare to the turbine and valve cycles respectively. The irreversibility in the ejector device is lower by 57.15 % and 70.01% than turbine and valve cycles respectively. In contrast, the turbine and valve cycles have achieved nearly similar heating and cooling capacities for this case study, which were higher than that of the ejector cycle.

Furthermore, the results plotted in Table 1 shows a good agreement between the results obtained from MATLAB and ASPEN PLUS models. Thus, MATLAB code can be used confidently to conduct further evaluations in this study.

3.1 Comparison of the energy recovered by the ejector and turbine cycles

Figures 4 and 5 show the energy recovered by the ejector and turbine cycles, respectively. For the ejector cycle (Figure 4), the energy recovered rises sharply with increasing the discharge pressure, reaching 0.63 kW at the optimum gas cooler pressure. In contrast, the energy recovered by the turbine cycle significantly decline with initial change in discharge pressure, achieving a value of 0.107 kW at the optimum pressure then gradually increase thereafter. Generally, the energy recovered by the ejector cycle is significantly higher than that for the turbine cycle. This is due to less compression work in the ejector cycle compared to the turbine cycle.
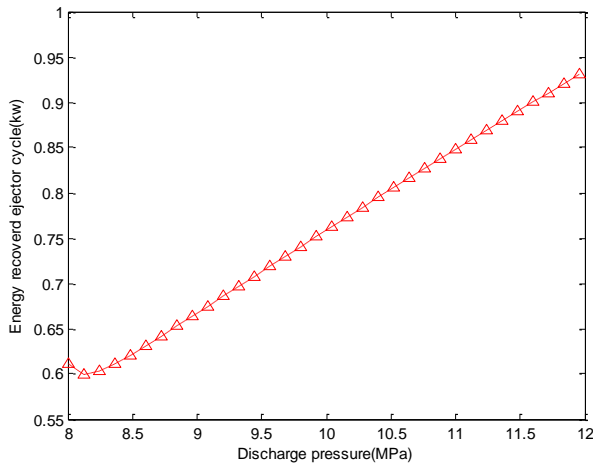
**Figure 4.** Energy recovered by the ejector cycle.
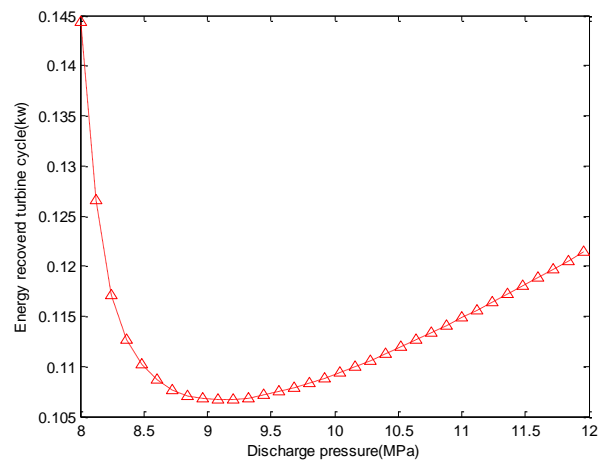
**Figure 5.** Energy recovered by the turbine cycle.

3.2 The effect of entrainment ratio on ejector cycle design parameters

The entrainment ratio of the ejector device has significant contribution to the ejector design and performance. In this section, the effects of the entrainment ratio on other design parameters are studied. Figure 6 shows that the COP_h is significantly affected by the ejector entrainment ratio. As the discharge pressure increased from (8-12 Mpa), the entrainment ratio also increases. In addition, the COP_h increased sharply to reach a maximum value of 4.9 then declined afterword. At this point, the entrainment ratio recorded a value of 0.55. This indicate that the COP_h of the ejector cycle can reach its maximum value when the mass of the secondary flow ($\dot{m}_s$) is approximately half the value of the primary mass flow ($\dot{m}_p$).

Figure 7 shows that the quality of CO_2 at the ejector outlet declines as the entrainment ratio increases. The quality decreased from 0.74 to reach approximately 0.65 at the optimum value of entertainment ratio. The decline in the CO_2 quality is due to the rise in the CO_2 mass fraction entrained from the evaporator (which is mostly vapour) against the mass fraction of the primary fluid.
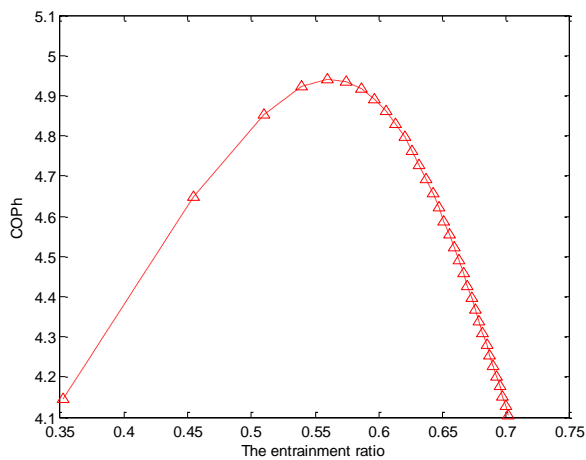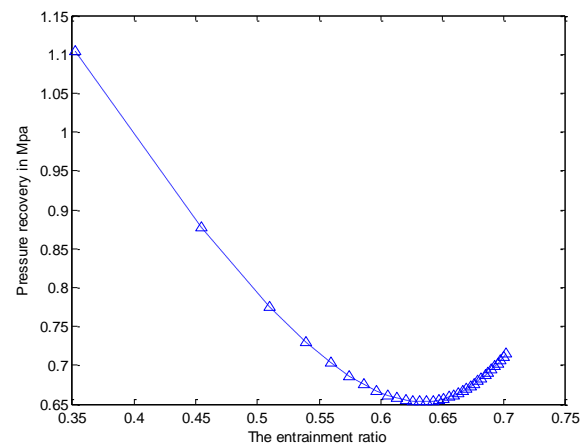

Figure 6. Relation between COP_h and entrainment ratio.
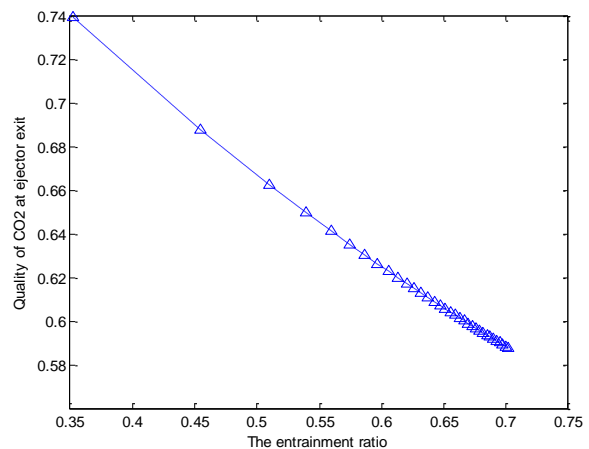
Figure 7. Relation between CO_2 quality and entrainment ratio.
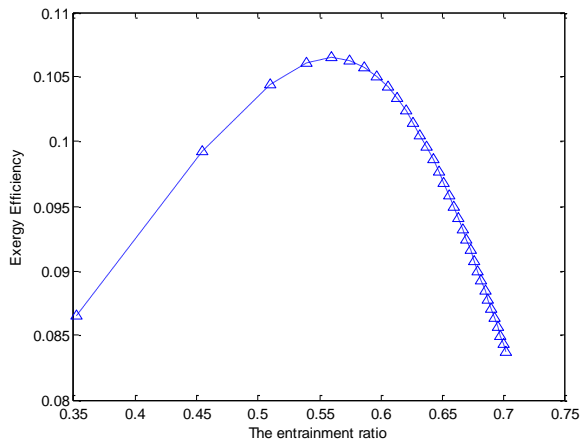
**Figure 8.** Relation between second law efficiency and entrainment ratio.

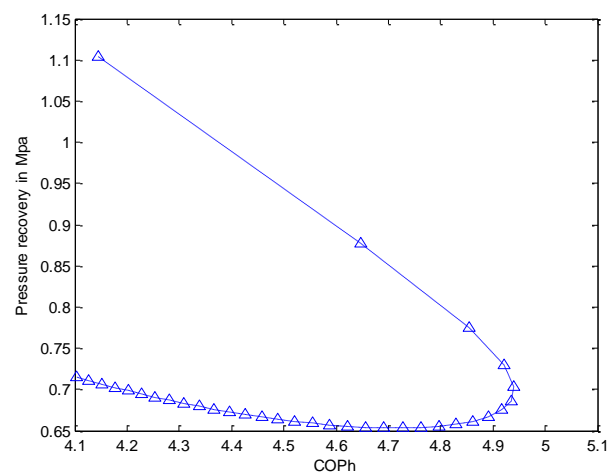**Figure 9.** Relation between pressure recovery and entrainment ratio.

Similarly, the exergy efficiency follows the same behaviour of the COP_h in relation to entrainment ratio (Figure 6). As the entrainment ratio approaches the optimum value, the exergy efficiency peaked to a value of 0.0106 before declining sharply, as shown in Figure 8. In contrast, the increase in the entrainment ratio has a negative impact on the pressure recovered by the ejector device, as shown in Figure 9. With the increase in the discharged pressure (Pd), the velocity of the CO_2 flow declined at the motive nozzle exit which causes decrease in the enthalpy of the flow ($h = u + PV$) and drops in the pressure at the ejector exit (Pc). This lead to decrease the difference between the Pc and Pe, (see equation (12)).

3.3 The effect of pressure recovery on ejector cycle design parameters
Figure 10 illustrates the pressure recovery of the ejector cycle and the refrigerant velocity (U6) in the diffuser section in correlation with the gas cooler pressure. The initial rise in discharged pressure caused significant reduction in the amount of the recovered pressure. However, after the optimum discharge pressure is reached, the pressure recovery started to improve gradually. This can be attributed to the mathematical relation between Pc, entropy and enthalpy at state 6, which are dependent on the refrigerant velocity at that state.
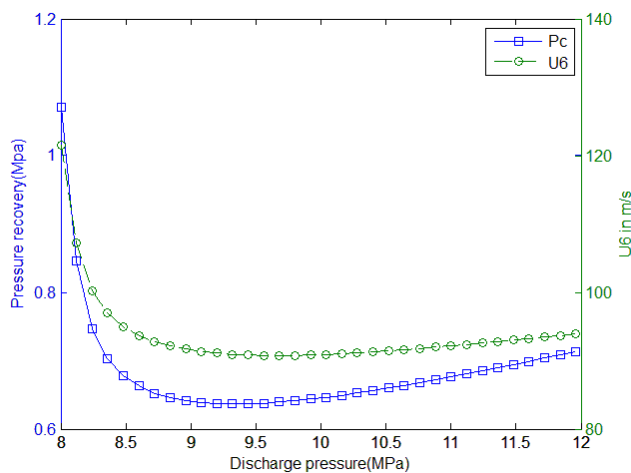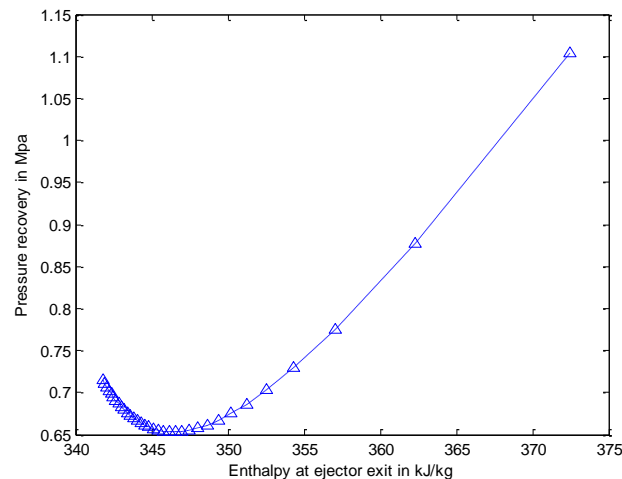

Figure 10. Relation between pressure recovery, CO_2 velocity and discharge pressure.

Figure 11. Relation between pressure recovery and COP_h.

Figure 11 shows the relation between the pressure recovery and the COP_h of the ejector cycle. It shows that the pressure recovered by the ejector device declines as COP_h decreases, however, at the highest COP_h value, the two phase ejector has recovered around 0.7 Mpa which explains the higher coefficient of performance of the ejector cycle compared with the other cycles.

Figure 12. Relation between pressure recovery and enthalpy at ejector outlet.

Figure 12 demonstrates the relation between the enthalpy of CO_2 at ejector exit and the pressure recovery of the ejector cycle. It shows that as enthalpy of the CO_2 increases, the pressure recovery slightly declines then increases. This explained as follow, the decline in the velocity of the CO_2 at the diffuser section and the amount of pressure recovered causes decline in the enthalpy at the ejector exit. After that, the enthalpy rise significantly with the improvement in the velocity and pressure recovery (as shown in Figure 10.

4. Conclusion

A simulation model has been developed to study and compare the thermodynamic performance of three transcritical CO_2 heat pump cycles utilizing different throttling devices. The mathematical modelling is conducted using MATLAB software linked to REFPROP database in order to obtain the thermophysical properties of the refrigerant in different thermodynamic states across the cycles. ASPEN plus software was used as a bench mark to validate the results obtained from the MATLAB model. The steady state comparison is based on the first and second law of thermodynamics, and the heating and cooling capacities for the three cycles. In addition, the energy recovered by the ejector and the turbine cycles is analysed. Furthermore, the pressure recovery and the entrainment ratio correlations with other design parameters for the ejector cycle were illustrated. The results show that, in term of first and second law of thermodynamics, the ejector cycle has achieved the height COP_h and exergy efficiency values of around 4.9 and 0.1, respectively. Also, the exergy destruction by the ejector device (0.2 kW) is the least of all three devices. For the heating and cooling capacities, the valve and the turbine cycles have achieved nearly similar results, which were higher than that of the ejector cycle. The ejector cycle has shown the potential advantage of energy recovery which is around five time higher than the turbine cycle.

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