MODELLING AND SIMULATION OF TRANSCRITICAL CO_2 HEAT PUMP CYCLE CONFIGURATIONS

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Abstract: Now-a-days, increased application of refrigeration and air conditioning systems is responsible for both additional energy demand and menace of ozone depletion. To fight with two jeopardizes of global warming and ozone depletion, CO_2 refrigeration and air-conditioning systems are potential replacement of current systems, due to some of the excellent properties of CO_2 such as zero ODP & GWP, compatibility with normal lubricants and common machine construction materials, non-flammability and non-toxicity, greatly reduced compression ratio, easy availability, high volumetric refrigerant capacity, and excellent heat transfer properties. In this paper thermodynamic analysis of the modified CO_2 transcritical heat pump cycles such as internal exchanger, work recovery turbine and ejector is discussed. Performances of different configurations are presented in terms of maximum COP and optimum discharge pressure. Results exhibit increase in performance of transcritical cycle by adopting ejector configuration.

***Keywords* – Transcritical CO_2, Heat Pump, Ejector.**

I. Introduction

Carbon dioxide which is a natural refrigerant and having negligible impact on climate change can be a one potential replacement refrigerant. However CO_2 refrigeration system suffers from some disadvantages too. The usually reported disadvantage of CO_2 system is loss of capacity and low COP at high heat rejection temperatures and this loss occurs due to larger throttling losses [1 & 2]. To overcome these problems, researchers suggested different cycle modifications.

One of the basic modification suggested by researchers is the use of internal heat exchanger (IHX). As proposed by Cho et al. [3], the use of IHX can improve performance of transcritical CO_2 cycle in terms of COP and cooling capacity. Sarkar et al. [4] in their study found that the optimum discharge pressure which is a critical issue in transcritical CO_2 cycle can be decreased with the use of IHX with longer length. Another modification which was put forward is the work producing expansion devices. Throttling loss in conventional transcritical CO_2 cycle is one of the major contributors to the cycle irreversibility and this irreversibility can be reduced by employing work producing expansion turbine as proposed by the Robinson and Groll [5]. Construction and testing of piston-cylinder type work producing expansion device by Baek et al. [6], showed enhancement in system performance.

Similarly, for improvement of conventional transcritical CO_2 cycle, several modifications were suggested such as multi-staging [7], parallel compression economization [8], vortex tube expansion cycle [9, 10], etc. Kronhauser [11] proposed modification of using ejector as expansion device which reduces throttling losses as well as work of compression. Also Sarkar [12] has mentioned in his study that ejector has no moving parts, low cost, simple structure and low maintaince requirements. Perez-Gracia et al. [13] and Li and Groll [14] have carried out the comparative study. However, modified ejector cycle is not included.

The proposed work is focused on the comparative study of energetic analysis of modified cycle configurations including the modified ejector cycle configuration.

II. CYCLES FOR COMPARISON

Four single stage transcritical CO_2 cycles are compared with respect to their energetic performances. These cycles are

2.1 Conventional Transcritical CO_2 Cycle (CC)

The layout and p-h diagram of conventional transcritical CO_2 cycle is shown in Fig. 1 (a). Simple throttle valve is used as expansion device. CO_2 is in saturated vapor condition at the entry to the compressor. The major disadvantage of this cycle is the loss due to throttling.

2.2 Cycle with Internal Heat Exchanger (IHXC)

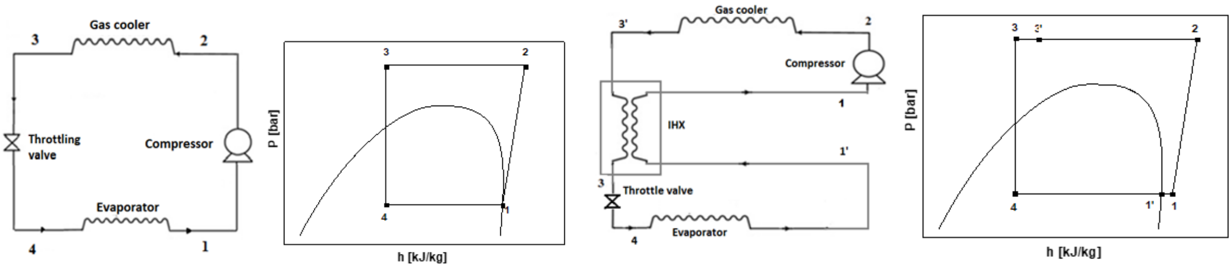
Cycle with IHX is shown in Fig. 1 (b). The use of IHX in conventional transcritical CO_2 system leads to the increase of cooling capacity [15]. But IHX cycle has one drawback that there is increase of compression work also [16].

2.3 Cycle with Work Recovery Turbine (TC)

The layout and p-h diagram of work recovery expansion turbine cycle is shown in Fig. 2 (c). Basically use of expansion turbine helps to reduce throttling loss which occurs in conventional cycle and ultimately reducing cycle irreversibility.

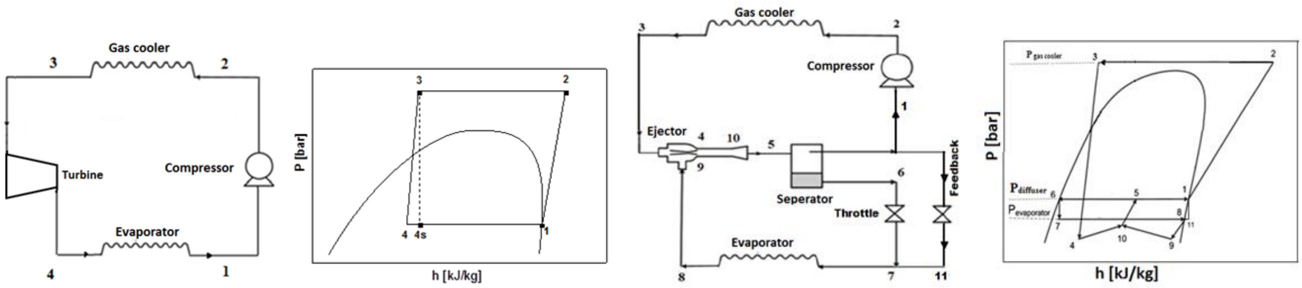
2.4 Modified Ejector Cycle (EC)

Modified ejector cycle is shown in Fig. 2 (d). A typical ejector consists of motive nozzle, suction nozzle, mixing chamber and diffuser. The high pressure vapor refrigerant from gas cooler flows through motive nozzle which is called as primary fluid and vapor refrigerant from evaporator flows through suction nozzle which is called as secondary fluid. The high pressure primary fluid gets expanded in motive nozzle and therefore low pressure region is created at the exit tip of motive nozzle. This low pressure creates the pressure difference between exit tip of motive nozzle and the inlet to the suction nozzle, which causes entrainment of secondary fluid. These fluids mix in mixing section and then the mixture goes to diffuser where kinetic energy is converted to pressure energy. The most important characteristic of ejector system is ejector entrainment ratio which is the ratio of mass flow rates of secondary fluid to that of primary fluid. In conventional ejector system for mass balance purpose, the relation between ejector outlet quality (X_5) and ejector entrainment ratio (μ), $X_5 = 1/(1+\mu)$, should be satisfied. Entrainment ratio depends on motive flow, suction flow and ejector outlet pressure which leads to arduousness to control real system operating conditions. Hence modified ejector system [14] was proposed, where some part of vapor from separator is sent to evaporator inlet through throttle valve, which helps to modulate refrigerant quality at evaporator inlet.



(a) (b)

Figure 1: The layout and p-h diagram of (a) conventional transcritical CO_2 cycle (b) transcritical CO_2 cycle with IHX



(c) (d)

Figure 2: The layout and p-h diagram of (c) transcritical CO_2 cycle with turbine (d) transcritical CO_2 modified ejector expansion cycle

III. SIMULATION CONDITIONS AND EQUATIONS

Operating conditions taken for simulation of cycles are: evaporator temperature (tev) is taken as 0 ^{O}C; gas cooler exit temperature (tgc) is taken as 40 ^{O}C; isentropic efficiency of compressor (ηc) is taken as 0.75 [17]; isentropic efficiency of motive nozzle (ηm) is taken as 0.8 [18]; isentropic efficiency of suction nozzles (ηs) is taken as 0.8 [18]; isentropic efficiency of diffuser (ηd) is taken as 0.75 [18]; ejector entrainment ratio (μ) is taken as 0.6 [18]; internal heat exchanger effectiveness (ε) is taken as 0.6 [18]. Table 1 shows the different components and their corresponding characteristic equations.

Table 1: Components and their corresponding characteristic equations

Sr. no.	Component	Cycle	Corresponding emblematic equations	Sr. no.	Component	Cycle	Corresponding emblematic equations
1	Evaporator	CC	$q_{ev} = h_1 - h_4$	3	Compressor	CC	$w_c = h_2 - h_1$
		IHXC				IHXC	
		TC				TC	
		EC	$q_{ev} = \left(\frac{\mu}{1+\mu}\right) h_8 - (1 - x_5) h_6 - \left(x_5 - \frac{1}{(1+\mu)}\right) h_1$			EC	
				4	Throttle valve	CC	$h_{in} = h_{out}$
						IHXC	
						EC	
				5	IHX	IHXC	$\epsilon = \frac{(t_1 - t_{1'})}{(t_3 - t_{1'})}$
				6	Turbine	TC	$\eta_c = \frac{(h_3 - h_4)}{(h_3 - h_{4s})}$
2	Gas cooler	CC	$q_{gc} = h_2 - h_3$	7	Motive nozzle	EC	$\eta_m = \frac{(h_4 - h_3)}{(h_{4s} - h_3)}$
		IHXC		8	Suction nozzle	EC	$\eta_s = \frac{(h_9 - h_8)}{(h_{9s} - h_8)}$
		TC		9	Diffuser	EC	$\eta_d = \frac{(h_{5s} - h_{10})}{(h_5 - h_{10})}$
		EC					

IV. RESULT AND DISCUSSION

All the cycle configurations mentioned above are analyzed based on the first law of thermodynamics considering the steady state flow. Evaporator temperature is taken as 0 ^{O}C and gas cooler exit temperature is take as 40 ^{O}C. Evaporator temperature is varied from 0 to -40 ^{O}C while gas cooler temperature varied from 40 ^{O}C to 60 ^{O}C. As mentioned earlier that owing to S shape of the isothermal lines there is an optimum gas cooler pressure exists at the given gas cooler outlet temperature.

Fig. 3 exhibits the variation of COP with gas cooler pressure. It can be seen that COP increases with increase in gas cooler pressure and attains maximum value at certain pressure which is known as optimum pressure and after that starts decreasing. This happens due to s-shape of isotherm above critical point. Also it can be observed that ejector cycle is profitable in terms of COP at high gas cooler pressures as compared to low gas cooler pressures. This happens because at low gas cooler pressure for given evaporator temperature, the pressure drop in motive nozzle is lower, which results lower kinetic energy of refrigerant at exit of motive nozzle and in turn lower pressure increase in diffuser. This means pressure lift ratio (PLR) is smaller at lower gas cooler pressure and this causes decease in COP values.

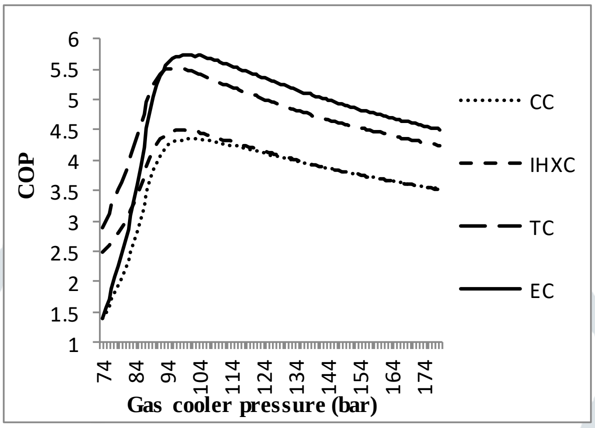
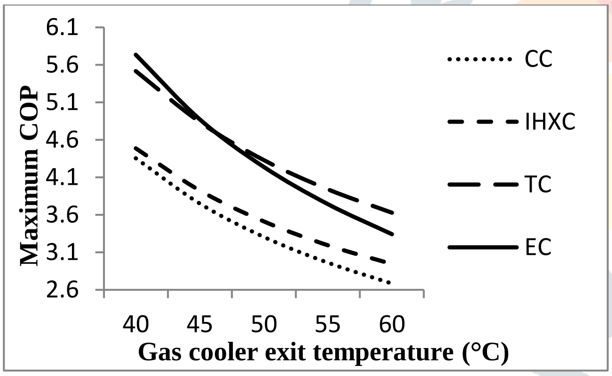
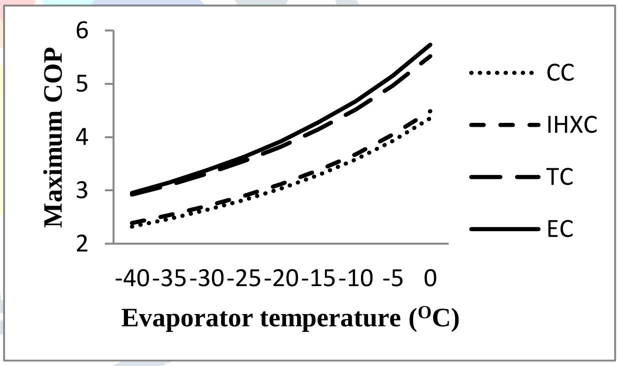


Figure 3: COP versus gas cooler pressure for given simulation conditions

Fig. 4 (e) and (f) exhibit that maximum COP values decrease with increase in gas cooler exit temperature and decrease with evaporator temperature. Effect of evaporator temperature is marginal on system COP as compared to that of gas cooler exit temperature.



(e)



(f)

Figure 4: (e) Maximum COP versus gas cooler exit temperature at 0 ^{O}C evaporator temperature (f) Maximum COP versus evaporator temperature at 40 ^{O}C gas cooler exit temperature

It can be seen that Fig. 5 that optimum gas cooler pressure increases with increase in gas cooler exit temperature. Use of internal heat exchanger or expansion turbine gives lower optimum pressure as compared to conventional and ejector cycles. Due to use of internal heat exchanger subcooling takes place and this leads to increase in specific refrigeration effect, which in turn causes decrease in optimum pressure. In case of expansion turbine, nearly isentropic expansion process causes slight increase in specific refrigeration effect and at the same time work produced by turbine is given to compressor which causes decrease in power consumption. Hence optimum pressure is lower in case of expansion turbine cycle.

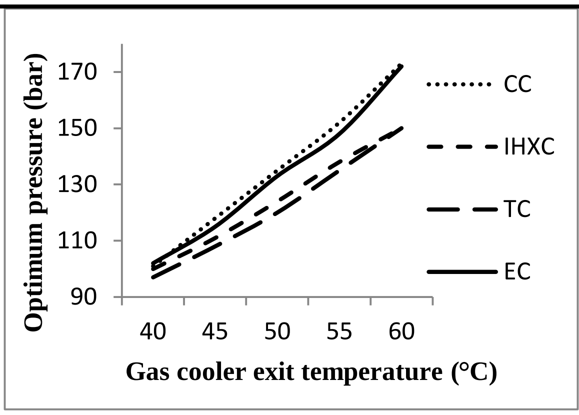


Figure 5: Optimum pressure versus gas cooler exit temperature at 0 ^{O}C evaporator temperature

V. CONCLUSION

Energetic analysis and comparative study of different cycle configurations has been carried out. Study shows that at higher gas cooler pressure ejector system is more beneficial in terms of system performance. Effect of evaporator temperature on system performance is negligible as compared to that of gas cooler exit temperature. The use of ejector as expansion device has reduced optimum pressure and has increased system COP as compared to conventional cycle.

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