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Performance Evaluation of Ejector Enhanced Transcritical R744 System

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***Abstract*—R744 is the preferred natural refrigerant. R744 cycle operates in transcritical mode with relatively high pressure which leads to high throttling loss. Employing ejector as an expansion device reduces throttling losses by gain in pressure at the compressor inlet. The present study deals with the optimization of the modified ejector transcritical R744 based on the steady state energetic analysis. Further, the results are also compared with the conventional transcritical R744 cycle (CTCC). The use of ejector shows the performance improvement in terms of COP, the reduction in optimum gas cooler pressure.**

Keywords—Carbon dioxide, Ejector, Transcritical R744

I. INTRODUCTION

The unprecedented growth of energy consumption due to increased demand of HVAC&R systems is putting the adverse effect on the environment in terms of carbon footprints. Moreover, finding a suitable alternatives of the existing refrigerants CFCs and HCFCs is a great concern because of ill effects caused by them: global warming as well as ozone layer depletion. One of the prospective replacement refrigerants is the R744, which is a natural occurring refrigerant and has trifling influence on climate change. There are other advantages also associated with the use of R744 as refrigerant such as non- flammability, non-toxicity, easy availability, high volumetric refrigerant capacity, and admirable heat transfer properties [1]. However, R744 systems are vulnerable at high heat rejection temperatures with loss of capacity and low COP. One of the reasons may be due to high throttling and superheat horn losses at high heat rejection temperatures [2]. Various modifications such as use of internal heat exchanger, multi-staging, work producing device as an expansion device, ejector as an expansion device are suggested by various researchers in the past to overcome these problems.

Expansion in the conventional throttling devices is always considered as an isenthalpic expansion process, however it brings the throttling losses which in turn lead to loss of cooling capacity and COP [3]. Ejector is one of the preferred choice expansion devices that has no parts with relative motion, affordable price, simple assembly and niggling upkeep requirements [1]. Ejector can help to recover some of the throttling losses by increasing the pressure at the compressor inlet as suggested by Kornhauser [4]. It is reported that an ejector expansion cycle with R12 as refrigerant brings 21% COP improvement over conventional cycle [4]. Takeuchi et al. [5] patented ejector system using R744 as refrigerant with critical refrigerant pressure. Elbel and Hrnjak [6] in their study of variable throat area ejector expansion transcritical R744 cycle found that employing ejector improves COP and cooling capacity by 7% and 8%, respectively by recovering the work of an about of 14.5%. For maintaining mass conservation, a relation between ejector entrainment ratio and quality at ejector outlet, is one of the constraints of the conventional ejector system which limit its use in practice. Li and Groll [7] proposed a modified ejector system with a feedback mechanism from the separator to evaporator to maintain quality at inlet of the evaporator via throttle valve that overcome the constraint of mass conservation of an ejector cycle. Sarkar [8] carried out optimization of the cycle proposed by Li and Groll [7] and developed the correlation for optimum system design with apposite operating conditions.

The paper includes a comparative study of ejector expansion transcritical R744 cycle with conventional transcritical R744 system based on the energetic analysis.

II. THERMODYNAMIC ANALYSIS

The schematic diagrams of cycle along with the corresponding p-h diagram of modified ejector expansion transcritical R744 cycle (MTCEEC) [7] is shown in Fig. 1.

The high-pressure primary refrigerant flows from gas cooler to motive nozzle. The refrigerant is accelerated in the motive nozzle, which enables to entrain the refrigerant from evaporator, which is known as secondary flow. The mixing of primary and secondary flows takes place in the constant area mixing section and as flow passes through diffuser deceleration of flow takes place and leads to increase in pressure. From diffuser flow leads to the separator where vapor and liquid are separated and the liquid refrigerant is passed to evaporator throttle valve. From separator, some part of vapor is passed to the evaporator via throttle valve, so that, at evaporator inlet quality of refrigerant is controlled and another part passes through the internal heat exchanger where it exchanges heat with the gas cooler exit stream and then it is passed to the compressor. Fig. 2 exhibits the schematic diagram and p-h diagram of the conventional transcritical R744 cycle (CTCC) with throttle valve as an expansion device.

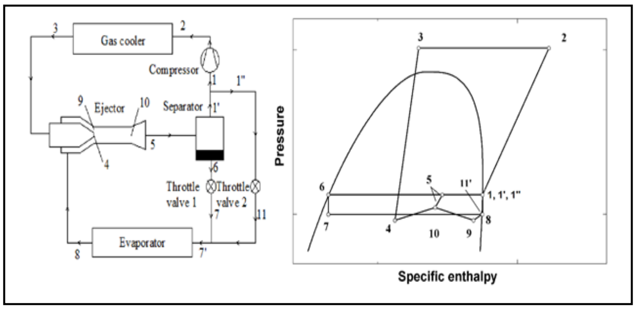

Fig. 1 Cycle layout and p-h diagram of MTCEEC

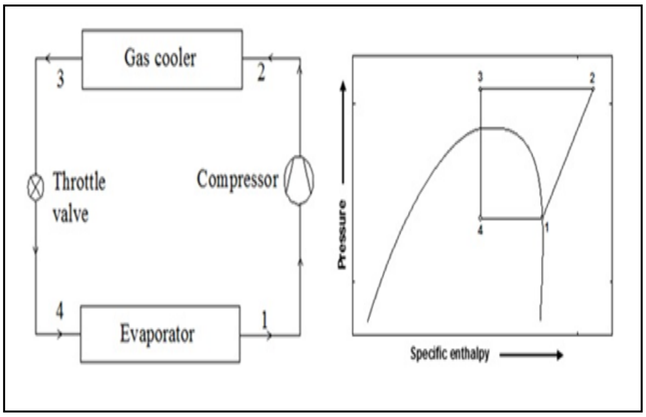

Fig. Cycle layout and p-h diagram of CTCC

The entrainment ratio of ejector (μ) is defined as

$$\mu = \frac{\text{suction fluid mass flow rate}}{\text{motive fluid mass flow rate}} = \frac{m_s}{m_m} \quad (1)$$

The pressure lift ratio (PLR) is defined as

$$PLR = \frac{\text{fluid pressure at inlet to the compressor}}{\text{fluid pressure at exit of evaporator}} \quad (2)$$

The analysis is done assuming unit mass flow rate at the exit of diffuser. Applying the conservation of mass at component level, the mass flow rates relation can be given as:

$$m_{ms} = m_d = 1 \quad (3)$$

$$m_{com} = m_{gc} = m_{IHX} = m_m = \frac{1}{1+\mu} \quad (4)$$

$$m_{ev} = m_s = \frac{\mu}{1+\mu} \quad (5)$$

The different components and their corresponding emblematic equations are shown in Table 1.

The combined system COP (cooling and heating) is given as:

$$COP = \frac{q_{ev} + q_{gc}}{w_{com}} \quad (6)$$

The 2nd law efficiency of the system is given as:

$$\eta_{II} = 1 - \frac{I_{tot}}{w_{com}} \quad (7)$$

To perform the thermodynamics analysis and to simulate the cycles, a computer programming code is developed. Isentropic efficiency of the motive nozzle and suction nozzle is taken as 0.8 while for diffuser it is assumed to be as 0.7 [8]. It is presumed that compressor isentropic efficiency is 0.75 [9].

III. RESULTS AND DISCUSSION

The performance evaluation is carried out for the modified ejector expansion transcritical R744 cycle (MTCEEC) based on the energetic analysis. Evaporator temperature is altered from 0^0C to – 40^0C while gas cooler temperature altered from 40^0C to 60^0C. It is well known fact that due to transcritical operation of the R744 where temperature lines are of 'S' shape, there occurs an optimum pressure for a specified exit temperature of gas cooler at which system runs optimally. The present analysis is based on the optimum conditions; accordingly, pressure in the gas cooler is varied from 74 to 180 bar with an increment of 1 bar. Results are also compared with Conventional Transcritical R744 (CTCC) cycle.

Fig. 3 exhibits the variation of combined COP with pressure in the gas cooler. One can observe that there is existence of an optimum pressure at which system performs optimally for all the two systems. Further, it is witnessed that employing the ejector is beneficial from point of COP as the ejector cycles exhibits higher COP compare to conventional transcritical R744 cycle (CTCC). This is because due to pressure lift in ejector and correspondingly reduction in compressor work. Optimum pressure value is also relatively less with MTCEEC compared to CTCC.

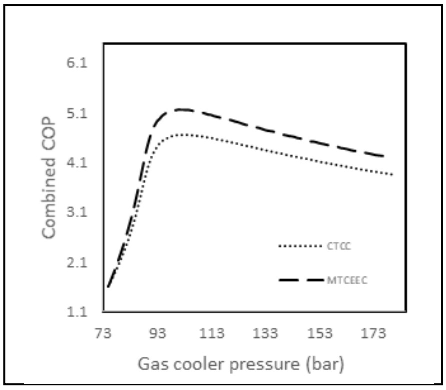

Fig. Variation of combined COP with gas cooler pressure at $t_{gc,out}$= 40 ^{0}C, t_{ev} = -5 ^{0}C

Fig. 4 shows the variation of the maximum combined COP and the optimum pressure in gas cooler with respect to the exit temperature of gas cooler at an evaporator temperatures of -40^0C. For all cycles, there is decrease in the maximum combined COP and increase in the optimum pressure in gas cooler when exit temperature of gas cooler increases. It can be seen that ejector cycle performs marginally better with relatively higher COP. Despite of that, the optimum pressure in gas cooler is significantly lower compare to CTCC.

The Fig. 5 depicts the change of the maximum combined COP and the optimum pressure with respect to evaporator temperature at chosen exit temperature of gas cooler 60^0C. The maximum combined COP increases and whereas the optimum pressure marginally drops with elevation in the evaporator temperature for all two cycles. Ejector cycle marginally better performance in terms of COP and has lower optimum gas cooler pressure in comparison to conventional systems.

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Table 1 Components and their corresponding emblematic equations

Sr. no.	**Component**	**Cycle**	**Corresponding emblematic equations**	
			Cooling/Heating capacity or Work of compression or Energy balance	Efficiency or other equations
1	Evaporator	CTCC	$q_{ev} = m_r(h_1 - h_4)$	---------
		MTCEEC	$q_{ev} = [m_s h_8 - m_6 h_6 - m_1'' h_1]$	
2	Gas cooler	CTCC	$q_{gc} = m_r(h_2 - h_3)$	---------
		MTCEEC	$q_{gc} = (m_m)(h_2 - h_3)$	
3	Compressor	CTCC	$w_{com} = m_r(h_2 - h_1)$	$\eta_{com} = \frac{(h_{com,out} - h_{com,in})}{(h_{com,out,is} - h_{com,in})}$
		MTCEEC	$w_{com} = (m_m)(h_2 - h_1)$	
4	Throttle valve	CTCC	$h_{in} = h_{out}$	---------
		MTCEEC		
5	Motive nozzle	MTCEEC	$h_3 = h_4 + \frac{V_4{}^2}{2}$	$\eta_m = \frac{(h_3 - h_4)}{(h_3 - h_{4is})}$
6	Suction nozzle	MTCEEC	$h_8 = h_9 + \frac{V_9{}^2}{2}$	$\eta_s = \frac{(h_8 - h_9)}{(h_8 - h_{9is})}$
7	Mixing section	MTCEEC	$\left(h_{10} + \frac{V_{10}{}^2}{2}\right) = (m_m)\left(h_4 + \frac{V_4{}^2}{2}\right) + (m_s)\left(h_9 + \frac{V_9{}^2}{2}\right)$	Momentum equation $P_{10}(a_4 + a_9) + V_{10} = P_9(a_4 + a_9) + \frac{1}{1+\mu}V_4 + \frac{\mu}{1+\mu}V_9$
8	Diffuser	MTCEEC	$h_5 = h_{10} + \frac{V_{10}{}^2}{2}$	$\eta_d = \frac{(h_{5is} - h_{10})}{(h_5 - h_{10})}$
9	Ejector	MTCEEC	Overall energy balance for ejector $h_5 = \frac{h_3 + \mu h_8}{(1+\mu)}$	

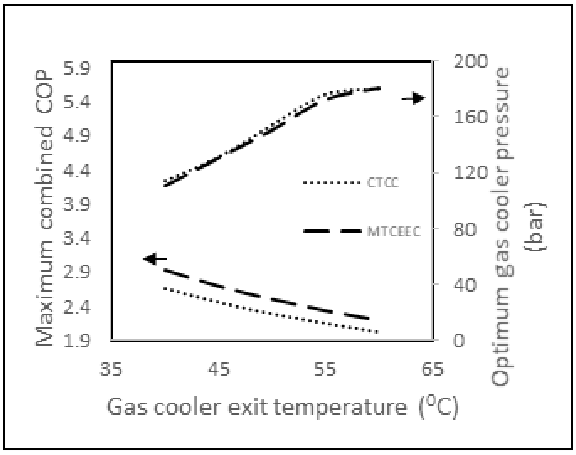

Fig. 4 Variation of maximum combined COP and optimum gas cooler pressure with gas cooler exit temperature at t_{ev} = -40 ^{0}C

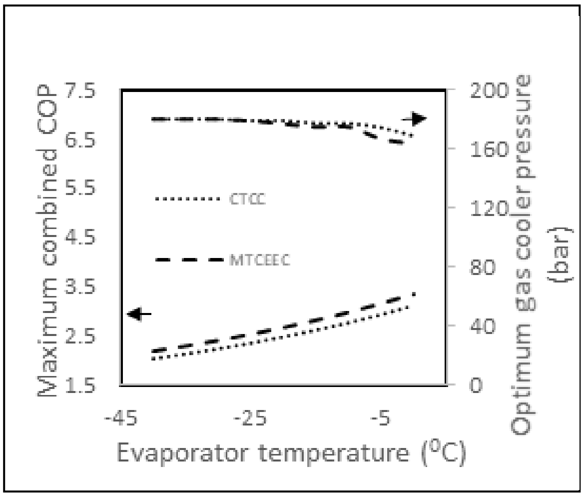

Fig. 5 Variation of maximum combined COP and optimum gas cooler pressure with evaporator temperature at $t_{gc,out}$= 60 ^{0}C

Pressure lift ratio (PLR) is one of the performance parameters for an ejector system defined as the ratio of compressor inlet pressure to that of the evaporator outlet pressure. Fig. 6 exhibits the variation of the pressure lift ratio of the chosen system It is observed that the pressure lift ratio increases with increase in gas cooler exit temperature and decrease in evaporator temperature. This may be because of the fact that, at higher temperature of gas cooler exit and at a chosen the gas cooler pressure, the exit stream from the motive nozzle of the ejector will be at higher velocity and correspondingly at a lower pressure which results in the higher lift of pressure in the diffuser section of the ejector.

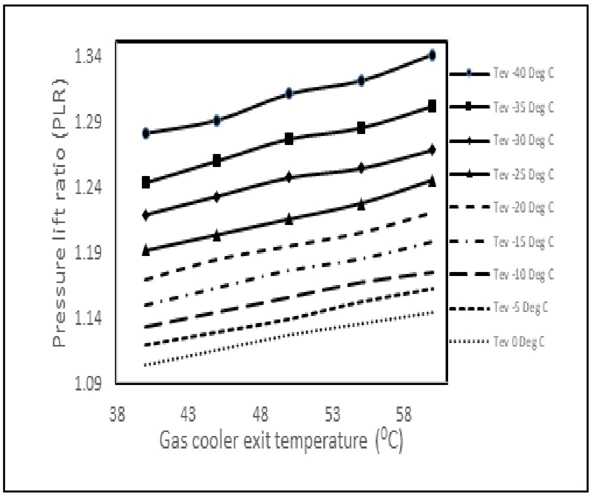

Fig. 6 Variation of pressure lift ratio with gas cooler exit temperature for different evaporator temperatures

IV. CONCLUSION

A comparative study based on the steady state thermodynamic analysis of the two different cycles; (CTCC) and (MTCEEC) is carried out. Study shows that the presence of ejector improves the functioning of MTCEEC in the terms of COP. Further, the use of ejector also reduces the system optimum pressure for the chosen operating conditions, which is beneficial in terms of system safety. As like conventional transcritical R744 system, ejector transcritical R744 cycle is also more influential with exit temperature of the gas cooler than that of temperature of the evaporator. The Pressure Lift Ratio (PLR) which is one of the design parameter of an ejector enhances with elevation in exit temperature of gas cooler and drop in temperature of evaporator.

NOMENCLATURE

A. *Nomenclature*

a	area (m2)
COP	coefficient of performance
CTCC	conventional transcritical R744 cycle
h	specific enthalpy (kJ/kg)
m	mass flow rate (kg/s)
MTCEEC	modified transcritical R744 ejector expansion cycle
P	pressure (bar)
PLR	pressure lift ratio
q	Specific cooling/heating capacity (kJ/kg)
T, t	temperature (K, 0C)
V	velocity (m/s)
w	specific work (kJ/kg)

B. *Greek symbols*

η	efficiency

μ	entrainment ratio

C. Subscripts

1-11, 1', 3', 7', 1'', 1'''	refrigerant state points
com	compressor
d	diffuser
ev	evaporator
gc	gas cooler
in	inlet condition
is	isentropic state
m	motive nozzle
ms	mixing section
out	outlet condition
r	refrigerant
ref	evaporator secondary fluid
s	suction nozzle
sep	separator
tot	total
tv1, tv2	throttle valve

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