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Thermodynamic Analysis and Optimization of CO_2 based Transcritical Cycle

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Abstract*—The Montreal and Kyoto protocol are two frameworks towards a single goal of environment safety. These protocols suggest prohibiting the usage of synthetic refrigerants to prevent ozone layer depletion and control global warming as well. Such conditions encourage us to consider CO_2 as a working fluid for refrigeration and air conditioning systems. In this paper, thermodynamic analysis of CO_2 based transcritical cycle is presented to show the effect of various operating parameters of transcritical cycle. The operating parameters considered in this study include heat rejection pressure in gas cooler, evaporator temperature and gas cooler exit temperature. At the end, three useful correlations that yield the optimal heat rejection pressure in gas cooler, the associated maximum COP, and optimum compressor discharge temperature in the transcritical cycle are presented.
***Keywords*— CO_2; Transcritical cycle; Heat rejection pressure; COP; Heat pump; Refrigeration**

I. INTRODUCTION

With the growth of world's economy use of refrigeration systems are increasing, which increases emission potential of refrigerants to the environment with its negative effect. It was discovered that some refrigerants causes ozone layer depletion and global warming, which is a serious hazard to environment. In order to control the depletion of ozone layer, in 1987 Montreal protocol phasing out some synthetic refrigerant but it did not cover the global warming potential of the refrigerants. Refrigerants which lead to global warming are advised to prohibit by Kyoto Protocol released in 2011. Therefore, for the sake of environmental safety[1], we again concern the natural refrigerants such as carbon dioxide, ammonia, hydrocarbons etc. due to their zero ozone depletion potential (ODP) and low global warming potential (GWP).

Among natural refrigerants CO_2 could be an important alternative to synthetic refrigerant due to its some useful characteristics such as non-toxic, odourless, non-flammable, low price and easy availability. Even carbon-dioxide is a greenhouse gas but as it is captured from environment therefore any leakage of CO_2 does not increase the overall volume of CO_2 present in environment thus does not contribute to global warming [2].

CO_2 had already been used as refrigerant in early twentieth century for marine application. In the history of refrigerants, CO_2 was used in vapour compression system as a refrigerant first time (proposed by Alexander Twining) in 1850. First CO_2 based ice production machine was built by Thaddeus S. C. Lowe in about 1869 in Jackson, Mississippi. In 1886, a German engineer, Franz Windhausen designed a CO_2 compressor after which CO_2 was used widely for marine and general refrigeration applications [3]. After the development of fluorocarbons during 1930s CO_2 based refrigeration systems were completely phased out till 1950 [4] because fluorocarbon based refrigeration system had low operating pressure which resulted in higher coefficient of performance (COP) as compared to CO_2 based refrigeration system. But because of above mentioned Montreal and Kyoto protocol our interest renewed in CO_2 based refrigeration systems.

Carbon dioxide has NBP is -87.84 ^{0}C and critical temperature 30.98 ^{0}C. Due to very low NBP it can be used in very low temperature refrigeration applications (deep freezing and lower circuit refrigerant of a cascade refrigeration system) and due to low critical temperature CO_2 works in transcritical cycle (modified vapour compression cycle) for heating purpose. Liao *et.al* (2000) presented a cycle simulation model to optimize the COP of transcritical cycle for air-conditioning. Sarkar *et.al.* (2004) presented energetic and exergetic analyses for optimization of a transcritical carbon dioxide heat pump system and shows that Carbon dioxide based transcritical cycle may be more attractive if heating effect by gas cooler and refrigeration effect by evaporator considered simultaneously [5]. A numbers of studies carried out in last decade on the thermodynamic analyses of transcritical systems with carbon dioxide as refrigerants [6-11].

In this study, thermodynamics analysis of CO_2 based transcritical cycle is presented to know the effect of various operating and design parameters which includes heat rejection pressure in gas cooler, gas cooler exit temperature, effectiveness of heat exchanger and evaporator temperature. In the present work Optimization of heat rejection pressure is done for, wide range of evaporator temperature from -20 ^{0}C to 20 ^{0}C and gas cooler exit temperature range from 32 ^{0}C to 50 ^{0}C.

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II. SYSTEM DESCRIPTION

CO_2 has low critical temperature of 30.98 ^{0}C, so a CO_2 based vapour compression system with normal refrigeration temperature will work close to and even partly above the critical pressure (7378 kPa), i.e. evaporation takes place below the critical pressure similar to other refrigerants and heat rejection takes place above the critical pressure.

Hence, the modified vapour compression cycle for CO_2 is called a transcritical cycle, which is partly subcritical during evaporation and partly supercritical during heat rejection. Above the critical pressure CO_2 becomes very dense gas. Under this condition heat rejection cannot take place in the condenser (constant temperature), so heat rejection is carried out in gas cooler with temperature gliding effect. In gas cooler CO_2 is cooled with the help of external fluid as shown in fig. 1. This has been used to great advantage in water-heating heat pumps for a range of applications from domestic to industrial.

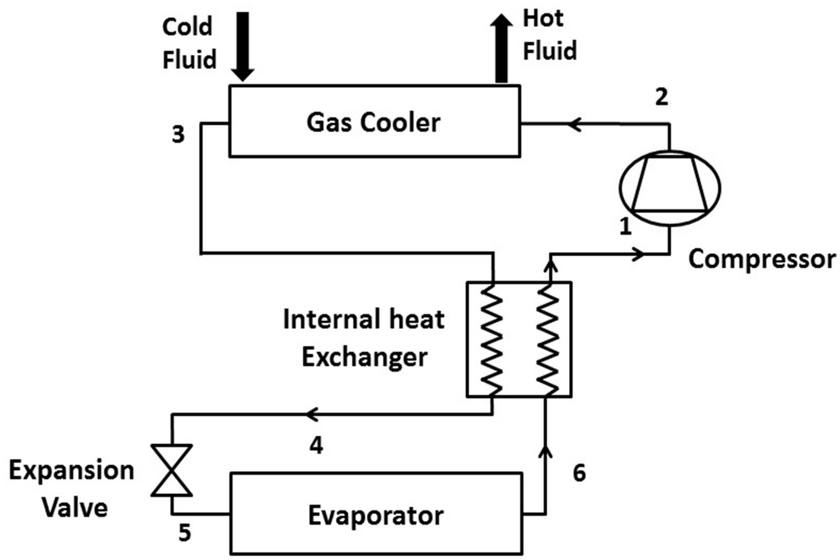

Fig. 1 Schematic diagram of a transcritical CO_2 system

Fig. 2 shows the T-s diagram for the transcritical cycle of CO_2. The superheated CO_2 is compressed by compressor from state 1 to state 2. In the gas cooler the compressed CO_2 rejects heat to the external fluid and reach at state 3. Unlike a condenser, in the gas cooler heat rejection take place with a gliding temperature. The refrigerant vapour from the evaporator is superheated (state 6 to 1) in the internal heat exchanger with consequent subcooling (state 3 to 4) then CO_2 expands through throttling valve to state 5. From state 5 the CO_2 evaporates in the evaporator to reach the saturated vapour state 6.

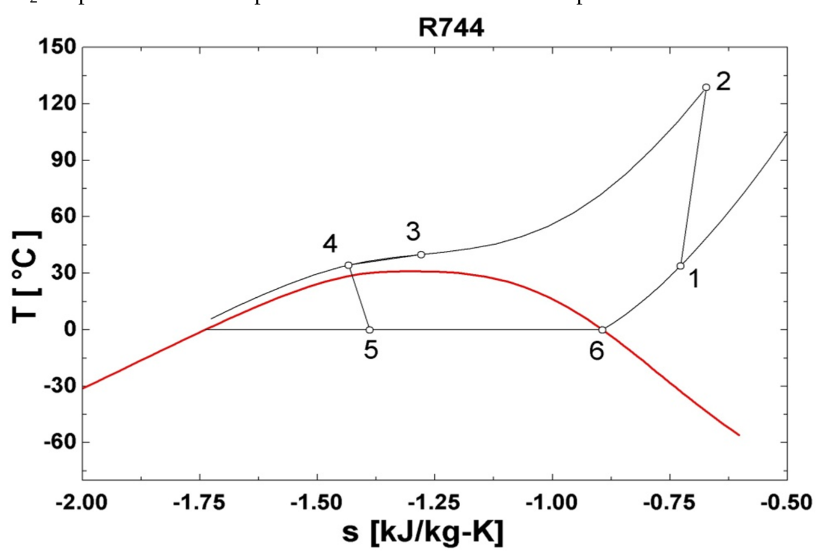

Fig. 2 Temperature-Entropy diagram of transcritical CO_2 cycle

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III.THERMODYNAMIC ANALYSIS

The thermo-physical properties of CO_2 specified in this work were calculating using a software package called engineering equation solver (EES) [12]. A major feature of EES is the high accuracy thermodynamic and transport property database that is provided for hundreds of substances in a manner that allows it to be used with the equation solving capability.

The following assumptions have been made in the analysis of this transcritical cycle:

A. The system is at steady state condition. All processes are steady flow processes.
B. Compression process is adiabatic with an isentropic efficiency of 0.70
C. Pressure drop in the connecting pipes and heat exchangers are negligible.
D. Heat transfer with the ambient is negligible
E. Single-phase heat transfer has been considered for the external fluid in gas cooler.

Refrigerating effect in evaporator:

$$q_E = (h_6 - h_5)$$

Work input to compressor:

$$w_C = \frac{(h_2 - h_1)}{\eta}$$

Heat rejected in gas cooler:

$$q_{GC} = (h_2 - h_3)$$

Energy balance for the internal heat exchanger:

$$(h_1 - h_6) = (h_3 - h_4)$$

and effectiveness:

$$\in = \frac{(T_1 - T_6)}{(T_4 - T_6)}$$

Energy balance for the entire system:

$$q_E + w_C = q_{GC}$$

η= 0.70 is an average value for the most modern compressors [14]. The exit temperature of CO_2 from gas cooler at state 3 is dependent on external fluid inlet temperature; hence, at any discharge pressure, cooler exit temperature will be fixed for a certain fluid inlet condition. Coefficient of performance for transcritical cycle is given by:

$$COP_{heating} = \frac{q_{GC}}{w_C}$$

$$COP_{cooling} = \frac{q_E}{w_C}$$

$$COP_{system} = COP_{heating} + COP_{cooling}$$

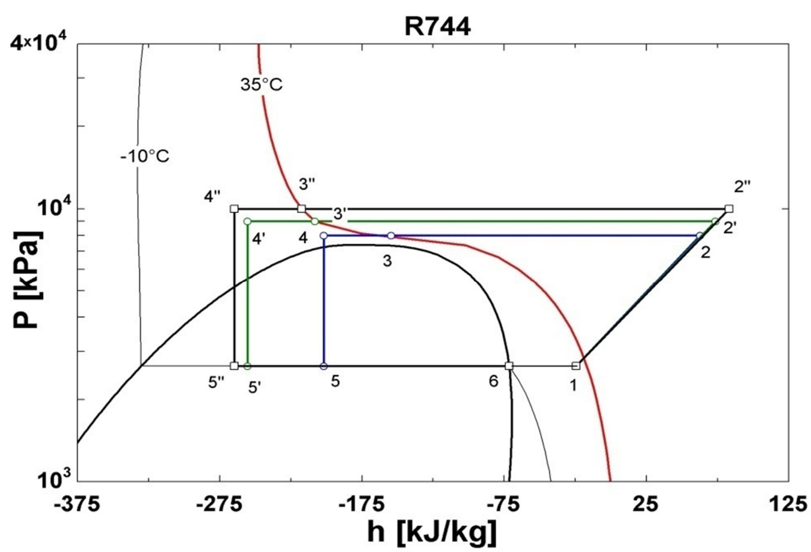

Fig. 3 Pressure-Enthalpy diagram of transcritical CO_2 cycle

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IV. RESULTS AND DISCUSSION

It is well known that for the conventional subcritical system the coefficient of performance (COP) increases with the decrease of the heat rejection (condensation) pressure. For the transcritical carbon dioxide cycle, however, the variation of the COP with the heat rejection pressure exhibits a non-monotonic change due to the fact that the heat rejection temperature is independent of the heat rejection pressure in the supercritical region.

Fig. 3 shows that above the critical point the slope of the isotherm is quite modest for a specific pressure range (from state 3 to state 3') but beyond this pressure range the isotherms are quite steep. Due to this fact COP of this system increases up to a certain pressure and after that COP will decreases with further increase in pressure. At a particular pressure, the COP attains a maximum value and the corresponding pressure is termed as optimum pressure for the cycle. Optimum pressure of heat rejection in the gas cooler depends on the operating conditions.

A. Effect of heat rejection pressure at different gas cooler exit temperature

Fig. 4 shows the effect of heat rejection pressure on COP at different gas cooler exit temperature and 0 ^{0}C evaporating temperature of CO_2. It is clear from the figure that COP of the system (simultaneous heating and cooling) increases with increase in heat rejection pressure and at a particular pressure it become maximum, and then decreases. The pressure, at which COP of the transcritical cycle is maximum, is known as optimum heat rejection pressure.

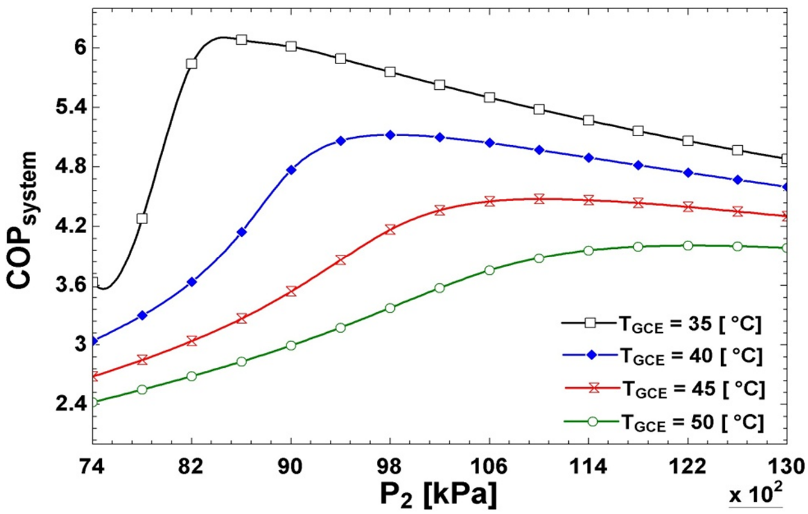


Fig. 4 Variation in COP with heat rejection pressure at different gas cooler exit temperature

It can also be observed from the figure that optimum heat rejection pressure increases with increase in gas cooler exit temperature, however COP of the transcritical cycle decreases for the same.

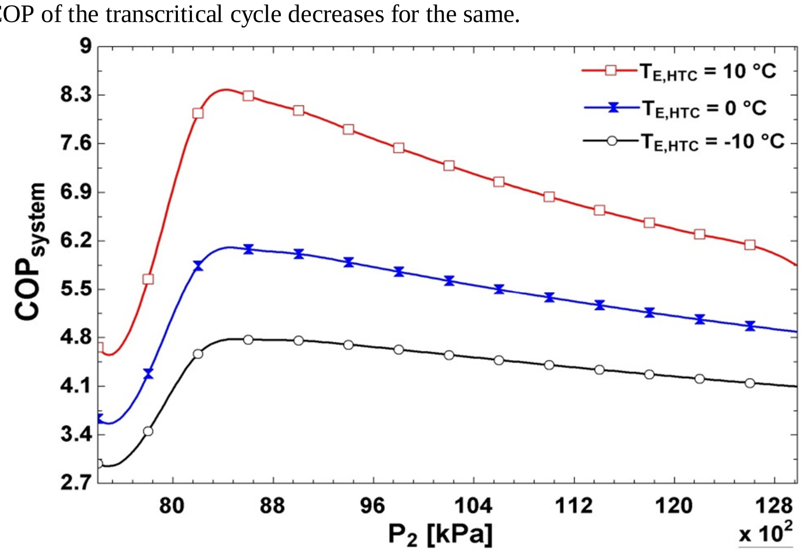


Fig. 5 Variation in COP of HTC with heat rejection pressure at different evaporator temperatures

B. Effect of heat rejection pressure at different evaporator temperatures

Fig. 5 shows the effect of heat rejection pressure on COP at different evaporating temperatures of CO_2 and fixed gas cooler exit temperature (35 ^{0}C). It is clear from the figure that COP of the system increases with increase in heat rejection pressure and become maximum at optimum pressure, then decreases.

It can also be observed from the figure that COP of the system increases with increase in evaporator temperature.

C. Effect of effectiveness of internal heat exchanger at different evaporator temperatures

Fig. 6 shows the effect of effectiveness of internal heat exchanger at different evaporating temperatures of CO_2 and fixed gas cooler exit temperature (35 ^{0}C). It can be observed from the figure that COP of system increases with evaporator temperature and effectiveness of internal heat exchanger has no effect on COP.

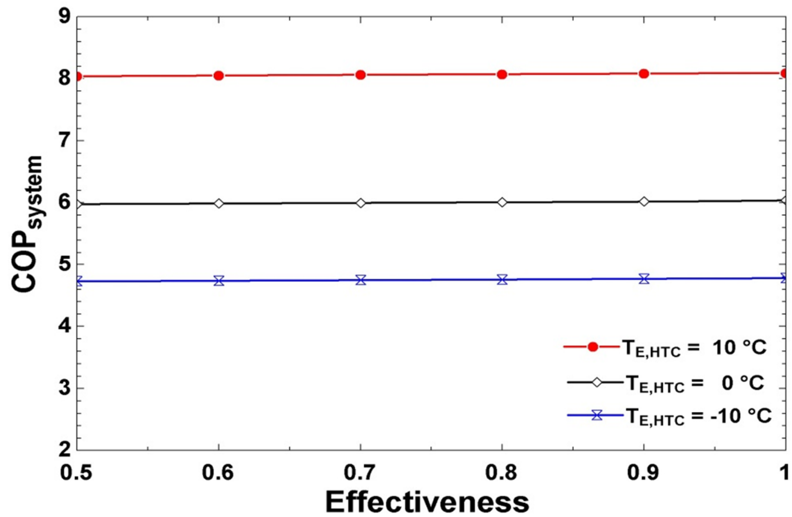


Fig. 6 Variation in COP with effectiveness of internal heat exchanger at different evaporator temperatures

From the above discussion, it is concluded that for fixed compressor isentropic efficiency, COP of the transcritical cycle is the function of evaporator temperature, gas cooler exit temperature and heat rejection pressure in gas cooler.

$$COP_{system} = f(T_E, T_{GCE}, P_2)$$

COP of the transcritical system is maximum when it operates at optimum heat rejection pressure which depends upon evaporator temperature and gas cooler exit temperature.

$$(COP_{system})_{MAX} = f(T_E, T_{GCE}); \qquad (P_2)_{OPT} = f(T_E, T_{GCE})$$

D. Optimization of heat rejection pressure

Large data is generated for the transcritical cycle for evaporator temperature from –20 ^{0}C to 20 ^{0}C and the gas cooler exit temperature from 32 ^{0}C to 50 ^{0}C. This data is reduced to establish a correlation for optimum pressure in gas cooler. Contours for optimum pressure of heat rejection in gas cooler are shown in fig. 7. Optimum pressure varies from 8200 kPa to 12200 kPa. Optimum pressure increases from maximum evaporating temperature and minimum gas cooler exit temperature to the minimum evaporating temperature and maximum gas cooler exit temperature.

The regression is performed on the data to predict the correlation for optimum pressure of heat rejection in gas cooler. The correlation obtained is as follows:

$$P_{2,OPT} = 32.936 - 18.868T_E + 224.63T_{GCE}$$

E. Optimization of COP of system

Contours for maximum COP are shown in Fig. 8 at fixed compressor efficiency with evaporator temperature varying from –20 ^{0}C to 20 ^{0}C and the gas cooler exit temperature varying from 32 ^{0}C to 50 ^{0}C. Maximum COP varies from 5 to 29. Maximum COP increases from minimum evaporating temperature and maximum gas cooler exit temperature to the maximum evaporating temperature and minimum gas cooler exit temperature.

The regression is performed on the large generated data to predict the correlation for maximum COP. The correlation obtained is as follows:

$$\mathrm{COP_{MAX}} = 62.236 + 1.2307T_E + 0.0078T_E^2 - 2.1845T_{GCE} + 0.0213T_{GCE}^2 - .0221T_E.T_{GCE}$$

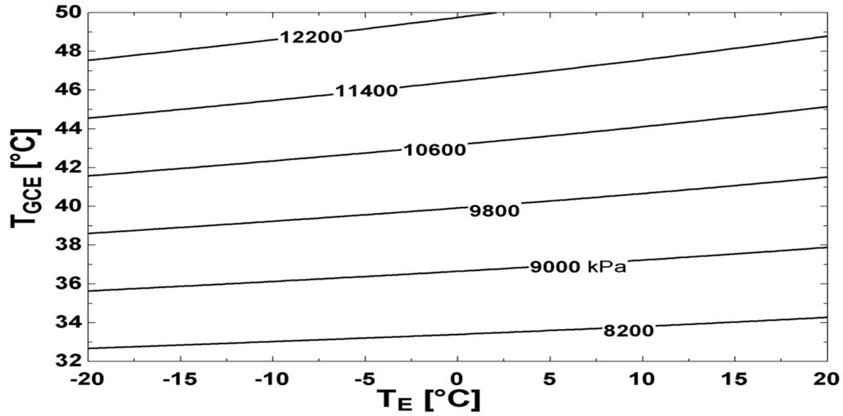


Fig. 7 Contours of optimum heat rejection pressure

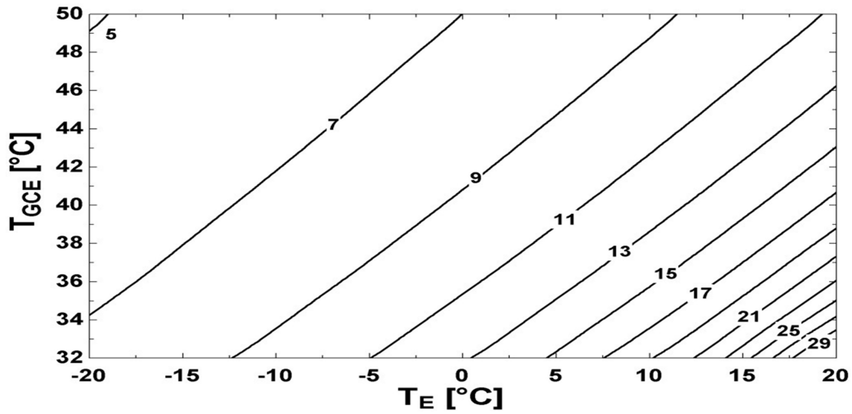


Fig. 8 Contours of maximum COP of transcritical cycle.

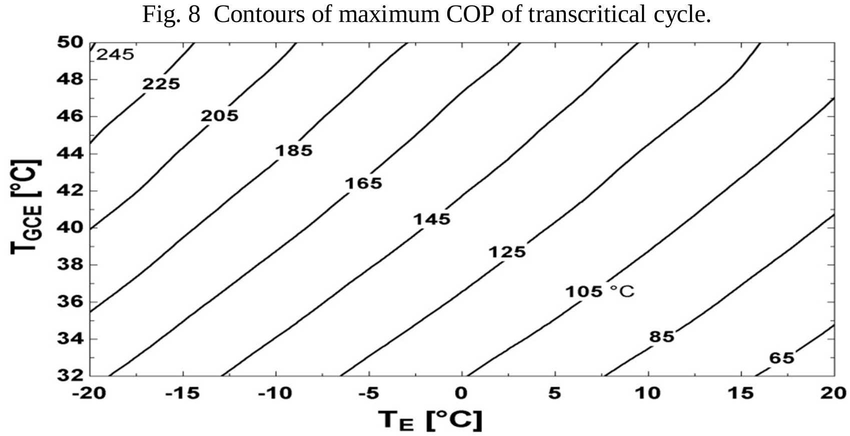


Fig. 9 Contours of optimum compressor discharge temperature

F. Optimization of compressor discharge temperature

Contours for optimum compressor discharge temperature of are shown in Fig. 9 at fixed compressor efficiency with evaporator temperature varying from –20 ^{0}C to 20 ^{0}C and the gas cooler exit temperature varying from 32 ^{0}C to 50 ^{0}C. The optimum compressor discharge temperature varies from 65 ^{0}C to 245 ^{0}C. The optimum compressor discharge temperature increases from maximum evaporating temperature and minimum gas cooler exit temperature to the minimum evaporating temperature and maximum gas cooler exit temperature.

The regression is performed on large generated data to predict the correlation for optimum compressor discharge temperature. The correlation obtained is as follows:

$$T_{2,OPT} = -12.6426 - 3.07601T_E + 38.1672T_{GCE}$$

Nomenclature:		
h	specific enthalpy	(kJ/ kg)
P	pressure	(kPa)
q	specific heat transfer	(kJ/ kg)
s	specific entropy	(kJ/kg- K)
T	temperature	(^{0}C)
w	specific work	(kJ/ kg)
ϵ	effectiveness	-
Greek		
η	efficiency	
Subscripts		
1–6	refrigerant state points	
C	compressor	
E	evaporator	
GC	gas cooler	
GCE	gas cooler exit	
MAX	maximum	
OPT	optimum	

V. CONCLUSIONS

Thermodynamic analysis presented in this paper leads to following conclusions:

A. COP of the system decreased with increase in gas cooler inlet temperature of external fluid. Hence gas cooler inlet temperature of external fluid should be low which depends on ambient conditions.

B. An increase in evaporator temperature resulted in increase in COP of the system.

C. Effectiveness of internal heat exchanger has no effect on COP of the transcritical system.

To optimize the COP, a regression analysis has been performed that could be useful to refrigeration engineers for setting optimum heat rejection pressure in gas cooler.

REFERENCES

[1] J.M. Calm, “The next generation of refrigerants – Historical review, considerations, and outlook”, International Journal of Refrigeration; 31 (2008) 1123–1133

[2] C.R. Taylor, “Carbon dioxide-based refrigerant system”, ASHRAE Journal; 44 (2002) 22- 27.

[3] S.M. Liaoa, T.S. Zhaoa, and A. Jakobsen, “A correlation of optimal heat rejection pressures in transcritical carbon dioxide cycles”, Applied Thermal Engineering; 20 (2000) 831–841.

[4] D.M. Robinson and E.A.Groll, “Efficiencies of transcritical CO_2 cycles with and without an expansion turbine”, International Journal of Refrigeration; 21 (1998) 577–589.

[5] J. Sarkar, S. Bhattacharyya, and M. Ramgopal, “Optimization of a transcritical CO_2 heat pump cycle for simultaneous cooling and heating applications” International Journal of Refrigeration; 27 (2004) 830–838.

[6] K. Srinivasan, P. Sheahen and C.S.P. Sarathy, “Optimum thermodynamic conditions for upper pressure limits of transcritical carbon dioxide refrigeration cycle”, International Journal of Refrigeration 33 (2010) 1395-1401

[7] S. Bhattacharyya, S. Bose and J. Sarkar, “Exergy maximization of cascade refrigeration cycles and its numerical verification for a transcritical CO_2-C_3H_8 system”, International Journal of Refrigeration 30 (2007) 624-632

[8] Y. Maa, Z. Liu and H. Tian, “A review of transcritical carbon dioxide heat pump and refrigeration cycles” Energy 55 (2013) 156-172

[9] E. Torrella, D. Sanchez, R. Llopis, and R. Cabello “Energetic evaluation of an internal heat exchanger in a CO_2 transcritical refrigeration plant using experimental data” International Journal of Refrigeration 34 (2011) 40-49

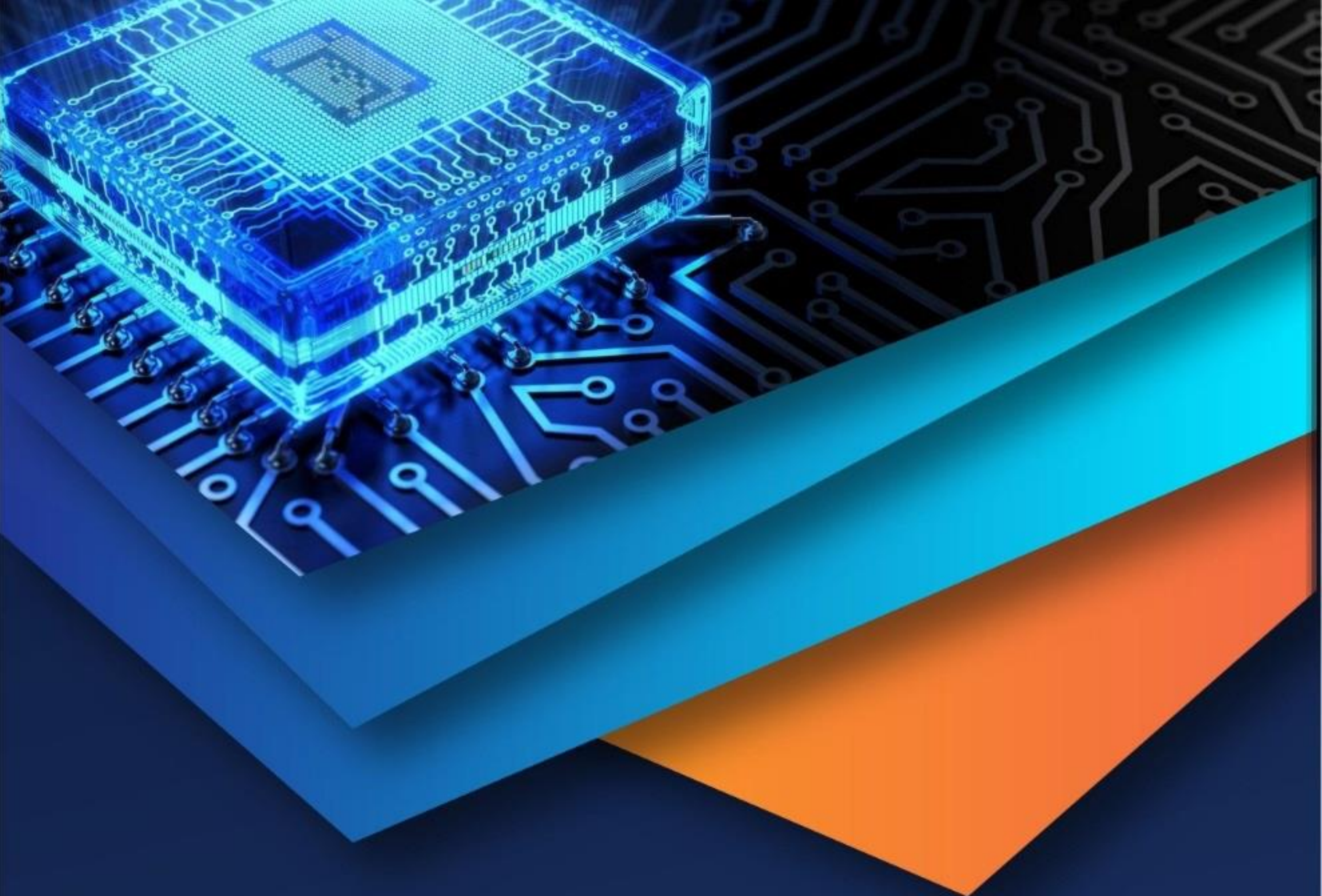
[10] V. Sharma, B. Fricke and P. Bansal, “Comparative analysis of various CO_2 configurations in supermarket refrigeration systems” International Journal of Refrigeration 46 (2014) 86-99

[11] S. Bhattacharyya, A. Garai, and J. Sarkar, “Thermodynamic analysis and optimization of a novel N_2O–CO_2 cascade system for refrigeration and heating”, International Journal of Refrigeration; 32 (2009) 1077–1084.

[12] EES: Engineering Equation Solver, 2014. fChart Software Inc.

[13] C.P. Arora, 2012. Refrigeration and Air Conditioning, 3rd ed. New Delhi: Tata McGraw Hill.

[14] A. Messineo, “R744-R717 Cascade Refrigeration System: Performance Evaluation compared with a HFC Two-Stage System”, Energy Procedia 14 (2012) 56 – 65



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