International Journal of Scientific and Technical Advancements

ISSN: 2454-1532

Thermodynamic Analysis of Trans-Critical CO_2 Refrigeration Cycle in Indian Context

Nilesh Purohit[1], Dileep Kumar Gupta[2], M.S. Dasgupta[3]
[1]Department of Mechanical Engineering, BITS-Pilani, Rajasthan, India
[2]Departmentof Mechanical Engineering, IITRAM, Gujarat, India
Email address: [1]purohitnilesh89@gmail.com

***Abstract*—**The coefficient of performance for CO_2 based trans-critical system is lower at high ambient temperature regions like India due to higher energy required for compression to a high temperature beyond critical point.This paper presents thermodynamic analysis of a basic trans-critical CO_2 refrigeration cycle. Further, the basic cycle is modified for two different configurations incorporating parallel compression and inter-cooling. Use of parallel compression and inter-cooling in cycle configuration are two of the most promising cycle modifications for improving the performance of trans-critical CO_2 refrigeration systems operating at high ambient temperatures. Simulation results show that parallel compression configuration is more effective. The maximum improvement in COP obtained is about 25% for parallel compression configuration. Also, the operating pressure of gas cooler is found lower for parallel compression configuration.

***Keywords*—** Trans-critical CO_2 refrigeration cycle; thermodynamic analysis; parallel compression; inter cooling.

I. INTRODUCTION

In modern life, demand for comfort guides us to build refrigeration and air conditioning system, and with improvement in quality of life, the demand is ever increasing. There is however grave concern about adverse environmental foot print of conventional refrigerants including CFCs, HCFCs and HFCs. The most talked about harmful effects includes very high ozone depletion potential (ODP) and global warming potential (GWP) of some of these synthetic refrigerants. On the other hand, natural refrigerants such as air, water, ammonia, carbon dioxide, isobutene, propane etc. are ecologically safe, have zero ODP and low GWP and are, therefore, gaining importance[1]. Among these, CO_2 is one of the preferred choices owing to its high specific heat, non toxicity, non flammability, eco friendliness and low cost.

From the engineering perspective, CO_2 used as a refrigerant in vapour compression cycle has number of advantages such as lower compression ratio, high volumetric cooling capacity, compatibility with normal lubricants and common machine construction materials and well defined thermo-physical properties[2]. CO_2 has a rather old history as refrigerant, widely used during (1930s), however was abandoned later due to the invention of the synthetic refrigerants which were more effective. Subsequently, it has been revived by Norwegian Professor Gustav Lorentzenin 1993 through his work on trans-critical automotive air conditioning (AAC) system[3]. Based on the same, studies on CO_2 as a refrigerant in trans-critical cycle gained considerable attention[4].

A CO_2 refrigeration system needs to operate above critical point due to its low critical temperature (31.2^{o}C), which requires replacement of conventional condenser with a gas cooler due to the existence of single phase above critical point. The coefficient of performance for CO_2 based trans-critical system is low. It is even lesser when operated at higher ambient temperature regions due to the requirement of high pressure and temperature to which the gas needed to be compressed for effective heat rejection. Gupta and Das Gupta [5] reported that for tropical or Indian climatic conditions, where the ambient temperature is higher, the performance of gas cooler is also highly sensitive to operating temperature and pressure as shown in Fig. 1. The system operates conventionally before the critical point where the sensitivity is low but after the critical point the COP decreases as well as its sensitivity to parameter variation increases. Therefore, pertinent need is felt for modification of the basic trans-critical cycle to improve its efficiency

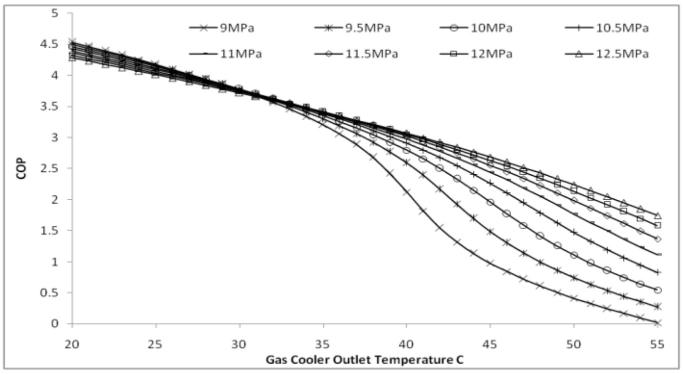

Fig. 1. Variation of COP with gas cooler outlet temperature (Gupta & Dasgupta, 2010).

Various cycle modifications using internal heat exchanger, multi staging, work recovery expansion device and ejector expansion device are reported in open literature [6]. The performance of basic trans-critical CO_2 refrigeration system has been shown to improve using a parallel compression, especially for hot climatic conditions[7], [8]. In this technique, the refrigerant vapor is compressed to gas cooler pressure in two separate non-mixing streams one coming from an intermediate receiver and the other coming from the evaporator. This implies that part of compressor is used to

Nilesh Purohit, Dileep Kumar Gupta and M. S. Dasgupta, "Thermodynamic analysis of trans-critical CO_2 refrigeration cycle in Indian context", *Journal of Scientific and Technical Advancements*, Volume 1, Issue 3, pp. 143-146, 2015.

compress vapor from an intermediate receiver. Groll and Kim [9] proposed improved inter-cooling between two stage trans-critical CO_2 system where the intermediate pressure is kept above critical point. In this paper, performance comparison is carried out for cycles utilizing parallel compression and improved inter-cooling cycle configurations for Indian conditions.

II. Cycle Configurations

A basic vapour compression trans-critical CO_2 cycle and its p-h diagram is shown in Fig. 2. The cycle is partly subcritical (low pressure side) and partly supercritical (high pressure side). Evaporation (4-1) takes place at sub critical pressure similar to conventional vapour compression refrigeration cycles but heat rejection takes place at supercritical pressure (2-3) due to low critical temperature of CO_2. In supercritical heat rejection no saturation point exists, so the gas cooler pressure is independent of the refrigerant temperature at the gas cooler exist (state 3). Pettersen et al. [10] reported that the gas cooler pressure is an influential parameter due to the s-shape of the isotherm in supercritical region. Since the throttling valve inlet condition determines the specific refrigeration effect, it is necessary to control the high side pressure for the optimal operations. Further there is additional challenge for tropical regions to operate trans-critical CO_2 based systems under tight control of pressure to obtain reasonable heat rejection in the gas cooler.

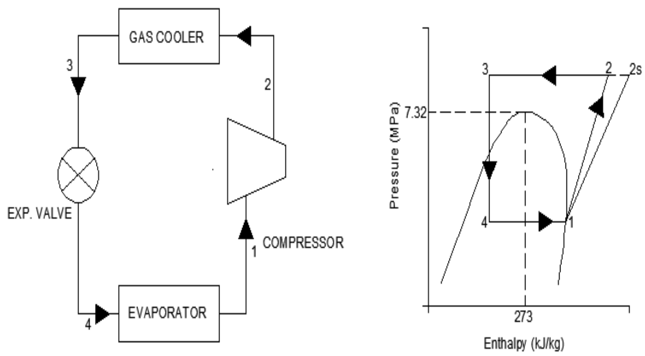

Fig. 2. Basic trans-critical CO_2 refrigeration cycle.

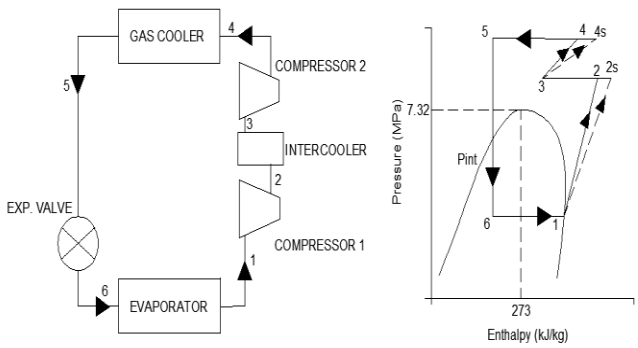

Fig. 3. Trans-critical CO_2 refrigeration cycle with inter-cooler.

Fig. 3 shows the incorporation of inter-cooling between two-stage compressorsin basic trans-critical CO_2 refrigeration cycle. The refrigerant is cooled from state 2 to state 3 using ambient air which in turn reduces the specific volume of the refrigerant.

Fig. 4 shows incorporation of parallel compression in the basic cycle. The refrigerant mist after expansion (state 5) separates as vapor from the receiver (state 8). This separated vapor is subsequently recompressed again to the gas cooler pressure using the main compressor itself (state 9). Thereafter, it is mixes with the compressed vapor coming out from the evaporators via the compressor (state 2). The mixed vapor enters the gas cooler at state 3 and the cycle continues.

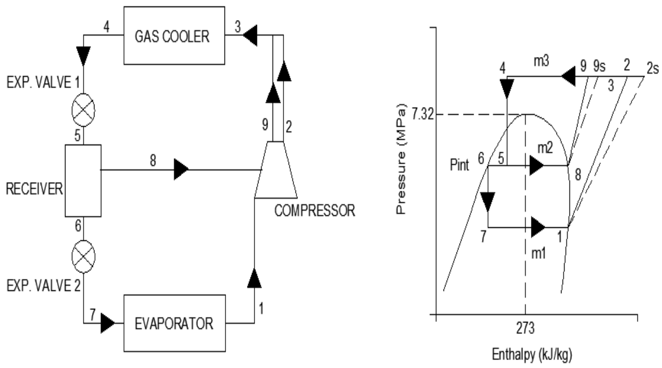

Fig. 4. Trans-critical CO_2 refrigeration cycle with parallel compression.

III. Thermodynamic Modelling

Steady flow energy and mass balance equations in the devices areas shown in Table 1. The sameare employed to simulate CO_2 refrigeration cycle configurations at high ambient temperature (typical Indian conditions). The basic cycle and its two modifications are analyzed to identify their effect on COP at high ambient temperature. In inter-cooling and parallel compression trans-critical CO_2 refrigeration cycle, the choice of intermediate pressure is vital in improving COP. Hence, the intermediate pressure is optimized simultaneously along with the gas cooler pressure. In order to simplify the computational modeling, following assumptions are made:

- Heat loss to the surroundings is negligible.
- Single phase heat transfer occurs with the external fluid in inter-cooling cycle.
- Evaporation, gas cooling and inter-cooling processes are isobaric.
- Compressor isentropic efficiency is taken as 70%
- Heat exchanger effectiveness is taken as 80%.

Based on the equations, models are constructed for the two modifications of trans-critical CO_2 refrigeration cycle in MATLAB. The program takes refrigerant data from REFPROP (9.0) automatically during computation. The approach temperature defined as difference between ambient air temperature and gas cooler outlet temperature is taken as zero.The simulations are so designed as to extract the maximum COP and corresponding operating parameters (discharge pressure and inter-stage pressure) for complete range of gas cooler outlet temperature and pressure.

Nilesh Purohit, Dileep Kumar Gupta and M. S. Dasgupta, “Thermodynamic analysis of trans-critical CO_2 refrigeration cycle in Indian context”, *Journal of Scientific and Technical Advancements*, Volume 1, Issue 3, pp. 143-146, 2015.

International Journal of Scientific and Technical Advancements

ISSN: 2454-1532

Table 1. Model equations.

Component	Cycle	Model Equation
Evaporator	Basic	$RE = m(h1 - h4)$
	Inter-cooling	$RE = m(h1 - h6)$
	Parallel Compression	$RE = m1(h1 - h7)$
Compressor	Basic	$W = m(h2 - h1)$
	Inter-cooling	$W1 = m(h2 - h1)$;1st stage
		$W2 = m(h4 - h3)$;2nd stage
	Parallel Compression	$W1 = m1(h2 - h2)$;1st stage
		$W1 = m2(h9 - h8)$;2nd stage
Inter- Cooler	Inter-cooling	$e = (T2 - T3)/(T2 - Ta)$
Gas Cooler	Basic	$Q = m(h2 - h3)$
	Inter-cooling	$Q = m(h4 - h5)$
	Parallel compression	$Q = m3(h3 - h4)$
Expansion Valve	Basic	$h3 = h4$
	Inter-cooling	$h5 = h6$
	Parallel compression	$Exp1; h4 = h5$
		$Exp2; h6 = h7$
Receiver	Parallel Compression	$m3h5 = m1h6 + m2h8$

RE=Refrigeration effect (KW), W=Work required by compressor (KW), e=Effectiveness of inter cooler; Q= Gas cooler capacity (KW); Exp=Expansion device; m=mass flow rate (kgs^{-1}); Ta=Air temperature (K)

IV. Results and Discussion

The range and values of the variables and constants used in the simulation based study is listed in Table 2.

Table 2. Range of parameters.

Parameter	Range/Steps
Evaporator temperature (K)	263-283 /10
Gas cooler outlet temperature (K)	313-323/2
Gas cooler pressure (MPa)	9-18/0.5
Inter-mediate pressure (MPa)	[a]4.5-6.5/0.1
	[b]6.5-12.5/0.5

a=Parallel compression cycle, b= Inter-cooling cycle

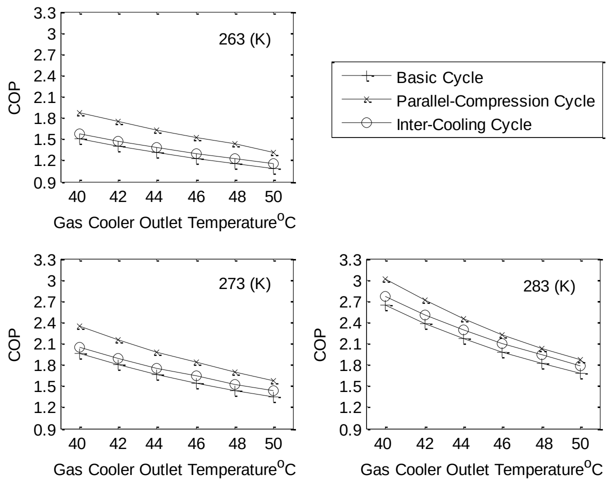

Fig. 5. Variation of COP with gas cooler outlet temperature.

Fig. 5 shows the comparative performance of tested configurations for trans-critical CO2 refrigeration cycle for various gas cooler outlet temperatures. The performance of the cycle improves with increase in evaporator temperature for all investigated cycles. Parallel compression configuration gives better performance for all gas cooler outlet and evaporator temperature. Fig. 6 shows the discharge pressure of compressor for various gas cooler outlet temperatures corresponding to maximum COP. Parallel compression cycle has lower gas cooler operating pressure. Inter-cooling cycle depicts higher pressure for all investigated cases.

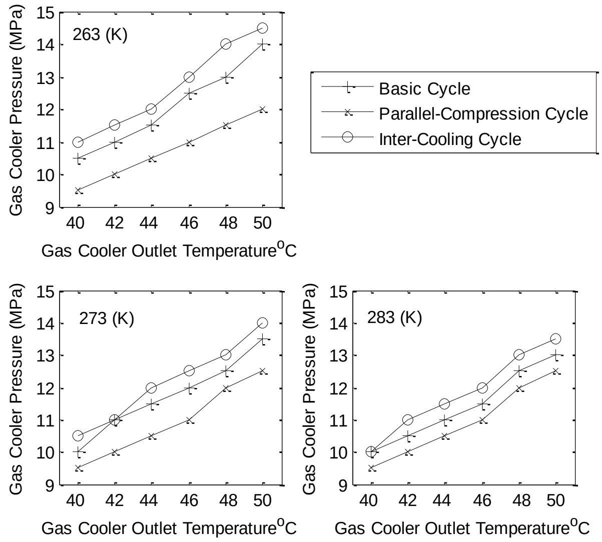

Fig. 6. Variation of discharge pressure with gas cooler outlet temperature.

Percentage improvement in COP for parallel compression cycle is shown in Fig. 7. Result shows that with increase in evaporator temperature the percentage improvement in COP decreases. Maximum improvement in COP is about 25% for 263 K evaporator temperature at lower gas cooler outlet temperature. Also, with increase in gas cooler outlet temperature the improvement in COP decreases for all evaporator temperature.

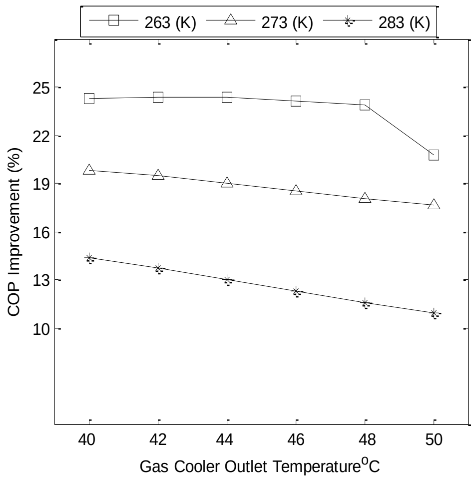

Fig. 7. Percentage improvement in COP for parallel compression cycle.

Nilesh Purohit, Dileep Kumar Gupta and M. S. Dasgupta, "Thermodynamic analysis of trans-critical CO_2 refrigeration cycle in Indian context", *Journal of Scientific and Technical Advancements*, Volume 1, Issue 3, pp. 143-146, 2015.

Fig. 8 shows the percentage reduction in gas cooler operating pressure for parallel compression cycle. As the evaporator temperature increases the reduction in pressure decreases. Further, with increase in gas cooler outlet temperature the reduction in operating pressure decreases.

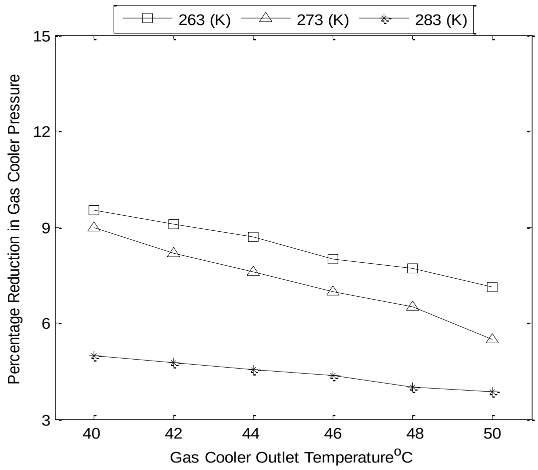

Fig. 8. Percentage reduction in gas cooler pressure for parallel compression cycle.

V. Conclusion

In this article, two modifications incorporating parallel compression and inter-cooler for the trans-critical CO_2 refrigeration cycle are thermodynamically analysed for typical Indian conditions. Further, the performance is compared to that of basic cycle. The conclusions made from the study are as follows:

- Parallel compression cycle configuration showed better performance for all investigated range of parameters.
- Inter-cooling cycle performance was comparable to the parallel cycle at higher ambient temperatures.
- The improvement in COP for parallel compression cycle decreases with increase in evaporator and gas cooler outlet temperature. Further, maximum improvement in COP is nearly 25% for low evaporator temperature.
- Parallel compression cycle shows lower gas cooler operating pressure for all investigated cases. The percentage reduction in operating pressure decreases with increase in evaporator and gas cooler outlet temperature.
- The IC multi-stage system is found to have a maximum operating pressure; this may be improved upon using more effective inter-cooler fluid than air.

Acknowledgement

Author will like to acknowledge partial financial support received from the host organization, BITS Pilani, India & Government of India under research grant No: DST/TSG/NTS/2012/19-G.

Nomenclature

p	Pressure, MPa
h	Specific enthalpy, kJ kg-1
Pint	Interstage pressure, MPa
W	Compressor work, kW
m	Mass flow rate, kg s-1
RE	Refrigeration effect, kW
Ta	Air Temperature, K
Acronyms	
CO2	Carbon dioxide
IC	Inter-cooler
COP	Coefficient of performance

References

[1] A. Pearson, "Carbon dioxide—new uses for an old refrigerant," *International Journal of Refrigeration,* vol. 28, pp. 1140-1148, 2005.

[2] M.-H. Kim, J. Pettersen, and C. W. Bullard, "Fundamental process and system design issues in CO 2 vapor compression systems," *Progress in energy and combustion science,* vol. 30, pp. 119-174, 2004.

[3] G. Lorentzen, "Revival of carbon dioxide as a refrigerant," *International Journal of Refrigeration,* vol. 17, pp. 292-301, 1994.

[4] Y. Ma, Z. Liu, and H. Tian, "A review of transcritical carbon dioxide heat pump and refrigeration cycles," *Energy,* vol. 55, pp. 156-172, 2013.

[5] D. K. Gupta and M. Dasgupta, "Gas cooler design issues for trans-critical carbon dioxide refrigeration system in Indian context," in *Proc. of the 3rd International Conference on Advances in Mechanical Engineering, SVNT, Surat, India*, 2010, pp. 229-233.

[6] J. Sarkar, "Review on cycle modifications of transcritical CO2 refrigeration and heat pump systems," *Journal of Advanced Research in Mechanical Engineering,* vol. 1, pp. 22-29, 2010.

[7] J. Sarkar and N. Agrawal, "Performance optimization of transcritical CO 2 cycle with parallel compression economization," *International Journal of Thermal Sciences,* vol. 49, pp. 838-843, 2010.

[8] I. Bell, "Performance increase of carbon dioxide refrigeration cycle with the addition of parallel compression economization," *in 6th IIR Gustav Lorenzen Natural Working Fluids,* 2004.

[9] E. A. Groll and J.-H. Kim, "Review article: review of recent advances toward transcritical CO2 cycle technology," *Hvac&R Research,* vol. 13, pp. 499-520, 2007.

[10] J. Pettersen, A. Hafner, G. Skaugen, and H. Rekstad, "Development of compact heat exchangers for CO 2 air-conditioning systems," *International Journal of Refrigeration*, vol. 21, pp. 180-193, 1998.

Nilesh Purohit, Dileep Kumar Gupta and M. S. Dasgupta, "Thermodynamic analysis of trans-critical CO_2 refrigeration cycle in Indian context", *Journal of Scientific and Technical Advancements*, Volume 1, Issue 3, pp. 143-146, 2015.