CO_2 Refrigeration System for High Ambient Temperatures

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Abstract

In developing countries like India, the introduction of natural refrigerants for Heating, Ventilation, Air Conditioning and Refrigeration (HVAC&R) will have enhanced impacts considering the current environmental protocols. Carbon dioxide (CO_2) appeared as the most attractive and promising aspirant as a refrigerant because of its low global warming potential (GWP) and zero ozone depletion potential (ODP). But its critical temperature is low at 31.1°C and hence it has to operate in trans-critical mode at high ambient temperatures. However, the system efficiency can be improved with the adoption of ejector technology, which can recover work energy from the expanding high pressure gas. The internal heat exchanger (IHX) also contributes to enhance the efficiency. In addition, CO_2 refrigeration system is attractive for applications with multi-evaporator temperatures. Such an ejector based, multi-evaporator CO_2 trans-critical refrigerant system is experimentally investigated with and without IHX to explore its suitability for applications with simultaneous air conditioning, refrigeration and freezing requirements for the typical Indian temperature conditions of 36 to 46°C. It is observed that the maximum enhancement in coefficient of performance (COP), exergy efficiency and power input ratio (PIR) of the CO_2 system with IHX are 15.18%, 15.38% and 19.15% respectively.

Keywords

_CO_2; IHX; Multi-ejector; Exergy; PIR_

Introduction

Popularity of the cooling system working with natural working refrigerants such as CO_2 for supermarkets is tremendously increasing in European countries and the present scenario is to adopt and extend it to developing countries like India (Pearson, 2005). The awareness of using CO_2 technology is increasing due to the harmful effect (global warming) of synthetic refrigerants (Calm, 2002). However, adoption of the CO_2 cooling technology in developing countries is relatively slow due to the unique and challenging behavior of CO_2 refrigerant properties at high ambient temperature (Lorentzen, 1994). The system needs to be operated in the trans-critical mode when the sink is above 31.1°C. Thus, its performance is relatively poor for a significant part of system operation in an year in India due to the high exergy losses in the system (Singh and Dasgupta, 2017). Various modifications are tried to improve the performance of the simple vapor compression trans-critical cycle. Multi-stage, work recovery expanders, two-phase ejectors, system cascading, internal heat exchanger and sub cooling are a few examples (Singh and Dasgupta, 2016). It was reported that the ejector is a potential solution

Table1: Studies of trans-critical CO_2 system configuration with IHX

Application	Study Focus	Improvement/Rem arks	Authors
Water heating	Effect of length of tube of IHX on overall performance	COP of system improves for a certain discharge pressure	(Goo et al., 2005)
Air conditioning	Variable speed of cycle with EEV opening	System reduced optimum compressor discharge pressure from 9.2 to 8.7 MPa	(Cho et al., 2007)
Air conditioning	Effect of with/without IHX on overall performance	Using the internal heat exchanger, COP is 10% better	(Aprea and Maiorino, 2008)
Air conditioning	Effect of with/ without IHX on overall performance	An increment of efficiency of plant up to 12%	(Rigola et al., 2010)
Air conditioning	Effect of length of tube of IHX on overall performance	IHX provided the maximum COP improvement of up to 27%	(Nakagawa et al., 2011)
Freezing	Effect of with/ without IHX on overall performance	An internal heat exchanger enhances exergy efficiency	(Shariatzadeh et al., 2016)
Air conditioning	Temperature and mass flow at inlet boundary of both hot and cold streams.	High inlet temperature allows internal heat exchanger to be more efficient	(Ituna-yudonago et al., 2017)
Refrigeration	Optimal control of gas cooler pressure with IHX by developing a control strategy	If capacity and appropriate overall heat transfer coefficient for gas cooler are provided, the introduced analysis concept can be used as a control method with quite a good accuracy	(Se et al., 2017)

for high ambient conditions (Banasiak et al., 2012; Banasiak and Hafner, 2013). Internal heat exchanger also appeared as a potential candidate to improve the performance of the CO_2 trans-critical system for various applications at high ambient temperature, and a few similar experimental studies are tabulated in *Table 1*.

Study Objective

The objective of the present experimental study is to evaluate the performance of the CO_2 trans-critical cooling system for high ambient temperatures, that are typically prevailing in India. The experimental study is focused on improving the overall performance of the system with the use of ejectors and internal heat exchanger (IHX). The overall performance is projected in terms of COP, exergy efficiency and PIR.

CO_2 Test Rig Description

A fully instrumented CO_2 cooling test facility is developed and installed at Indian Institute of Technology Madras as shown in *Figure 1*. It has a total cooling capacity of 33 kW and is designed for supermarket application to maintain three different evaporation temperature levels: +6°C (20 kW) for air conditioning, -6°C (10 kW) for refrigeration and -29°C (3 kW) for freezing. The test facility is equipped with a heat recovery system too at high temperature with a glycol solution loop to be able to supply heat towards the evaporators.

Figure 1: CO_2 cooling test facility

Two glycol circuits are arranged with different glycol concentrations. Load of medium temperature (*MT*) and low temperature (*LT*) evaporators on secondary loop are controlled by manually controlled electronic expansion valves (*EEVs*) installed upstream to the evaporators. Their temperatures are controlled by compressors. The temperature level of the AC evaporator is controlled by the receiver pressure. Apart from LT and MT compressors, an additional auxiliary (*AUX*) compressor is installed to handle high amounts of flash gas from the receiver. This operates in parallel to MT compressor; thus the term 'parallel compression'.

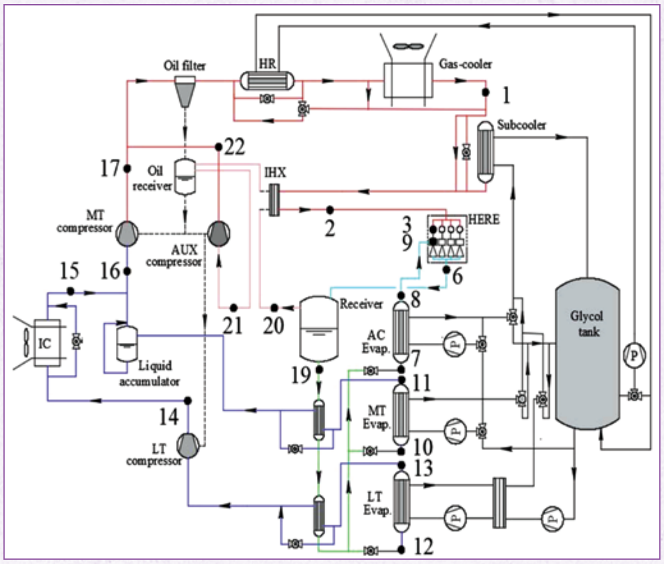

Figure 2: Schematic of the CO_2 cooling test rig

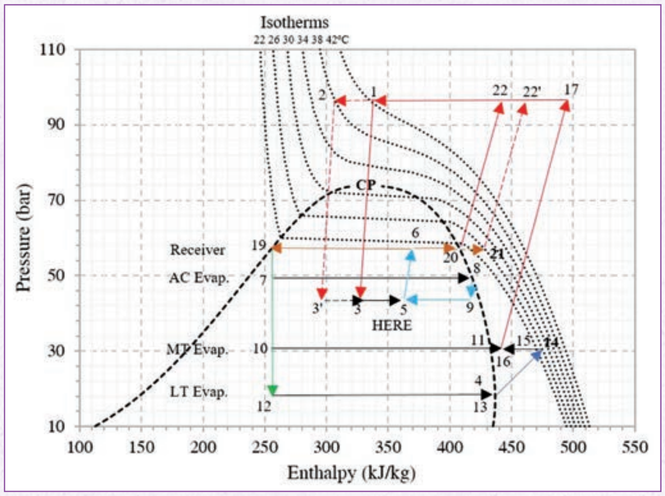

Figure 3: Ph plot of the trans-critical CO_2 cooling cycle

Table 2: Details of CO_2 refrigeration system

86.95 mm	Details	Units	Value /Type
Gas cooler	Rated capacity	kW	54
	Cooling medium		Air
	Heat exchanger type		Fin-tube
	Pipe material		Copper
	Fin material		Aluminum
	Fin thickness	mm	0.3
	Fin spacing	mm	2.1
	Coil rows		4
Compressor	Design		Reciprocating
	No of cylinders		2
	Bore/stroke	mm	42/37
	Displacement	$m^3 h^{-1}$	10.70
	Cooling oil		PAG, Daphere hermetic oil PR
Evaporator	Rated capacity AC	kW	20
	Rated capacity MT	kW	10
	Rated capacity LT Design	kW	3 Tube-in-tube
	Tube material		Copper

Two ejectors are installed: one with low ejection ratio and a second unit with high ejection ratio. One liquid suction accumulator is also installed to enable a secure separation of liquid and vapor upstream of the compressors to allow a liquid over-feed operation of the evaporators throughout the year. Temperature and pressure sensors and energy meters are installed at various locations to measure and evaluate the performance of the system and examine the variation of various parameters.

The system facilitates both manual and automatic optimization of the gas cooler pressure level with respect to the gas cooler outlet temperature. The facility also has a manually operated controller for the rpm of the gas cooler fans to enable various airflow rates to adjust the refrigerant exit temperature. The schematic of the CO_2 cooling system with various installed components is projected in *Figure 2* and the details of the same are tabulated in *Table 2*. Ph plot for the CO_2 cooling cycle is shown in *Figure 3*.

Operational Mode: CO_2 Refrigeration System for Supermarket

The CO_2 trans-critical refrigeration test rig is designed to operate in different modes. The present study is aimed at exploring the suitability of CO_2 refrigeration for supermarket applications. Thus, the operational mode consists of three evaporators (*AC, MT* and *LT*), three compressors (*AUX, MT* and *LT*) and the low ejection ratio ejector (*LERE*). This operational mode simulates an actual supermarket operation for handling all air conditioning, refrigeration and freezing evaporation loads. The operating parameters of the CO_2 supermarket cooling system during the experimental performance evaluation are tabulated in *Table 3*.

Table 3: Parameters used for performance evaluation

Operating Parameter	Units	Value/Range
Gas cooler outlet temperature	°C	36, 41, 46
Gas cooler outlet pressure	bar	80 to120
Air conditioning evaporator temperature	°C	+6
Refrigeration evaporator temperature	°C	–6
Freezing evaporator temperature	°C	–29
Receiver pressure	bar	46

Experimental Data Reduction

The CO_2 cooling test rig is designed with the latest operative system. The service tool installed in the rig has both CO_2 and glycol side controls of the various components highlighted in *Figure 2*. Before switching ON the test rig, the CO_2 side parameters such as evaporator temperature, receiver pressure, gas cooler outlet temperature, gas cooler pressure, etc. are fixed in the service tool and then the system is manually switched ON from the main controller. Steady state of the system is considered with the constant evaporator load, evaporator temperature, gas cooler outlet temperature, receiver pressure and compressor frequency. The system performance is evaluated for each gas cooler outlet temperature after achieving the steady state which typically requires ~45 minutes. Hourly based data extracted for a required single gas cooler outlet temperature and averaged value of the various parameters is further used for the performance evaluation. The details of the measuring instruments used and equipped in the test rig and their uncertainty are tabulated in *Table 4*.

Table 4: Details of the instruments used in the test rig

Sensor(s)	Details	Units	Symbol	Accuracy	Uncertainty
Mass flow	Rheonic RHE08 RHM04	kg min^{-1}	m	+/– 0.20%	0.002
Temperature	Omega TMTSS-020U-6-196"	°C	T	+/– 0.075%	0.13
Pressure	Endress and Hauser Deltabar S PMD75	MPa	p	+/– 0.075%	0.003

Performance Evaluation

Performance of the multi-ejector based CO_2 cooling system for supermarket application at high ambient temperature is computed using the following equations:

- COP of the CO_2 cooling system is computed by

$$COP = \frac{Q_{AC}+Q_{MT}+Q_{LT}}{P_{AUX,}+P_{MT}+P_{LT}} \quad (1)$$

Where, load on the three evaporators AC, MT and LT of the system are computed from the glycol side by

$$\dot{Q}_{AC} = \dot{m}_{AC} * c_p * (T_{AC\_outlet} - T_{AC\_inlet}) \quad (2)$$

$$\dot{Q}_{MT} = \dot{m}_{MT} * c_p * (T_{MT\_outlet} - T_{MT\_inlet}) \quad (3)$$

$$\dot{Q}_{LT} = \dot{m}_{LT} * c_p * \left(T_{LT_{outlet}} - T_{LT_{inlet}}\right) \quad (4)$$

- Exergy efficiency of the CO_2 cooling system is computed by

$$\eta_{ex} = \frac{B_{AC} + B_{MT} + B_{LT}}{P_{AUX,} + P_{MT} + P_{LT}} \quad (5)$$

where, BAC, BMT and BLT are exergies of the three evaporators AC, MT and LT, respectively, and are computed by

$$B_{AC} = \dot{Q}_{AC} * \left(\frac{T_{ambient}}{T_{UT\_AC}} - 1\right) \quad (6)$$

$$B_{MT} = \dot{Q}_{MT} * \left(\frac{T_{ambient}}{T_{UT\_MT}} - 1\right) \quad (7)$$

$$B_{LT} = \dot{Q}_{LT} * \left(\frac{T_{ambient}}{T_{UT\_LT}} - 1\right) \quad (8)$$

The utility temperatures (T_{UT}) of the three evaporators AC, MT and LT are 22°C, 4°C and -18°C, respectively.

- Power Input Ratio (PIR) of the CO_2 cooling system is computed by

$$PIR = \frac{P_{AUX} + P_{MT} + P_{LT}}{P_{AUX,Carnot} + P_{MT,Carnot} + P_{LT,Carnot}} \quad (9)$$

where, Carnot power consumptions of the three compressors AUX, MT and LT are computed by

$$P_{AUX,Carnot} = \frac{\dot{Q}_{AC}}{T_{UT\_AC}} * (T_{ambient} - T_{UT\_AC}) \quad (10)$$

$$P_{MT,Carnot} = \frac{\dot{Q}_{MT}}{T_{UT\_MT}} * (T_{ambient} - T_{UT\_MT}) \quad (11)$$

$$P_{LT,Carnot} = \frac{\dot{Q}_{LT}}{T_{UT\_LT}} * (T_{ambient} - T_{UT\_LT}) \quad (12)$$

Results and Discussion

Coefficient of Performance (COP)

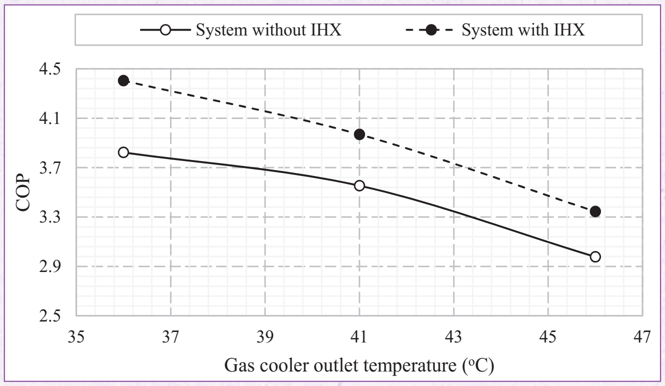

Figure 4: System COP w.r.t. gas cooler outlet temperature for with and without IHX

Figure 4 shows the variation of COP with respect to gas cooler outlet temperature for the ejector based CO_2 trans-critical refrigeration system with and without IHX. The gas cooler outlet temperatures were set to map the high ambient temperature conditions. There exists an optimum pressure at each gas cooler outlet temperature maximizing the COP of the CO_2 system and it is identified by following the S-shape isotherm of the CO_2 refrigerant (Lorentzen, 1994). In the present experimental set-up, an automatic control system maintains the optimum pressure at each set gas cooler outlet temperature. Further, it is observed that this optimum COP of the system was improved with the support of IHX in the system configuration. It can be ascribed to the fact that the system optimum pressure reduces with the use of IHX in the system configuration, which further results in the reduction of total power consumption by the compressor in the system. The maximum COP of the system observed with IHX is 4.4, which is 15.18% more than without IHX in system configuration at high ambient temperature.

Exergy Efficiency

Figure 5 shows the variation of exergy efficiency with gas cooler outlet temperature of the CO_2 system with and without IHX. The system with IHX configuration performs better than the system without IHX configuration as projected. It is observed that as the gas cooler temperature increases, the exergy efficiency of the system also increases. The maximum exergy efficiency realized is 0.33 for the system with IHX, which is 15.3% more than without IHX at 36°C gas cooler outlet temperature.

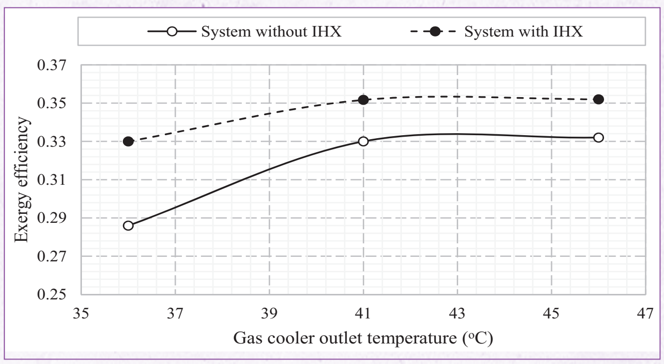

Figure 5: Exergy efficiency w.r.t. gas cooler outlet temperature for with and without IHX

Power Input Ratio (PIR)

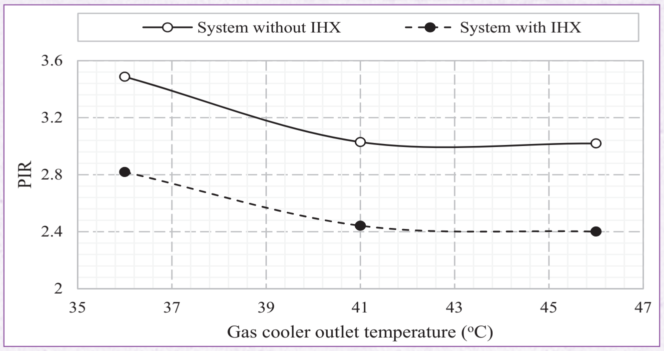

Figure 6: System PIR w.r.t. gas cooler outlet temperature for with and without IHX

Figure 6 shows the variation of power input ratio (PIR) with gas cooler outlet temperature of the system with and without IHX. It is observed that as the gas cooler temperature increases, the power input ratio of the system decreases. This is ascribed to the reduction in compressor power consumption of the system with IHX, which is also projected in terms of COP. The maximum PIR observed for the system with IHX is 2.82, which is 19.15% less than without IHX in system configuration.

Conclusions

A multi-temperature CO_2 trans-critical refrigeration system is experimentally studied for its suitability for high ambient temperature conditions typical of India with add-on components like ejector and internal heat exchanger (IHX) to improve system performance. It is found that IHX significantly improves the performance in terms of COP, exergy efficiency and power input ratio (PIR). The values of these parameters obtained are 4.4, 0.3 and 2.82 respectively for the system with IHX which are better by 15.18, 15.3 and 19.15% respectively than without IHX.

Nomenclature

AC	Air conditioning
AUX	Auxiliary compressor
B	Exergy (kW)
COP	Coefficient of Performance
CO_2	Carbon dioxide
c_p	Specific heat (kW kg^{-1} K^{-1})
GWP	Global warming potential
IHX	Internal heat exchanger
LERE	Low ejection ratio ejector
LT	Low temperature
$\dot{m}$	Mass flow rate (kg s^{-1})
MT	Medium temperature
ODP	Ozone depletion potential
P	Power consumption (kW)
PIR	Power input ratio
RP	Receiver pressure
$\dot{Q}$	Heat transfer rate (kW)
UT	Utility Temperature

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References

Aprea, C., Maiorino, A., 2008. An experimental evaluation of the transcritical CO_2 refrigerator performances using an internal heat exchanger. Int J Refrig 31, 1006–1011. doi:10.1016/j.ijrefrig.2007.12.016

Banasiak, K., Hafner, A., 2013. Mathematical modelling of supersonic two-phase R744 flows through converging-diverging nozzles: The effects of phase transition models. Appl Therm Eng 51, 635–643. doi:10.1016/j.applthermaleng.2012.10.005

Banasiak, K., Hafner, A., Andresen, T., 2012. Experimental and numerical investigation of the influence of the two-phase ejector geometry on the performance of the R744 heat pump. Int J Refrig 35, 1617–1625. doi:10.1016/j.ijrefrig.2012.04.012

Calm, J.M., 2002. Emissions and environmental impacts from air-conditioning and refrigeration systems. Int J Refrig 25, 293–305.

Cho, H., Ryu, C., Kim, Y., 2007. Cooling performance of a variable speed CO_2 cycle with an electronic expansion valve and internal heat exchanger. Int J Refrig 30, 664–671. doi:10.1016/j.ijrefrig.2006.10.004

Goo, S., Jo, Y., Lee, G., Soo, M., 2005. The performance of a transcritical CO_2 cycle with an internal heat exchanger for hot water heating. Int J Refrig 28, 1064–1072. doi:10.1016/j.ijrefrig.2005.03.004

Ituna-yudonago, J.F., Belman-flores, J.M., Elizalde-blancas, F., García-valladares, O., 2017. Numerical investigation of CO_2 behavior in the internal heat exchanger under variable boundary conditions of the transcritical refrigeration system. Appl Therm Eng 115, 1063–1078. doi:10.1016/j.applthermaleng.2017.01.042

Lorentzen, G., 1994. Revival of carbon dioxide as a refrigerant. Int J Refrig 17, 292–301. doi:10.1016/0140-7007(94)90059-0

Nakagawa, M., Marasigan, A.R., Matsukawa, T., 2011. Experimental analysis on the effect of internal heat exchanger in transcritical CO_2 refrigeration cycle with two-phase ejector. Int J Refrig 34, 1577–1586. doi:10.1016/j.ijrefrig.2010.03.007

Pearson, A., 2005. Carbon dioxide — new uses for an old refrigerant. Int J Refrig 28, 1140–1148. doi:10.1016/j.ijrefrig.2005.09.005

Rigola, J., Ablanque, N., Pérez-Segarra, C.D., Oliva, A., 2010. Numerical simulation and experimental validation of internal heat exchanger influence on CO_2 trans-critical cycle performance. Int J Refrig 33, 664–674. doi:10.1016/j.ijrefrig.2009.12.030

Se, M., Hoon, D., Soo, M., Kim, M., 2017. Investigation on the optimal control of gas cooler pressure for a CO_2 refrigeration system with an internal heat exchanger. Int J Refrig 77, 48–59. doi:10.1016/j.ijrefrig.2017.03.002

Shariatzadeh, O.J., Abolhassani, S.S., Rahmani, M., Nejad, M.Z., 2016. Comparison of transcritical CO_2 refrigeration cycle with expander and throttling valve including/excluding internal heat exchanger: Exergy and energy points of view. Appl Therm Eng 93, 779–787. doi:10.1016/j.applthermaleng.2015.09.017

Singh, S., Dasgupta, M.S., 2017. CO_2 heat pump for waste heat recovery and utilization in dairy industry with ammonia based refrigeration. Int J Refrig 78, 108–120. doi:10.1016/j.ijrefrig.2017.03.009

Singh, S., Dasgupta, M.S., 2016. Evaluation of research on CO_2 trans-critical work recovery expander using multi attribute decision making methods. Renew Sustain Energy Rev 59, 119–129. doi:10.1016/j.rser.2016.01.013