Thermodynamic Analysis of CO_2 Transcritical Booster System for Supermarket Refrigeration in Warm Climatic Conditions

[a]Nilesh Purohit* ***[b]Dileep Kumar Gupta*** ***[a]Mani Sankar Dasgupta***

[a]Department of Mechanical Engineering, BITS-Pilani, Rajasthan; [b]Department of Mechanical Engineering, IITRAM, Gujarat

**Corresponding Author: +91-99509 80335; email: p2014010@pilani.bits-pilani.ac.in*

Abstract

This article presents a thermodynamic analysis of a hypothetical trans-critical CO_2 refrigeration system with enhanced booster and parallel compression for supermarket application in warm climatic conditions (35-50°C). The proposed system simultaneously incorporates two medium temperature and three low temperature loads for five different cooling demands typically found in supermarket application. These are HVAC load at 5°C, storage of dairy products at 0°C, frozen ice cream, vegetable and animal products at -10°C, -20°C and -30°C respectively. Some of the major parameters affecting performance like gas cooler operating pressure, inter-mediate vessel pressure, mass flow rate and gas cooler inlet temperature are investigated to reveal the practicality of the proposed system. The simulation results show that the new cycle configuration has a distinct advantage over a few other modified cycles proposed from time to time for supermarket refrigeration in warm climatic conditions. A few potential improvement strategies of COP are also discussed.

Keywords

Trans-critical booster system, Refrigeration, Supermarket, Thermodynamic analysis, Parallel compression, Carbon dioxide.

1. Introduction

Carbon dioxide (CO_2) based refrigeration systems for supermarkets are gaining acceptance in colder countries as a viable green technology. Design of supermarket refrigeration system has to meet additional challenges due to strict regulations laid down by various environmental protection protocols. Conventional supermarkets employ synthetic refrigerant based direct expansion refrigeration systems. Such designs contribute both directly and indirectly towards global warming. Half of the total energy consumption at supermarket is by the refrigeration system, consequently, operation of such systems contributes to global warming (Zhang, 2006). This energy consumption can be regulated mostly by adopting doored display cabinets and better control system (Fricke and Becker, 2010). It is estimated that refrigerants from these systems leak directly to the extent of 3 to 35%, depending on the age of the equipment and its usage (ICF Consulting, 2005). An obvious solution is to replace artificial refrigerants with natural ones. Among the many recognized natural refrigerants, CO_2 finds wider acceptance owing to its favourable properties (Kim et al., 2004). Potential challenges of trans-critical (TR) CO_2 cycle for warm climatic conditions have been discussed by Gupta et al., (2014).

Literature survey reveals that CO_2 systems have been employed for the supermarket refrigeration application as indirect, cascade and TR system. Among these, cascade and TR systems are gaining popularity globally. Kim et al., (2004) reported that CO_2 based refrigeration system has 3 to 10 times larger volumetric refrigeration capacity compared to systems based on artificial refrigerants such as R22, R404a etc. The comparison of annual electricity use of CO_2/R404A cascade system over direct expansion multiplex R404A and R22 systems is done by da Silva et al., (2012). They reported lower energy consumption for CO_2 based cascade system. Several prominent studies investigating TR CO_2 based system can be found in literature (Cecchinato and Corradi, 2012; Ferrandi and Orlandi, 2012; Ge and Tassou, 2011; Finckh et al., 2011); all these studies focus on improving the cycle performance for colder climatic condition. In this paper, a stage wise development of an efficient TR enhanced booster cycle is presented for warmer climatic condition. The performance of the same is also compared to a R22 based direct expansion cycle. Further, the same is applied for a hypothetical supermarket incorporating five commonly encountered refrigeration loads that are HVAC load at 5°C, storage of dairy products at 0°C, frozen ice cream at -10°C, vegetable at -20°C and animal products at -30°C. The novelty of the study lies in incorporation of an inter-mediate vessel between medium and low temperature evaporators which enables cooling of refrigerant. Moreover, replacing throttling valve with expander to generate and provide electric energy to partially drive the bypass compressor is also explored.

2. Development Phases for Proposed Cycle

Sawalha et al., (2015) carried out field measurements evaluating five supermarket refrigeration systems in Sweden. They reported that the systems with booster configuration as shown in *Figure 1* performed better for all investigated conditions and this particular cycle configuration is selected as the first stage or configuration A. Sharma et al., (2014) investigated seven cycle configurations for supermarket refrigeration at various locations in U.S.A. They concluded that TR booster cycle with parallel compression perform well among all kinds of tested TR cycle configurations. This cycle, configuration B as shown in *Figure 2*, is second stage explored here. Various studies investigating parallel compression in TR CO_2 system can be found from the literature (Purohit et al., 2015; Chesi et al., 2012; Sarkar and Agarwal, 2010). Introducing an intermediate vessel between low and medium temperature load in the cycle configuration B can improve the cycle performance as shown in *Figure 3* and is termed as configuration C.

In literature, many prominent studies can be found investigating expander as work producing device (Zhang et al., 2013), but none of the study is reported alongside TR booster cycle configuration for supermarket refrigeration. Incorporation of a work recovery expander between the gas cooler and the intermediate vessel pressure is explored as configuration D as shown in *Figure 4*.

Thermodynamic models are formulated by applying energy and mass balance across various components of the investigated cycles. The equations involved are summarized in *Table 1* and the parameters used for the simulation are listed in *Table 2*. The models are solved in MATLAB environment with Refprop 10 for refrigerant properties. A fixed upper pressure limit of 12 MPa is assumed considering material properties limitations. The mass flow across evaporator is calculated by dividing refrigeration load by specific enthalpy change across evaporator. The simulations are designed to extract the maximum (best possible) COP and corresponding operating parameters for complete range of ambient temperature and gas cooler pressure.

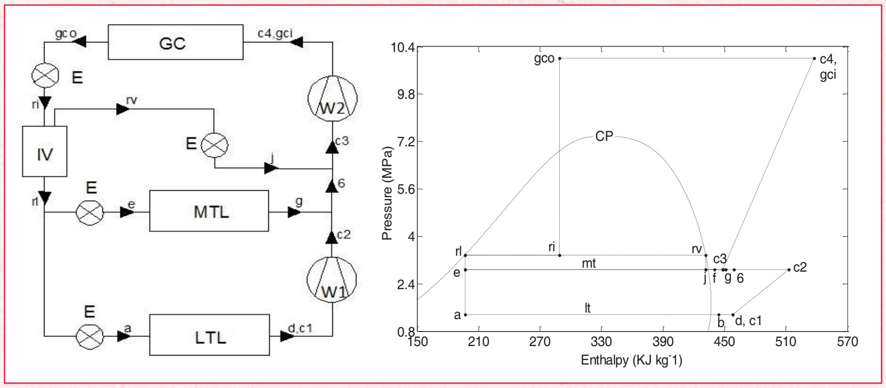


Figure 1: Cycle configuration A and P-h chart

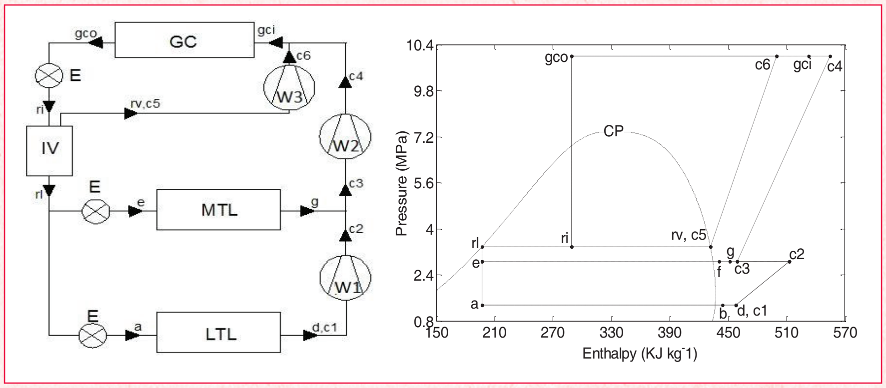


Figure 2: Cycle configuration B and P-h chart

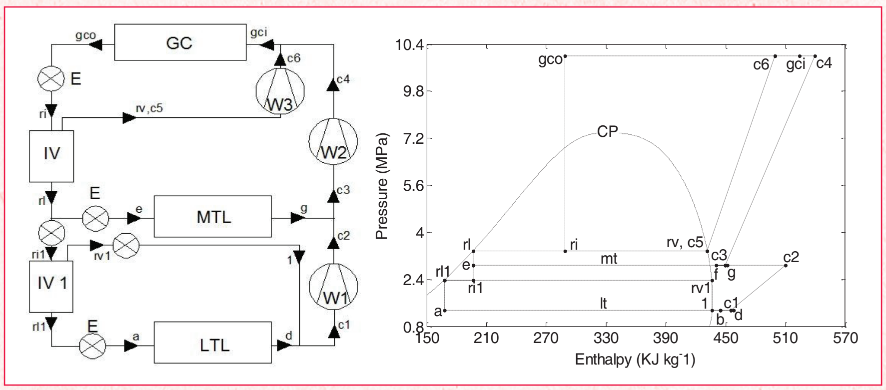


Figure 3: Cycle configuration C and P-h chart

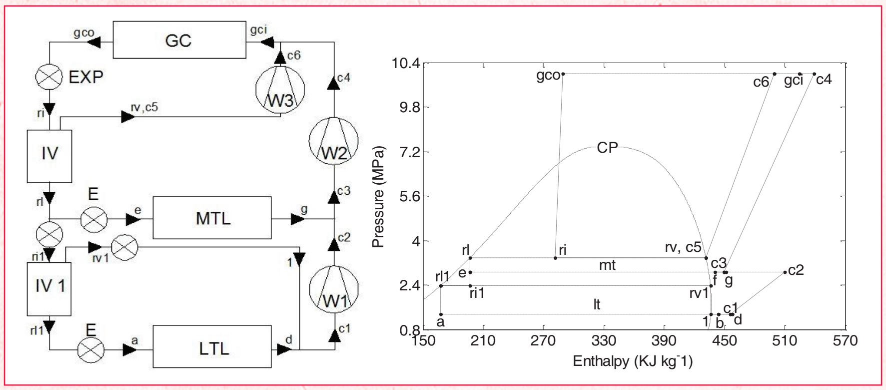


Figure 4: Cycle configuration D and P-h chart

Table 1: Refrigeration capacity, compressor work and mass flow rate

Conf.	Work of compressor (kW)	Mass flow rate (kg s^{-1})				
		m_1	m_2	m_3	m_v	m_{v1}
A	$W_{(i/2)} = m_{(i/2)}(h_{c(i)}-h_{c(i-1)})$	m_l	m_e+m_v	-	$m_e(h_{rl}-h_{ri})/h_{ri}-h_{rv})$	-
B	$W_{(i/2)} = m_{(i/2)}(h_{c(i)}-h_{c(i-1)})$	m_l	m_e	m_v	$m_e(h_{rl}-h_{ri})/h_{ri}-h_{rv})$	-
C	$W_{(i/2)} = m_{(i/2)}(h_{c(i)}-h_{c(i-1)})$	$m_{v1}+m_l$	m_e+m_{v1}	m_v	$m_t(h_{ri}-h_{rl})/h_{rv}-h_{rl})$	$m_l(h_{rl1}-h_{ri1})/h_{ri1}-h_{rv1})$
D	$W_{(i/2)}$ - $m_t(h_{expi}-h_{expo})$	$m_{v1}+m_l$	m_e+m_{v1}	m_v	$m_t(h_{ri}-h_{rl})/h_{rv}-h_{rl})$	$m_l(h_{rl1}-h_{ri1})/h_{ri1}-h_{rv1})$

i = 2,4,6

Table 2: Input parameters for the simulation

S. No.	Parameter	Range or Value
1	Evaporator temperature, mt, lt (°C)	-7, -31.3
2	Ambient temperature, Approach temperature (°C)	34-50, 3
3	Gas cooler operating pressure (MPa)	8-12
4	Refrigeration loads (kW) Q_{mt}, Q_{lt}	[(4×Tgco)+100], 35 (Sawalha et al., 2015)
5	Superheat, internal (mt, lt), external (mt, lt) (°C)	(4, 7), (8, 13)
6	Compressor, expander efficiencies	0.65, 0.60

The results of simulation from the investigated four cycle configurations (A, B, C & D) for best possible COP are summarized in *Figure 5*. The results are validated with the help of field data provided by Sawaha et al. (2015) as shown in *Figure 6 (a)*. However, the validation presented is restricted to sub-critical zone. *Figure 6 (b)* compares the simulated results of the cycle configurations (C & D) to that of R22 direct expansion system.

The conclusions made from the *Figure 5 & 6* are as follows:

- COP of all cycles decreases with increase in ambient temperature.
- The proposed cycle D gives better COP over other investigated cycle configurations for all ambient temperatures.
- Gas cooler operating pressure is found to be lower for cycle D at all investigated cases. However, at higher ambient temperatures, the operating pressure for the cycles converges to 12 MPa owing to the limiting pressure (12 MPa) kept during simulation.
- Gas cooler inlet temperature for cycle D reaches its operating limit at relatively higher ambient temperature compared to other cycles. The slope of the curves changes their nature as they cross operating limit. This can be attributed to the higher amount of vapour generation at intermediate vessel due to increase in ambient temperature at same operating gas cooler pressure. Further, at the onset

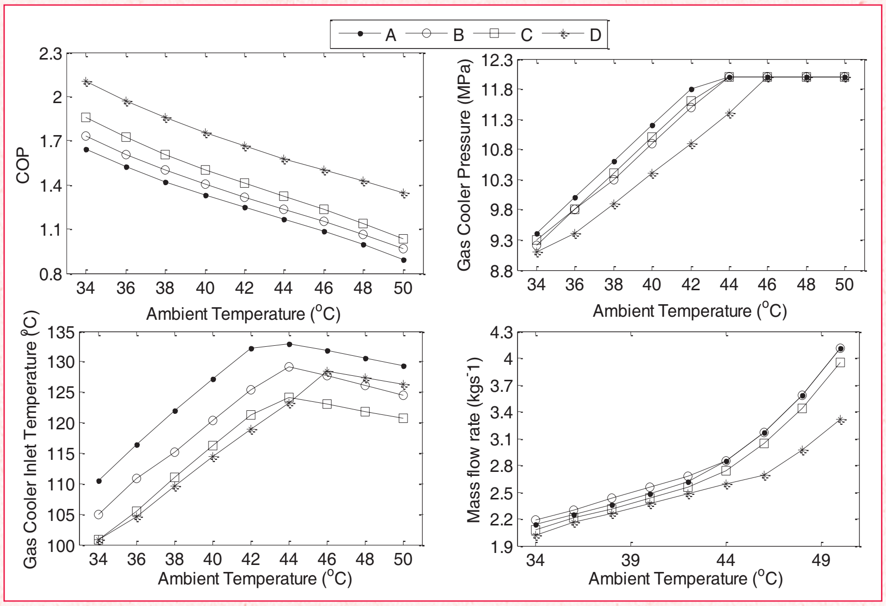


Figure 5: Comparison of results for simulated cycle configurations

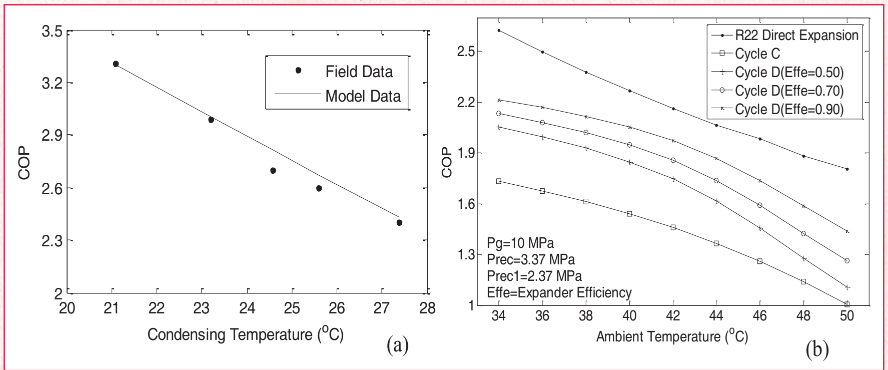


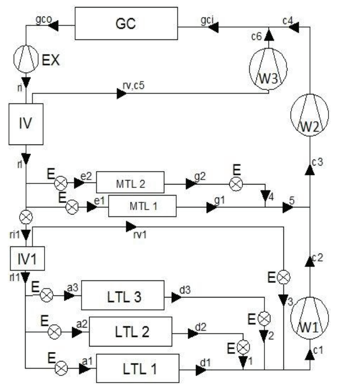
Figure 6: (a) Model validation; (b) Comparison of results of cycle C & D with R22 cycle

of operating limit for cycle D, the gas cooler inlet temperature is found to be higher than that for cycles B & C owing to the expansion process with expander which contributes to lower vapour generation at intermediate vessel.

- The mass flow rate for all cycles are found to increase with increase in ambient temperature. Beyond the operating pressure limit, the rate of increment in mass flow rate increases sharply. However, for cycle D it is lower.
- Simulation shows a good agreement with the field data extracted from the Sawalha et al., (2015).
- The performance of cycle configuration C and D is lower to R22 cycle. Further, at ambient temperature beyond 40°C the slope of COP for the cycle C and D is steeper and this may be attributed to the imposed constant gas cooler pressure of 10 MPa.
- Comparing all tested configurations, cycle D is chosen for application for the hypothetical supermarket for warm climatic conditions.

3. Proposed Cycle Configuration Applied to Supermarket

In this section implementation of a cycle configuration D for supermarket refrigeration application for warm climatic conditions is discussed. Five refrigeration loads as discussed in Introduction are assumed. The schematic of proposed cycle and its operating parameters are shown in *Figure 7* and its P-h diagram is shown in *Figure 8*. The cycle is thermodynamically modelled in MATLAB environment for the best possible COP. *Figure 9* depicts the performance of the proposed cycle with expander and compares the same with a cycle without expander. The effect of intermediate pressure and the real time constraints on the cycle COP is also presented as shown in *Figure 10 (a) and (b)*. *Table 3* shows the individual component irreversibility for proposed cycle with and without expander.



Parameter	Range or Value
lt1, lt2, lt3 (°C)	-30, -20, -10
mt1, mt2 (°C)	-5, 0
Q_{lti}, Q_{mtj} (kW), i=1, 2, 3 j=1, 2	35, ($4\times T_{amb}$)+100
t_{amb} (°C)	34-50
AT (°C)	3
p_{gco} (MPa)	8-12
Internal superheat lt1, lt2, lt3 (°C)	6, 5, 3
External superheat lt1, lt2, lt3 (°C)	13, 9, 5
Internal super. mt1, mt2 (°C)	4, 3
External super. mt1, mt2 (°C)	8, 6
Compressor, Expander efficiencies	0.65, 0.60
Reference Temp. for evap. (°C)	5°C above evaporator

Figure 7: Schematic of proposed hypothetical supermarket and the operating parameters range or values

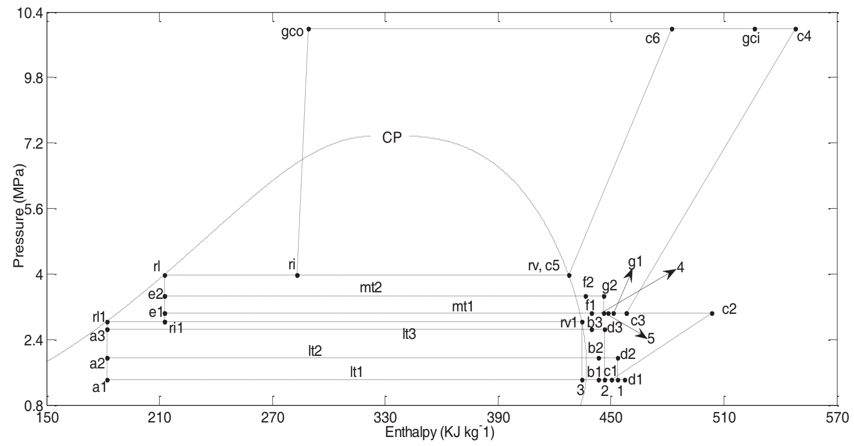


Figure 8: P-h chart for proposed hypothetical cycle for supermarket

The energy, mass and exergy balance equations for the proposed cycle are given as in Equation 1-16.

$$W_{(i/2)} = m_{(i/2)}(h_{c(i)}-h_{c(i-1)}),\ i = 2,4 \quad (1)$$

$$W_{(i)} = m(h_{c(2i)}-h_{c(2i-1)})-m(h_{expi}-h_{expo}),\ i = 3 \quad (2)$$

$$m_{li} = Q_{lt}/(h_{bi}-h_{ai}),\ i = 1,2,3 \quad (3)$$

$$m_{mi} = Q_{mt}/(h_{fi}-h_{ei}),\ i = 1,2 \quad (4)$$

$$m_1 = m_{l1}+m_{l2}+m_{l3}+m_{v1} \quad (5)$$

$$m_2 = m_1+m_{m1}+m_{m2} = m_e \quad (6)$$

$$m_3 = m_{v1} \quad (7)$$

$$m_v = m_t(h_{ri}-h_{rl})/h_{rv}-h_{rl}) \quad (8)$$

$$m_{v1} = (m_{l1}+m_{l2}+m_{l3})(h_{rl1}-h_{ri1})/h_{ri1}-h_{rv1}) \quad (9)$$

$$COP = (Q_{lt1}+Q_{lt2}+Q_{lt3}+Q_{mt1}+Q_{mt2})/(W_1+W_2+W_3-W_e) \quad (10)$$

$$Ievaporator_{lti} = m_{li}[(s_{bi}-s_{ai})-((h_{bi}-h_{ai})/(t_r))][t_{amb}],\ i = 1,2,3 \quad (11)$$

$$Ievaporator_{mti} = m_{li}[(s_{fi}-s_{ei})-((h_{fi}-h_{ei})/(t_r))][t_{amb}],\ i = 1,2 \quad (12)$$

$$Icompressor_{(i/2)} = m_{(i/2)}(s_{c(i)}-s_{c(i-1)})(t_{amb}),\ i=2,4,6 \quad (13)$$

$$IGC = m_t[(h_{gci}-h_{gco})-(t_{amb}(s_{gci}-s_{gco}))] \quad (14)$$

$$Ivalve = m_{valve}(t_{amb})(\Delta s)_{valve} \quad (15)$$

$$Iexp = m_t(t_{amb})(s_{expo}-s_{expi}) \quad (16)$$

The conclusions made from *Figure 9, 10* and *Table 3* are as follows:

- The cycle with expander has better COP than that without expander.
- The mass flow rate and the operating pressure are lower for the proposed cycle with expander. Beyond operating pressure limit, the gas cooler inlet temperature is higher for cycle with expander.

- The effect of gas cooler pressure on COP of cycle is significant while effect of intermediate pressure is less significant.
- The variation of cycle performance with real time parameters like approach temperature, pipe loss, compressor efficiency and expander efficiency are also studied. Effect of compressor efficiency is found most prominent. The real time constraints have more significance on the cycle performance at lower gas cooler operating pressure.
- Exergy analysis of the cycle reveals that the gas cooler, the compressor and the expansion valves are the three most sensitive components of the cycle. Using expander, the irreversibility of the expansion valves can be reduced. Second law efficiency is found better for proposed cycle with expander.
- At higher ambient temperature, the further improvement strategies can be: use of mechanical and thermoelectric sub-cooling unit beyond gas cooler and intermediate vessels. However, for large capacity systems incorporating such devices needs further investigations.

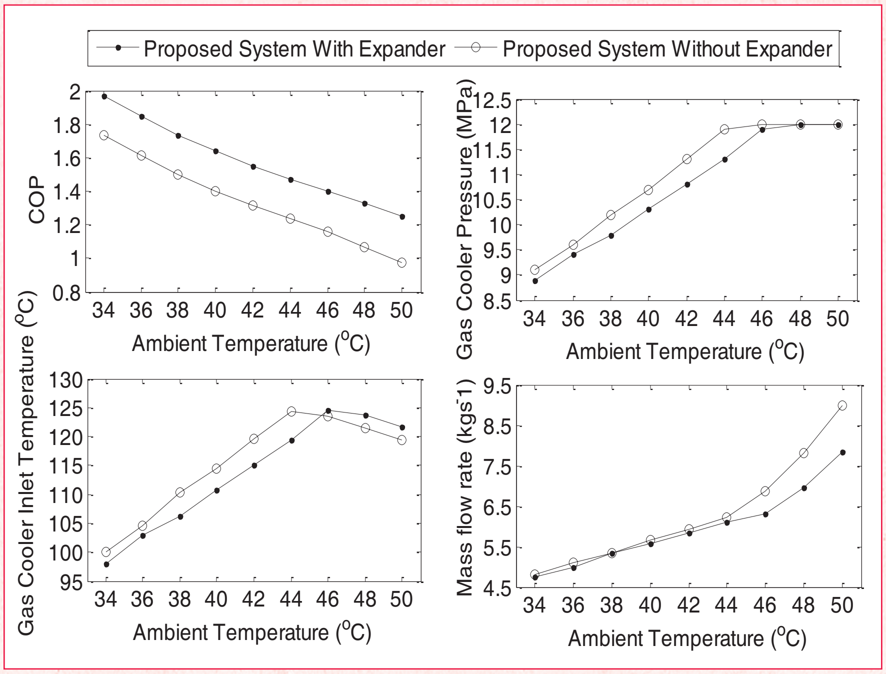


Figure 9: Proposed supermarket cycle performance results with respect to ambient temperature (°C)

4. Conclusions

Comparative analysis of various booster trans-critical CO_2 cycles and proposed enhanced booster system reveals the following:

- The proposed cycle has expander, parallel compression and an intermediate vessel placed between medium and low temperature load. It performs better as compared to other conventional booster cycles. The thermodynamic model developed for conventional booster cycle shows good agreement with published field data.
- Proposed cycle is successfully applied to a hypothetical supermarket refrigeration system operating in warm climatic conditions incorporating five refrigeration loads.
- The performance of proposed supermarket cycle with expander is higher as compared to that without expander. Also, the mass flow rate and gas cooler operating pressure is lower for cycle with expander. The gas cooler inlet temperature is higher for cycle with expander beyond the operating limit (12 MPa).

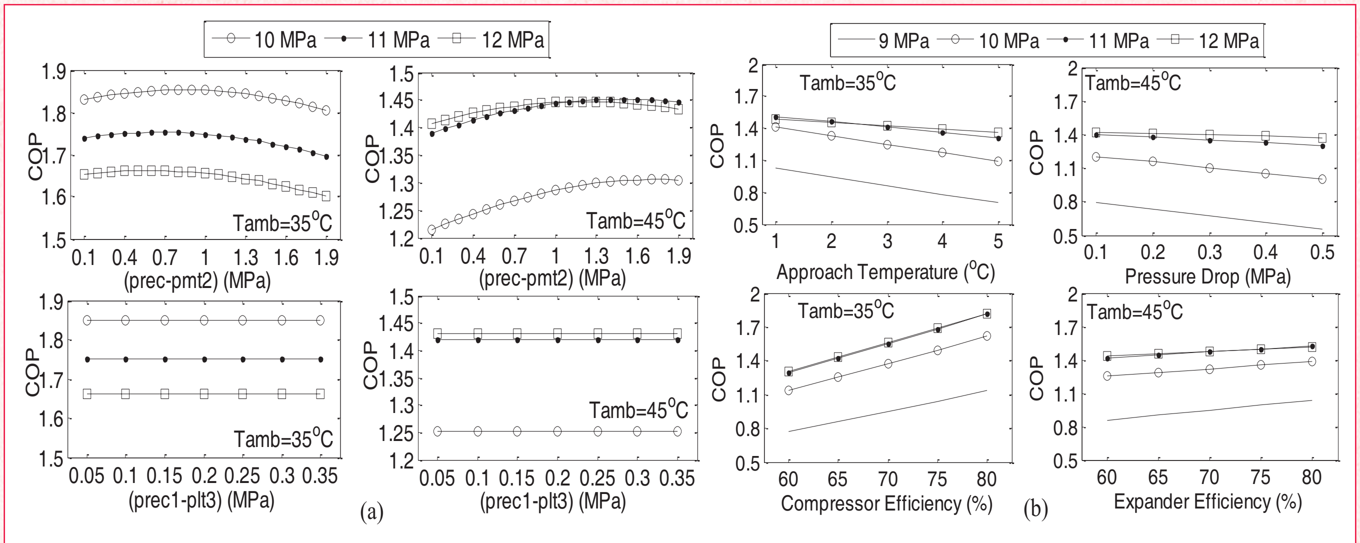


Figure 10: (a) Effect of intermediate pressure on proposed supermarket cycle (with expander) COP; (b) Effect of real time constraints on proposed supermarket cycle (with expander) performance

Table 3: Entropy generation (kW) for components in proposed supermarket refrigeration cycle (at 45°C)

Component	Proposed supermarket cycle without Exp.			Proposed supermarket cycle with Exp.		
	10 MPa	11 MPa	12 MPa	10 MPa	11 MPa	12 MPa
Evaporators	15.02	15.02	15.02	15.02	15.02	15.02
Compressors	232.5	178.30	166.70	200.6	163.5	155.24
Gas cooler	78.61	87.73	103.48	70.18	82.7	97.92
Exp. valves	290.12	181.83	154.52	20.8	20.8	20.8
Expander	-	-	-	106.13	67.81	57.30
I_t for cycle	616.31	462.89	439.72	412.8	349.4	346.36
II Law Eff (%)	18.46	22.87	23.67	24.97	28.01	28.08

- Effect of gas cooler pressure compared to intermediate vessel pressure on the cycle performance is found to be more prominent. Also, the effect of a few real time constraints on the cycle COP is studied. Among these, compressor efficiency is found to be the most sensitive parameter. Exergy analysis depicts higher second law efficiency for the proposed cycle with expander.

Nomenclature

AT..................Approach temperature(°C)
C....................Compressor
Eff..................Efficiency
GC.................Gas cooler
E.....................Expansion valve
EXP...............Expander
I......................Irreversibility (kW);
subscript t.....total
IV, rec............Int. vessel b/w p_{gco} and p_{mt}
IV1, rec1........Int. vessel b/w p_{mt} and p_{lt}
Q....................Refrigeration load (kW)
W...................Compressor work (kW)
a.....................lt evaporator inlet
b.....................lt evaporator outlet
c.....................compressor (inlet & exist)
d.....................lt external superheat
e.....................mt evaporator inlet
f......................mt evaporator outlet
g.....................mt external superheat
gci.................GC inlet
gco................GC outlet
h.....................Enthalpy (kJ kg^{-1} K^{-1})
lt.....................Low temperature (°C)
mt..................Medium temperature (°C)
m_l...................Mass flow (lt evap.) (kg s^{-1})
m_m..................Mass flow (lt evap.) (kg s^{-1})
m_1..................Mass flow (W1) (kg s^{-1})
m_2..................Mass flow (W2) (kg s^{-1})
m_3..................Mass flow (W3) (kg s^{-1})
m_v..................Mass flow (IV) (kg s^{-1})
m_{v1}................Mass flow (IV1) (kg s^{-1})
m_t...................Total mass flow (kg s^{-1})
p.....................Pressure (MPa)
s.....................Entropy (kJ kg^{-1} K^{-1})
t......................Temperature (°C)
CP..................Critical point
amb................Ambient

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