International Journal of Thermal Sciences 49 (2010) 838–843

Contents lists available at ScienceDirect

International Journal of Thermal Sciences

journal homepage: www.elsevier.com/locate/ijts

# Performance optimization of transcritical $CO_2$ cycle with parallel compression economization

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ARTICLE INFO

*Article history:*
Received 15 May 2009
Received in revised form
28 November 2009
Accepted 1 December 2009
Available online 30 December 2009

*Keywords:*
Transcritical $CO_2$ cycle
Parallel compression economization
Optimization
Performance improvement
Comparison

ABSTRACT

Being a low critical temperature fluid, $CO_2$ transcritical system offers low COP for a given application. Parallel compression economization is one of the techniques to improve the COP for transcritical $CO_2$ cycle. An optimization study of transcritical $CO_2$ refrigeration cycle with parallel compression economization is presented in this paper. Further, performance comparisons of three different COP improvement techniques; parallel compression economization alone, parallel compression economization with recooler and multistage compression with flash gas bypass are also presented for chosen operating conditions. Results show that the parallel compression economization is more effective at lower evaporator temperature. The expression for optimum discharge pressure has been developed which offers useful guideline for optimal system design and operation. Study shows that the parallel compression with economizer is promising transcritical $CO_2$ cycle modifications over other studied cycle configurations. A maximum improvement of 47.3% in optimum COP is observed by employing parallel compression economization for the studied ranges.



## 1. Introduction

While the synthetic refrigerants exhibit considerably high ODP and GWP, lately the environment friendly natural refrigerant carbon dioxide has gained considerable interest and identified as a suitable alternative refrigerants owing to its excellent heat transfer properties and it is non-flammable and non-toxic [1]. However, critical temperature for $CO_2$ is quite low (31.1 °C). A fluid with a lower critical temperature will tend to have a higher volumetric capacity and a lower COP for a given application [2]. The lower COP is related to high level of irreversibility because of the superheated vapour horn and the throttling process [3,4]. A significant amount of research has been carried out to improve the COP of transcritical $CO_2$ systems by cycle modification such as employing internal heat exchanger, multistage compression, expansion turbine, vortex tube and ejector expansion device [5]. Cecchinato et al. [6] carried out thermodynamic analysis on two-stage transcritical $CO_2$ cycles. It is shown that employing double compression with intercooling improves the performance significantly and also governs the choice of optimum intermediate pressure.

Parallel compression economization is one of the techniques where refrigerant vapour is compressed to supercritical discharge pressure in two separate non-mixing streams; one coming from an economizer and the other coming from the main evaporator to improve the performance of transcritical $CO_2$ refrigeration cycle. The parallel compression system will have wide application for automotive air conditioning, window air conditioners and small water chillers where it is not appropriate to use screw or scroll compressors [7]. Bell [8] has carried out theoretical and experimental study on parallel compression economization $CO_2$ refrigeration cycle considering suction superheat. It is concluded that parallel compression economization is more beneficial with $CO_2$ transcritical system than similar hydrocarbon system in terms of efficiency and capacity under the same conditions. However, this simplified study did not include the optimization issue of the cycle.

In the present study an optimization of transcritical $CO_2$ refrigeration cycle with parallel compression economization has been carried out. Further, performance comparisons with other similar cycle layouts (parallel compression with recooler [9,10] and multistage with flash gas bypass [11]) are presented.

## 2. Mathematical modeling and simulation

The flow diagram and corresponding pressure–enthalpy diagram of a transcritical $CO_2$ cycle with parallel compression

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1290-0729/$ – see front matter 
doi:10.1016/j.ijthermalsci.2009.12.001

**Nomenclature**

| | |
|---|---|
| COP | coefficient of performance |
| GWP | Global warming potential |
| $h$ | specific enthalpy (kJ kg$^{-1}$) |
| $\dot{m}$ | mass flow rate of refrigerant (kg s$^{-1}$) |
| ODP | Ozone depletion potential |
| $p_{d,opt}$ | optimum discharge pressure (bar) |
| $q$ | refrigerating effect (kJ) |
| $t$, $T$ | temperature (°C, K) |
| V | valve |
| $w$ | compressor work (kJ) |
| $x$ | vapour mass fraction |
| $\varepsilon$ | recooler effectiveness |

*Subscripts*

| | |
|---|---|
| 1–10 | refrigerant state points |
| b | basic cycle |
| c | compressor |
| co | gas cooler exit |
| ev | evaporator |

economization is shown in Fig. 1. The liquid (state 5) and vapour (7) are separated in economizer after the expansion of transcritical fluid from states 3 to 4 in primary expansion valve $V_1$. The liquid from the separator is further expanded in expansion valve $V_2$ to provide cooling effect in the evaporator (from states 6 to 1). The saturated vapour from evaporator and economizer is compressed in the compressor simultaneously to the states 2 and 8, respectively. The mixed stream (state 9) enters to the gas cooler for heat rejection to the external fluid (states 9–3). The entire system has been modeled based on the energy balance of individual components of the system. Steady flow energy equations based on first law of thermodynamics have been employed in each case and specific energy quantities are used. The following assumptions have been made in the thermodynamic analysis:

- Heat transfer with the ambient is negligible
- Compression process is adiabatic but non-isentropic
- Evaporation and gas cooling processes are isobaric
- Separation and mixing processes are isobaric
- Refrigerant at evaporator outlet is saturated vapour

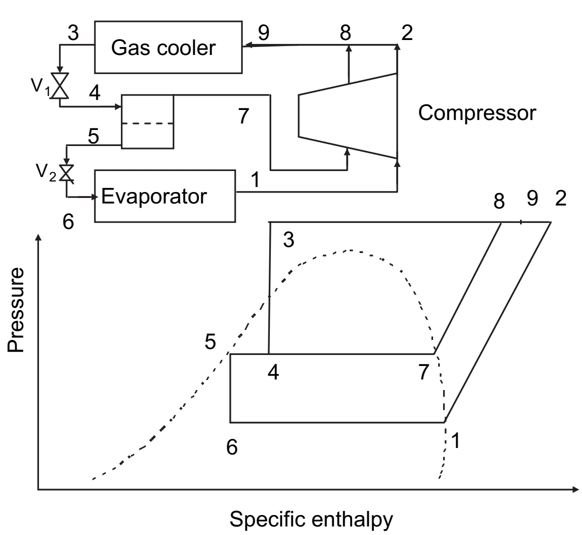


**Fig. 1.** Layout and $p$–$h$ diagram of $CO_2$ cycle with parallel compression economization.

Further it is assumed that process in both the expansion valves is isenthalpic which brings

$$h_3 = h_4; \quad h_5 = h_6 \tag{1}$$

For unit total mass flow rate, the mass flow rates through the economizing and main compressors are $x_4$ and $1 - x_4$, respectively, where, $x_4$ is given as:

$$x_4 = (h_4 - h_5)/(h_7 - h_5) \tag{2}$$

The isentropic efficiency of the compressor is given as:

$$\eta_{is,comp} = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{h_{8s} - h_7}{h_8 - h_7} \tag{3}$$

Refrigerating effect of the evaporator:

$$q_{ev} = (1 - x_4)(h_1 - h_6) \tag{4}$$

Work input to the compressor:

$$w_c = x_4(h_8 - h_7) + (1 - x_4)(h_{2s} - h_1) \tag{5}$$

The cooling COP for parallel compression economization is given by:

$$\mathrm{COP} = q_{ev}/w_c \tag{6}$$

The cooling COP of corresponding basic cycle is given by:

$$\mathrm{COP_b} = (h_1 - h_3)/(h_2 - h_1) \tag{7}$$

A computer code has been developed for the steady state simulation to evaluate the system performance of the proposed parallel compression economization $CO_2$ cycle under different operating conditions. Subcritical and supercritical thermo-physical properties of $CO_2$ are estimated employing a precision property code CO2PROP developed locally [12]. For the given evaporator and gas cooler exit temperatures, and gas cooler and economizer pressures, properties at states 1, 3, 5 and 7 are calculated. Properties of states 4 and 6 are calculated employing Eq. (1) and then economizing mass fraction is evaluated by Eq. (2). Compressor exit conditions (2 and 8) are evaluated using compressor isentropic efficiency relation (Eq. (3)). Further, state 9 is also calculated. Using Eqs. (4)–(7), the performance parameters: cooling capacity, compressor work, COP and $\mathrm{COP_b}$ are calculated.

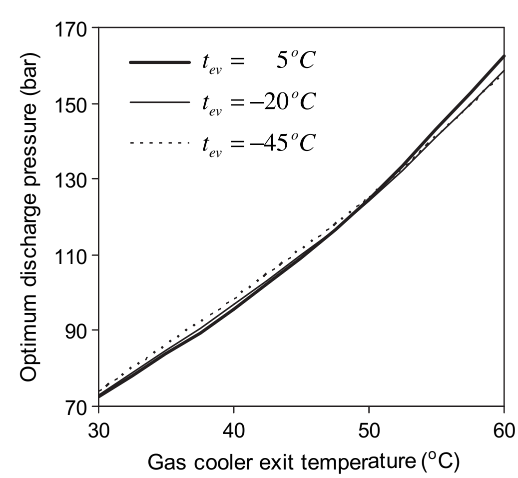


**Fig. 2.** Variation of optimum discharge pressure with gas cooler exit temperature.

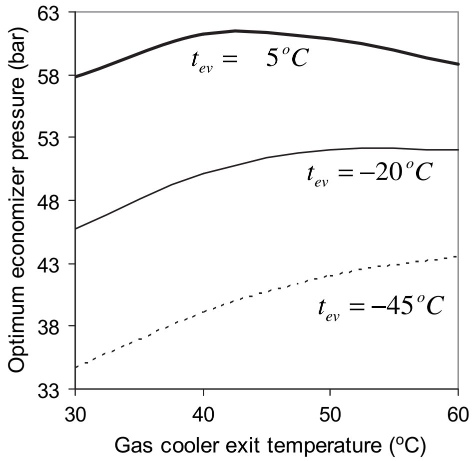


**Fig. 3.** Variation of optimum economizer pressure with gas cooler exit temperature.

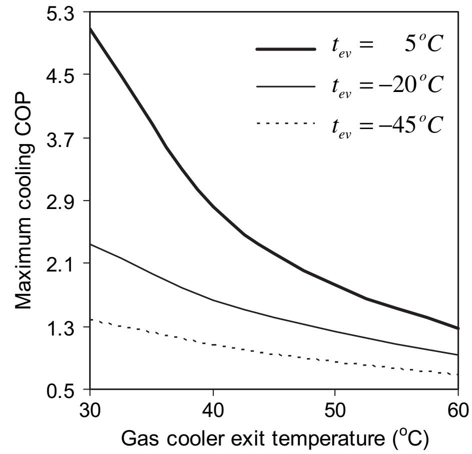


**Fig. 5.** Variation of maximum cooling COP with gas cooler exit temperature.

## 3. Results and discussion

It is reported [11,12] that an optimum gas cooler pressure exists for the transcritical $CO_2$ cycle where it exhibits the maximum COP for a given gas cooler exit temperature. However, in case of two-stage compression transcritical $CO_2$ cycle, the intermediate pressure is also an influential parameter to decide the best COP along with the gas cooler pressure [6,11]. It makes necessary to optimize the gas cooler pressure and intermediate pressure simultaneously.

The performance of the optimized transcritical carbon dioxide cycle with parallel compression economization on the basis of maximum cooling COP are presented for various evaporator and gas cooler exit temperatures. Isentropic efficiency depends on both compressor design and pressure ratio. However, it is observed that it has negligible effect on optimum pressures. Hence, the isentropic efficiency for both main and economizing compressors is taken a constant value of 75% to accommodate non-isentropic compression. In the search for optimum gas cooler pressure and economizer pressure, simultaneous variation of the gas cooler pressure and economizer pressure with a step size of 0.2 bar was taken in numerical simulation. Existence of the optimum economizer pressure is mainly on account of the changing slope of saturation curve while optimum discharge pressure is existed due to the unique behavioural pattern of $CO_2$ properties around the critical point where the slope of the isotherms is quite modest for a specific pressure range; at pressure above and below this range, the isotherms become much steeper. Improvement in COP and reduction in discharge pressure are compared with optimized basic cycle by keeping operating conditions the same.

Variation of optimum discharge pressure with gas cooler exit temperature for three evaporator temperatures, $t_{ev} = 5$ °C, −20 C and −45 °C is shown in Fig. 2. Optimum discharge pressure increases rapidly with gas cooler exit temperature. However, variation of optimum economizer pressure is quite modest with gas cooler exit temperature (Fig. 3). This can be attributed to the fact that although the shape of isotherm changes with temperature, the shape of saturation curve remains unaltered. Optimum economizer pressure is comparatively more influenced by evaporator temperature than that of gas cooler exit temperature (Fig. 3). It is shown in Fig. 2 that evaporator temperature has negligible effect on optimum discharge pressure.

As discussed earlier that the liquid and vapour are separated in economizer after the expansion of transcritical fluid in primary expansion valve $V_1$ to minimize the vapour entry in the evaporator. Fig. 4 exhibits the variation of mass fraction '$x_4$' goes in the economizer with gas cooler exit temperature at various evaporator temperatures. It can be seen that economizer mass fraction increases with increase in cycle temperature lift (Fig. 4). This may

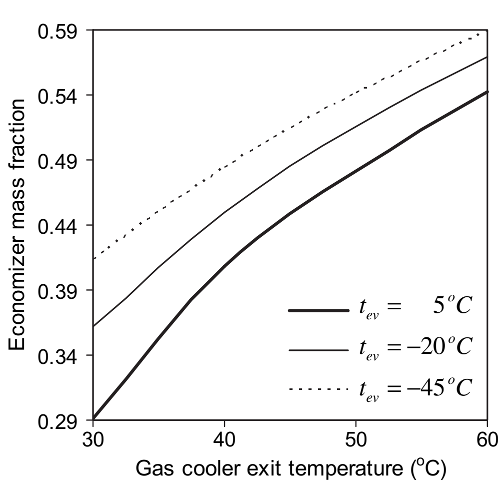


**Fig. 4.** Variation of optimum economizer mass fraction with gas cooler exit temperature.

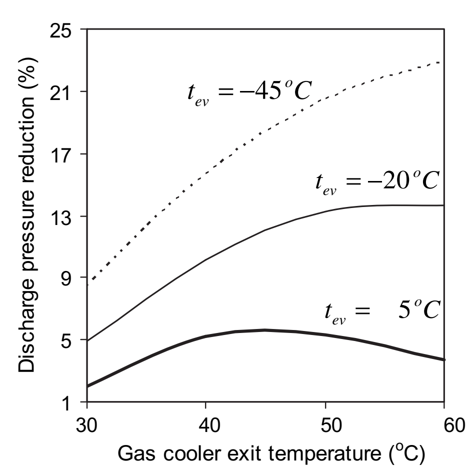


**Fig. 6.** Variation of percentage discharge pressure reduction in comparison with basic optimum cycle with gas cooler exit temperature.

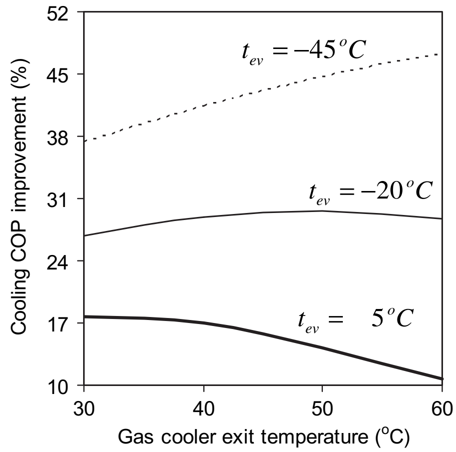


**Fig. 7.** Variation of percentage cooling COP improvement in comparison with basic optimum cycle with gas cooler exit temperature.

be due to the fact that at high gas cooler exit temperature, optimum discharge pressure is high while optimum economizer pressure decreases with lowering the evaporator temperature (Figs. 2 and 3). With decrease in evaporator temperature, optimum economizer pressure decreases, although difference increases which leads to increase in quality at the evaporator inlet. Fig. 5 shows that optimum cooling COP increases with decrease in temperature lift similar to the basic $CO_2$ transcritical cycle. This implies that $CO_2$ transcritical system should operate at lower gas cooler exit temperature to have higher optimum cooling COP and lower optimum discharge pressure.

Reduction in optimum discharge pressure improves the COP [11]. One of the techniques to reduce the optimum discharge pressure is by adopting parallel compression economization. Percentage reduction (compare to optimum basic cycle) in optimum discharge pressure and improvement in optimum cooling COP with gas cooler exit temperature are shown in Figs. 6 and 7, respectively. Results show that the percentage improvement in cooling COP and percentage reduction in discharge pressure first increase and then decrease with gas cooler exit temperature at all the chosen evaporator temperatures with a peak value. The peak value shifts towards higher gas cooler temperature as the evaporator temperature decreases. This may be attributed to the fact that as the gas cooler temperature increases, initially vapour quality increases with faster rate than that of later stages (due to unique behaviour of isentropic and isotherm) and hence the compressor work reduction increases initially at faster rate and then at slower rate. Consequently, parallel compression cycle COP decreases at slower rate initially than that of basic cycle COP and hence percentage improvement of COP increases and then decreases. Never-the-less there is gain in terms of reduction in optimum discharge pressure and improvement in maximum cooling COP and both increase with decrease in evaporator temperature. It may be noted that gain in cooling COP is as high as 47.3% while reduction in optimum discharge pressure in the range of 2–23% for the chosen conditions. Hence, it may be concluded that parallel compression economization is more profitable at lower evaporator temperature. Fig. 8 shows the variations of discharge pressure, economizer pressure, COP and COP improvement for pressure ratio dependent compressor isentropic efficiency (correlation given in Ref. [12]). Results show that the variation trends of all parameters are similar to those of constant isentropic efficiency as shown above, whereas absolute values are slightly different due to different isentropic efficiency value ranges.

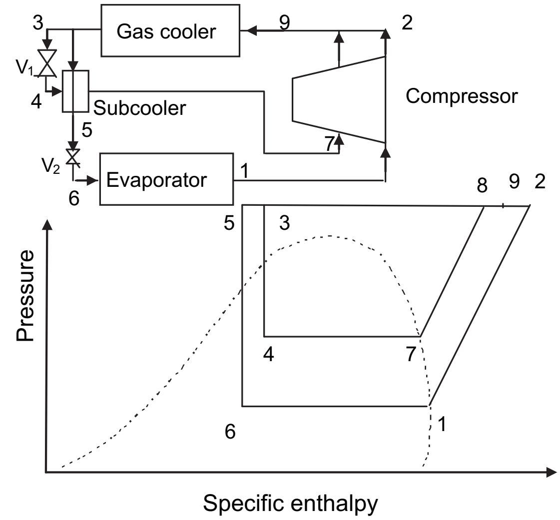


**Fig. 9.** Layout and $p$–$h$ diagram of parallel compression cycle with recooler.

Performing a regression analysis on the data obtained from the cycle simulation, the following relation are obtained to predict the optimum discharge pressure for the parallel compression system studied here for the chosen temperature ranges; $t_{ev} = -45$ °C to 5 °C and $t_{co} = 30$ °C–60 °C.

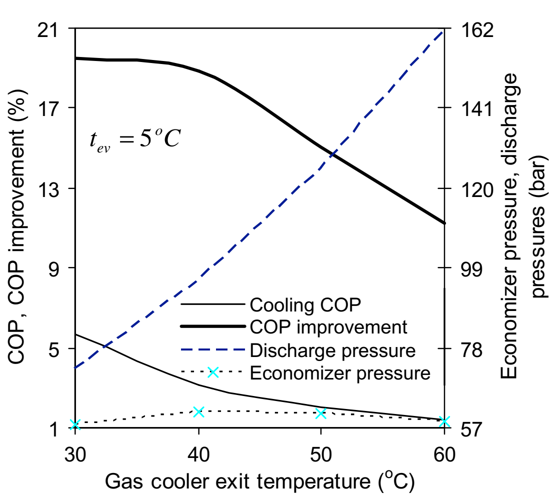


**Fig. 8.** Variations of optimum parameters for variable compressor isentropic efficiency.

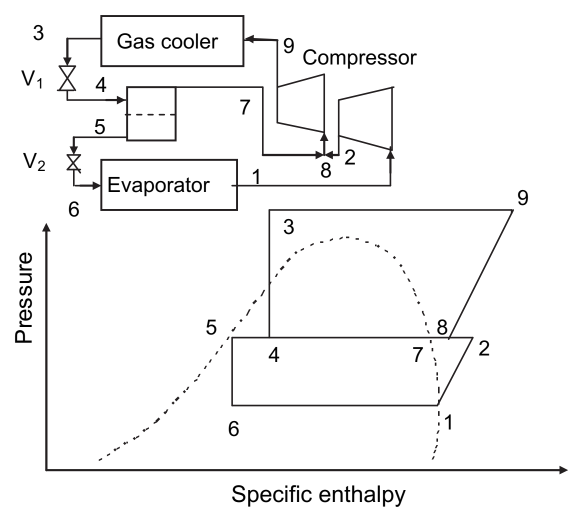


**Fig. 10.** Layout and $p$–$h$ diagram of two-stage cycle with flash gas bypass.

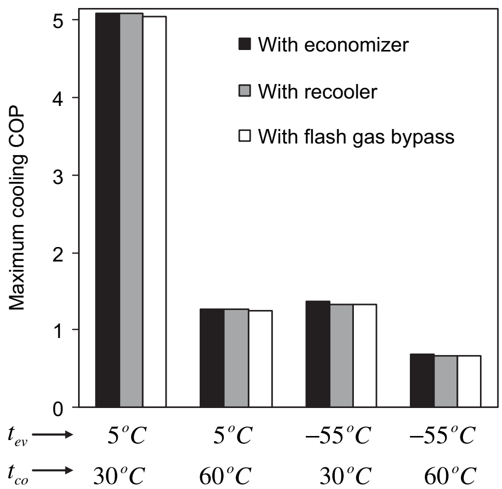


**Fig. 11.** Comparison based on maximum cooling COP.

$$p_{d,opt} = 36.877 - 0.00004 t_{ev} + 0.38234 t_{co} + 0.027667 t_{co}^2 \quad (8)$$

where temperature in °C.

Effect of employing parallel compression with recooler and multistage compression with flash gas bypass is also presented here. Flow diagram and corresponding *p*–*h* representation of parallel compression cycle with recooler are shown in Fig. 9. The exit transcritical vapour from gas cooler is re-cooled (3–5) in recooler by the secondary stream (4–7) which is at lower pressure and temperature due to expansion in the valve $V_1$, prior to entry in the recooler. The re-cooled liquid is further expanded (5–6) in valve $V_2$ to provide useful cooling effect in the evaporator (6–1). Compression processes are similar to parallel compression economization cycle. It is assumed that exit state (7) of the cooling stream in recooler is dry saturated, which can be maintained by the proper splitting of refrigerant flow from gas cooler.

Properties at 5 can be found by using effectiveness of recooler, which is taken as 0.8, given as:

$$T_5 = T_3 - \varepsilon (T_3 - T_7) \quad (9)$$

Applying energy conservation for recooler, the mass flow rate at 7 for unit total mass flow rate is given by:

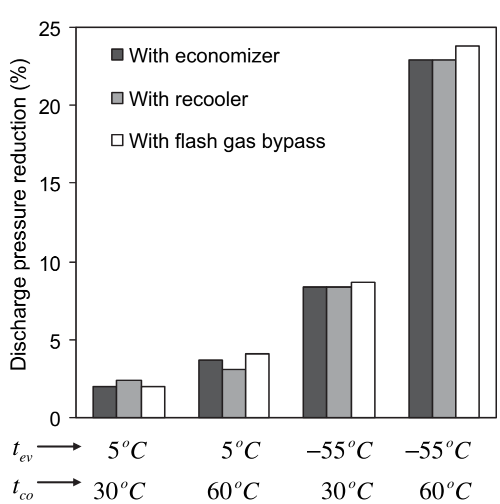


**Fig. 12.** Comparison based on optimum discharge pressure reduction.

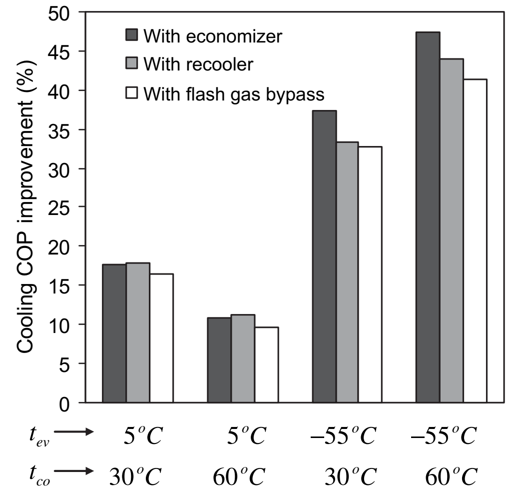


**Fig. 13.** Comparison based on maximum cooling COP improvement.

$$\dot{m}_7 = (h_3 - h_5)/(h_7 - h_5) \quad (10)$$

then performance of the recooler system is given as:

$$\text{COP} = \frac{(1-\dot{m}_7)(h_1 - h_6)}{(1-\dot{m}_7)(h_2 - h_1) + \dot{m}_7(h_8 - h_7)} \quad (11)$$

Flow diagram of two-stage transcritical $CO_2$ system with flash gas bypass is shown in Fig. 10 where vapour from low pressure compressor (1–2) mixes with flash gas (state 7) and then compressed (8–9) to gas cooler pressure. The cooling COP is given as [11]:

$$\text{COP} = \frac{(1-x_4)(h_1 - h_6)}{(1-x_4)(h_2 - h_1) + (h_9 - h_8)} \quad (12)$$

where $x_4$ is the vapour mass fraction at state 4.

Performance comparisons in terms of cooling COP, percentage reduction in discharge pressure and percentage improvement in optimum COP of the three systems explained earlier are shown in Figs. 11–13, respectively, for various evaporator and gas cooler exit temperatures. Results show that the parallel compression cycles with economizer and recooler are similar in terms of cooling COP, discharge pressure reduction and COP improvement (Figs. 11–13). However, use of parallel compression with economizer is more profitable for lower temperature applications. Increase in recooler effectiveness may give more COP improvement. Considering the economics, parallel compression cycle with economizer is better than that with recooler due to the lower cost of separator than recooler. Figs. 11 and 13 exhibit that performance of the parallel compression cycle with economizer is better than two-stage cycle with flash gas bypass. It can be said that employing parallel compression economization in a simple transcritical $CO_2$ cycle is the most effective way to improve the cycle performance.

## 4. Conclusions

A detailed optimization study of transcritical $CO_2$ cycle with parallel compression economization is presented here. Further, a comparative study of three systems; transcritical $CO_2$ system with parallel compression economization, recooler and flash gas bypass are also included. Discharge pressure and intermediate pressure are simultaneously optimized based on cooling COP. Employing parallel compression economization not only improves the optimum cooling COP, but also brings down the optimum discharge pressure. Optimum discharge pressure varies significantly with gas cooler exit temperature. However, the optimum economizer pressure varies marginally, in contrast to the variation with evaporator temperature with gas cooler pressure. The expression for optimum discharge pressure has been developed which offers useful guideline for optimal system design and operation. Usefulness of parallel compression economization is more significant at lower evaporator temperature. It is observed that cycle configuration is insignificant in terms of maximum cooling COP. However, the cycle configuration is significant with respect to percentage improvement in COP. Use of parallel compression with economizer is more profitable for lower temperature applications. Employing parallel compression economization improves the optimum COP of $CO_2$ transcritical refrigeration cycle by 47.3% for the chosen ranges over basis $CO_2$ transcritical refrigeration cycle. Present study reveals that the parallel compression with economizer is promising modifications to improve the transcritical $CO_2$ cycle performance.

## References

[1] M.H. Kim, J. Pettersen, C.W. Bullard, Fundamental process and system design issues in $CO_2$ vapor compression systems. Progress Energy Combustion Science 30 (2) (2004) 119–174.

[2] S. Yana Motta, P.A. Domanski, Impact of elevated ambient temperature on capacity and energy input to a vapour compression system – literature review, Letter report for ARTI 21-CR Research Project: 605–50010/605-50015.

[3] G. Lorentzen, Revival of carbon dioxide as a refrigerant. International Journal of Refrigeration 17 (5) (1994) 292–300.

[4] J. Sarkar, S. Bhattacharyya, M. Ramgopal, Transcritical $CO_2$ heat pump systems: exergy analysis including heat transfer and fluid flow effects. Energy Conversion and Management 46 (13–14) (2005) 2053–2067.

[5] E.A. Groll, J.H. Kim, Review of recent advances toward transcritical $CO_2$ cycle technology. HVAC&R Research 13 (3) (2007) 499–520.

[6] L. Cecchinato, M. Chiarello, M. Corradi, E. Fornasieri, S. Minetto, P. Stringari, C. Zilio, Thermodynamic analysis of different two-stage transcritical cycles. International Journal of Refrigeration 32 (5) (2008) 1058–1067.

[7] S.F. Pearson, Improved transcritical refrigeration cycle, European Patent EP1498667 2005.

[8] I. Bell, Performance increase of carbon dioxide refrigeration cycle with the addition of parallel compression economization, in: 6th IIR Gustav Lorentzen Natural Working Fluid, Glasgow UK, 29–1 September 2004, p.2/A/4.30.

[9] E. Torrella, R. Llopis, R. Cabello, Experimental evaluation of the inter-stage conditions of a two-stage refrigeration cycle using a compound compressor. International Journal of Refrigeration 32 (2009) 307–315.

[10] W.J. Zhang, C.L. Zhang, G.L. Ding, Transient modeling of an air-cooled chiller with economized compressor, part I: model development and validation. Applied Thermal Engineering 29 (11–12) (2009) 2396–2402.

[11] N. Agrawal, S. Bhattacharyya, J. Sarkar, Optimization of two-stage transcritical carbon dioxide heat pump cycles. International Journal of Thermal Science 46 (2) (2007) 180–187.

[12] J. Sarkar, S. Bhattacharyya, M. Ramgopal, Optimization of a transcritical $CO_2$ heat pump cycle for simultaneous cooling and heating applications. International Journal of Refrigeration 27 (8) (2004) 830–838.