Trade-off between Evaporator and gas cooler heat exchangers of a Transcritical CO_2 based Heat Pumps

J. Sarkar[1], R. R. Sahoo

Department of Mechanical Engineering
Institute of Technology, BHU, Varanasi, India 221005

Souvik Bhattacharyya, M. Ram Gopal

Department of Mechanical Engineering
Indian Institute of Technology, Kharagpur, West Bengal, India 721302

Abstract

Carbon dioxide as a working fluid for refrigeration and heat pump systems is increasingly importance in view of both ecological and economical aspects. The present investigation includes the theoretical work followed by numerical work for UA inventory. A simulation model has been developed to evaluate the system performance parameters for a transcritical carbon dioxide cycle for simultaneous heating and cooling. The simulated results are found to be in reasonable agreement with reported experimental results. This paper presents the simulation results of a transcritical CO_2 Heat Pump system for dairy applications where simultaneously cooling at 4 ˚C and heating at 73 ˚C are required. The optimal COP was found to be a function of the compressor speed, the evaporator water-inlet temperature and the gas cooler water-inlet temperature and discharge pressure. A study for optimally allocating the fixed total heat exchanger inventory between the evaporator and the gas cooler has been carried out. The COP of the system reached a maximum value and then decreased as the evaporator UA-value was monotonically increased and indicates optimum UA inventory between evaporator and gas cooler for different working condition and component specifications. These studies offer useful guidelines for an optimal system design.

[1] Corresponding author; E-mail: jahar_s@hotmail.com (J. Sarkar); Tel. 0542-2309083.

Keywords: CO_2 heat pump, Dairy system, System simulation, Heat exchanger's UA inventory, Optimization.

Introduction

With the discovery of the harmful effects of the synthetic refrigerants on environment, there is a renewed interest in natural refrigerants such as carbon dioxide. Lorentzen through his seminal studies has shown that the problem of low critical temperature of carbon dioxide can be effectively overcome by operating the system in the transcritical region. This has led to the development of transcritical carbon dioxide cycles where the condenser gets replaced by a gas cooler. It was found that the use of a gas cooler with heat rejection taking place over an unusually large temperature glide offers several unique possibilities such as simultaneous refrigeration and hot water heating/steam production, simpler control of capacity, etc. Environment friendliness, low price, easy availability, non-flammability, non-toxicity, compatibility with various common materials, compactness due to high operating pressures, excellent transport properties are cited as some of the reasons behind the revival of carbon dioxide as a refrigerant. Carbon dioxide based systems have shown strong potential in two sectors: mobile air-conditioning and heat pump applications. The unique feature of transcritical CO_2 cycle is the heat rejection taking place over an unusually large temperature glide in the gas cooler instead of condensation for other conventional cycles. This feature gives us opportunity to study the heat conductance inventory of evaporator and gas cooler to get higher performance. Over a last decade, researchers have been done theoretical works on heat conductance inventory. Bejan (1993) first reported the minimization of total heat exchanger inventory for endoreversible cycles with constant temperature heat reservoirs. Wu et al (1998) reported the optimal allocation of heat exchanger inventory for endoreversible cycles. El-Din (2001) has optimized both the conductance and area ratios for refrigeration and heat pump cycles with both finite and infinite heat reservoirs. Numerical validation of the analytical results is more valuable to the design engineer. Klein et al. (1998) numerically optimized the conductance ratio based on COP, cooling load and entropy generation. However, no such study was presented for transcritical CO_2 heat pump cycle.

In the present study, heat conduction inventory between evaporator and gas cooler of a transcritical carbon dioxide heat pump system for dairy applications have been presented; in such systems simultaneous cooling and heating at 4^{o}C and 73^{o}C, respectively, is required. The optimization results have been compared with the analytical formulation for irreversible heat pump cycle with variable temperature reservoirs. To study the optimally allocation of the fixed total heat exchanger inventory between the evaporator and the gas cooler, realistic data obtained from experiment have been used.

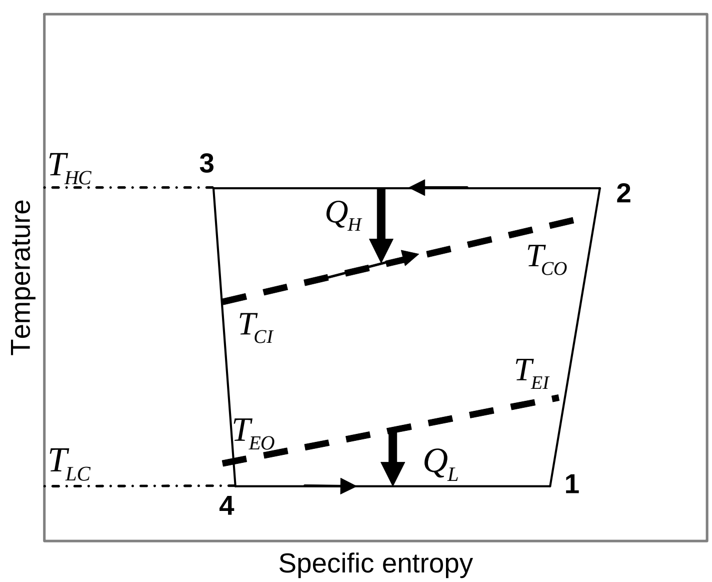

Fig. 1 Irreversible heat pump cycle with finite heat reservoirs

Analytical formulation

The T-s diagram of irreversible heat pump with finite capacity heat reservoirs (variable temperature heat reservoirs, i.e. single phase heat transfer for secondary fluids) is shown in Fig. 1. The heat conduction ratio ($x = UA_C / UA_E$) between evaporator and gas cooler can be optimized on the basis of performance parameters COP or cooling/heating load or the design parameter total conduction.

El-Din et al. (2001) obtained following expressions for optimum UA inventory:

For maximum heating load, $x_{opt} = \dfrac{\sqrt{I}}{\sqrt{I} + \sqrt{E_h / E_c}}$

For maximum cooling load, $x_{opt} = \dfrac{\sqrt{E_h / E_c}}{\sqrt{I} + \sqrt{E_h / E_c}}$

where, $E_c = \dfrac{1 - e^{-NTU_c}}{NTU_c}$, $E_h = \dfrac{1 - e^{-NTU_h}}{NTU_h}$, $I = \dfrac{\Delta S_H}{\Delta S_L}$

For endoreversible Carnot cycle, I = 1 and for ideal heat transfer processes in the heat exchangers, $E_H = E_L = 1$. So, for both refrigeration and heat pump applications, the optimum x reduced to $x = 1$ i.e. $UA_C = UA_E$. The same expression was earlier derived by Wu et al. (1998) for maximum value of heating load for same heat capacities of both reservoirs.

Transcritical CO_2 cycle simulation

The schematic diagram of a carbon dioxide based heat pump cycle for simultaneous heating and cooling is shown in Fig. 2 and corresponding temperature-entropy diagram is shown in Fig. 3. It can be noted that the CO_2 gas cooling process (2-3) is not isothermal and occurs above critical point. Water is supplied as the heat exchanger fluid to both gas cooler and evaporator. The temperature of water at gas cooler outlet is assumed at 73^{o}C and it is 4^{o}C at evaporator outlet as is typically required in dairy plants. Both these heat exchangers are of double-pipe counter flow type. The following assumptions have been made in the analysis: saturated refrigerant outlet in evaporator, compression process is adiabatic but not isentropic and pressure drop in both evaporator and gas cooler are negligible. The entire system has been modelled based on energy balance of individual components yielding conservation equations presented below.

The refrigerant mass flow rate through the compressor suction volume V_s is given by,

$$\dot{m}_r = \rho_1 \eta_v V_s \frac{N}{60} \tag{1}$$

Where, N = speed and ρ_1 and V_s are suction density and suction volume respectively.

The isentropic efficiency of the compressor in term of enthalpy is given by,

$$\eta_{is,c} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{2}$$

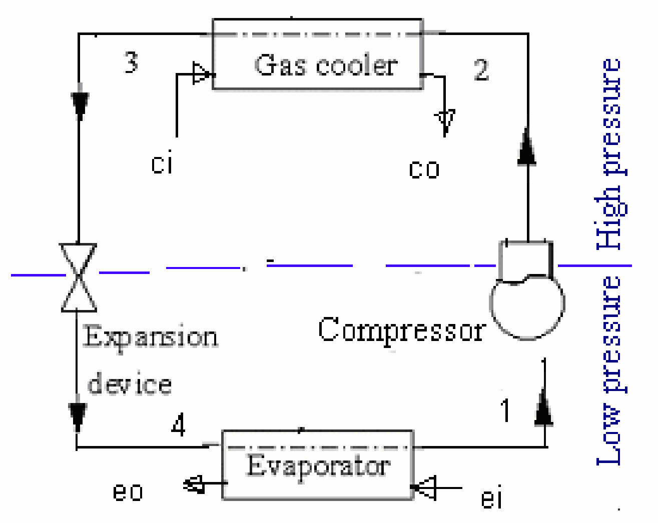

Fig. 2 Transcritical CO_2 heat pump for water cooling and heating

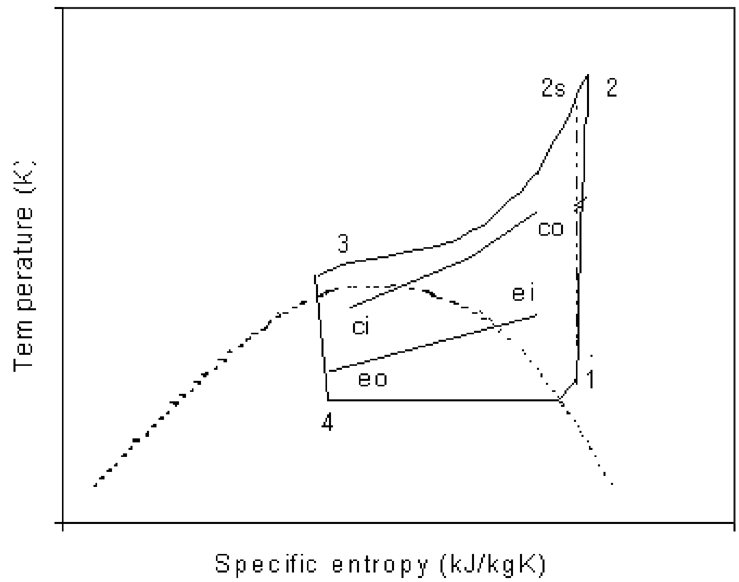

Fig. 3 T-s diagram of transcritical CO_2 heat pump

Specifications of a Dorin compressor (Model TCS113) have been used in the simulation and the following correlation have been used for volumetric and isentropic efficiency respectively, in term of pressure ratio, based on regression of experimental data:

$$\begin{aligned} \eta_v &= 1.1636 - 0.2188\, r_p + 0.0163\, r_p^{\,2} \\ \eta_{\mathrm{is,c}} &= 0.61 + 0.0356\, r_p - 0.0257\, r_p^{\,2} + 0.0022\, r_p^{\,3} \end{aligned} \tag{3}$$

Employing LMTD expression in evaporator,

$$Q_L = UA_E \frac{(T_{EI} - T_1) - (T_{EO} - T_4)}{\ln\left(\dfrac{T_{EI} - T_1}{T_{EO} - T_4}\right)} \tag{4}$$

Energy balance in evaporator for both the fluids yield:

$$Q_L = \dot{\mathrm{m}}_{\mathrm{r}}(h_1 - h_4) = \dot{\mathrm{m}}_{\mathrm{w,E}}\, c_{pw}(T_{EI} - T_{EO}) \tag{5}$$

Employing LMTD expression in gas cooler,

$$Q_H = UA_C \frac{(T_2 - T_{CO}) - (T_3 - T_{CI})}{\ln\left(\dfrac{T_2 - T_{CO}}{T_3 - T_{CI}}\right)} \tag{6}$$

Energy balance in gas cooler for both the fluids yield:

$$Q_H = \dot{m}_r (h_2 - h_3) = \dot{m}_{w,c} c_{pw} (T_{co} - T_{ci}) \tag{7}$$

The expansion process is considered to be isenthalpic, yielding:

$$h_4 = h_5 \tag{8}$$

The compressor discharge pressure or gas cooling pressure (in bar) is set at the optimum value to get maximum COP and is given by (Sarkar et al., 2004):

$$P_{opt} = 4.9 + 2.256 t_3 - 0.17 t_1 + 0.002 t_3^{\,2} \quad (\text{t in } ^{o}\text{C}) \tag{9}$$

A simulation model has been developed based on the above equations to find the state points and performance. For the thermodynamic properties of CO_2, an exclusive code *CO2PROP* based on available correlations (Span and Wagner 1996) has been developed and employed. The code solves the system model equations by suitable iterative method to get maximum accuracy integrated with the subroutine code CO2PROP. The performance parameter COP in term of cooling and heating load is evaluated by,

$$COP = \frac{Q_L}{Q_H - Q_L} \tag{10}$$

To compare the optimization results with analytical model, a thermodynamic average temperature for process 2-3 has been used, which is given by following:

$$T_{HC} = \left(h_2 - h_3\right) / \left(s_2 - s_3\right) \tag{11}$$

and the processes are same as analytical model. By effective iterative method the optimum value has been evaluated.

Result and discussion

The simulation results of heat conduction inventory between evaporator and gas cooler of a transcritical carbon dioxide heat pump system are presented for simultaneous water cooling and heating at 4°C and 73°C, respectively, required in dairy applications. A Dorin compressor (model TCS113) with specifications of 2.2 m^3/h displacement rate @ 2900 rpm has been used for this simulation. The compressor has been tested in Refrigeration and air-conditioning laboratory, Department of Mechanical Engineering, IIT Kharagpur, to evaluate the performance. Fig. 4 shows the variation of isentropic efficiencies with pressure ratio for different evaporator pressures, which is based on the experimental study conducted for CO2 hear pump system with simultaneous water heating and cooling. The result shows the equation used for simulation give fair agreement with tested values with maximum deviation of 5%. Although the value of total UA depends on the plant capacity, 900 W/K has been used in this simulation, which was used by Byon et al. (2000) in mobile A/C system model. Experimental study shows that this value is quite reasonable.

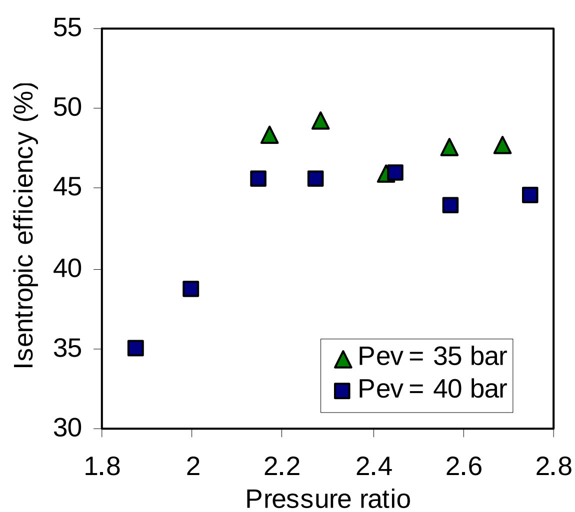

Fig. 4 Variation of compressor isentropic efficiency with pressure ratio

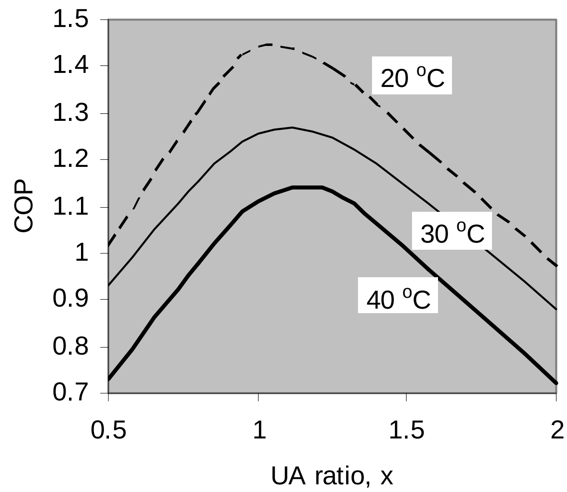

Fig. 5 Variation of cooling COP with UA ratio with different water inlet temperatures

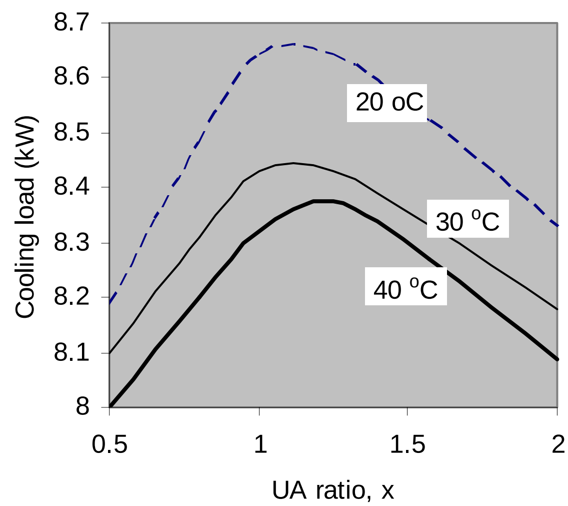

Fig. 6 Variation of cooling load with UA ratio with different water inlet temperatures

Fig. 5 shows the variation of cooling COP with UA ratio with water inlet temperatures of 20 ^{o}C, 30 ^{o}C and 40 ^{o}C for compressor speed of 2900 rpm. The optimum values are varies within 1.1 to 1.15, whereas the lower water inlet temperature gives closer value to 1. This is due to the fact of lower temperature glide in gas cooler at lower water inlet temperature as well as better heat transfer properties of refrigerant at lower temperature and pressures. Fig. 6 shows the variation of cooling loads with UA ratio with same values of water inlet temperatures and compressor speed. The optimum values are varies within same range, whereas the lower water inlet temperature gives closer value to 1. The maximum deviation of numerical optimum values of UA ratio with analytical optimum values is 10%. Fig. 7 shows the variation of cooling COP with UA ratio with compressor speeds of 2500 rpm and 2900 rpm for water inlet temperature of 30 ^{o}C. The optimum values are varies within 1.1 to 1.18, whereas the lower compressor speed gives closer value to 1. Above results show that the transcritical cycle deviates more with analytical model compared to conventional heat pump cycles, optimum values of UA ratio are close to 1.

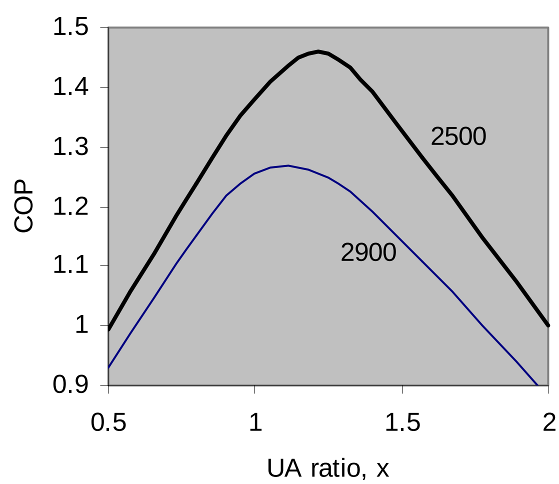

Fig. 7 Variation of cooling COP with UA ratio with different water inlet temperatures

Conclusions

Heat conduction inventory between evaporator and gas cooler of a transcritical carbon dioxide heat pump system for dairy applications have been presented in this study. The optimization results have been compared with the analytical formulation for irreversible heat pump cycle with variable temperature reservoirs. To study the optimally allocation of the fixed total heat exchanger inventory realistic data obtained from experiment have been used. Results show that optimum values of x are varies within 1.1 to 1.15 for both maximum COP and load, whereas the lower water inlet temperature, due to lower temperature glide in gas cooler, gives closer value to 1. The maximum deviation of numerical optimum values of UA ratio with analytical optimum values is 10%. The lower compressor speed gives x value closer to 1. Comparison with literature shows that the transcritical cycle deviates more with analytical model compared to conventional heat pump cycles.

References

Bejan A. Power and refrigeration plants for minimum heat exchanger inventory. J Energ Res Tech, 115; 1993; 148-150.

Wu C, Chen L, Sun F. Optimization of steady flow heat pumps. Energ Convers Mgmt, 39; 1998; 767-773.

Al-Din MMS. Performance analysis of heat pumps and refrigerators with variable resorvoir temperatures. Energ Convers Mgmt, 42; 2001; 201-216.

Klein SA, Reindl DT. The relationship of optimum heat exchanger allocation and minimum entropy generation rate for refrigeration cycles. J Energ Res Tech, 20; 1998; 172-178.

Byon J.S, Domm P.A. Semi-theoretical simulation model for transcritical carbon dioxide mobile A/C system, ASHRAE Transactions, 2000; 1576-1586.

Sarkar J, Bhattacharyya S, Ramgopal M. Optimization of a transcritical CO_2 heat pump cycle for simultaneous cooling and heating applications, *Int J Refrig,* 27(8); 2004; 830-838.

Span R, Wagner W. A new equations of state for Carbon dioxide covering the fluid region from triple point temperature to 1100 K at pressure up to 800 MPa, J Phys Chem Ref Data, 25; 1996; 1509-1596.