Comparative Study of Transcritical CO_2 Cycle with and Without Suction Line Heat Exchanger at High Ambienttemperature

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Abstract

Global warming and ozone depletion potential of chemical refrigerants have motivated to develop the refrigeration systems with natural refrigerants. Transcritical CO_2 refrigeration cycle losses its performance at higher ambient temperature due to lower critical temperature of CO_2. This paper discusses the comparative analysis of the transcritical cycle with and without suction line heat exchanger. At higher ambient conditions, the use of suction line heat exchanger improves the cycle performance by 2 to 4%. Due to more heat transfer rate the specific refrigeration capacity, mass flow rate and compressor power consumption. The performance of cycle does not improve significantly with increase in the effectiveness of the suction line heat exchanger.

Key Words: Transcritical cycle; CO_2; suction line heat exchanger; cycle simulation; gascooler; refrigerant

1. Introduction

Environmental issues have enforced all the researchers to find sustainable permanent alternative solutions to the current working fluids used in the refrigeration and air conditioning systems. Prof. Gustav Lorentzen through his research on natural refrigerant has given rebirth to CO_2,which was very popular in the late nineteen and early twenty century [7]. CO_2 is available in ample quantity in the nature and is not a toxic and inflammable refrigerant. It has very good thermo-physical and transport properties. CO_2 is being used in the heat pumps, display cabinets, and mobile air conditioners in the low ambient European countries. This paper discusses about the use of CO_2 in the air conditioning system working at high ambient conditions witnessed in the subtropical and tropical countries like India.

Few researchers have studied the effect of suction line heat exchanger (SLHX) on the energy performance of transcritical CO_2 cycle for low ambient temperatures and mobile air conditioning systems. Torrella et al. have evaluated the performance of transcritical CO_2 refrigeration system with internal heat exchanger for the evaporator saturation temperatures -5^{o}C, -10^{o}C, -15^{o}C and gascooler outlet temperature 31^{o}C and 34^{o}C. They have concluded that use of internal heat exchanger increases the cooling capacity and COP by 12% [3]. The authors have not covered higher evaporator temperature range 0 to 5^{o}C and ambient temperatures above 35^{o}C. Apreaetal.have studied the transcritical CO_2 refrigeration system with internal heat exchanger for ambient temperature range 25 to 40^{o}C and concluded 10% rise in the COP of the system at higher ambient temperatures [2]. Kim et al have studied the use of internal heat exchanger in transcritical CO_2 water heating system and concluded that optimum discharge pressure, which gives maximum capacity and COP decreases with increase of length of heat exchanger and noticed slight increase in the COP of the system with internal heat exchanger [8]. The Klein et al. have reported the influence of SLHX on the cycle performance for chemical refrigerants like R410a, R22, R134a, etc. excluding CO_2. and concluded there is 13% to 53% gain in the cooling capacity the subcritical refrigeration cycle [5]. Boewe et al. have found the gain in the capacity for the mobile CO_2 air conditioning system at higher ambient conditions [1].

2. Simulation Parameters

The cycle simulation for transcritical CO_2 system has carried out using numerical program Engineering Equation Solver (EES) [6]. Table 1 presents the input operating parameters considered for the cycle simulation.

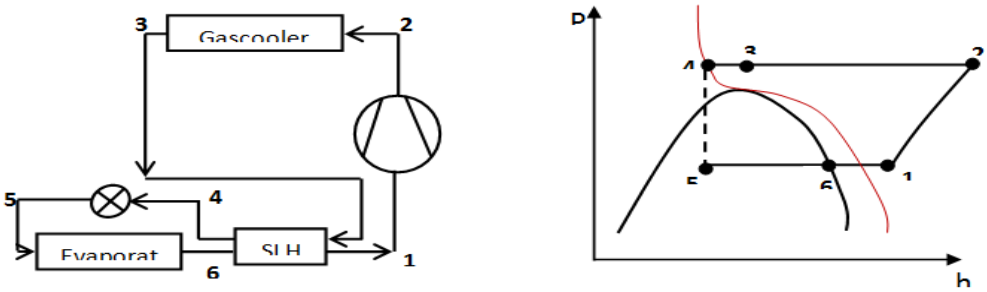

Figure 1. Schematic layout and P-h chart of transcritical CO_2 air conditioning system

The heat balance equations used in simulation are given in the Table 2. These are presented with their corresponding input and output parameters. The assumptions considered in the analysis are;

- steady state condition
- pressure drop losses in the heat exchangers and the connecting hoes lines are neglected
- convective and radiation heat losses from the components are neglected

Table 1. Input parameters for cycle simulation

Evaporator cooling load, kW	5.2
Compressor speed, rps	20
Isentropic efficiency of the compressor, [%]	70
Volumetric efficiency of the compressor, [%]	70
Evaporator saturation temperature, [^{o}C]	5
Evaporator useful superheat, ^{0}C	5
Suction line heat ingress, %	10
Overall heat transfer coefficient for evaporator, U_oA_o, [W/k]	120
Overall heat transfer coefficient for gascooler, U_oA_o, [W/k]	310
Ambient temperature range, [^{o}C]	30 - 45

As shown in Figure 1, due to 'S' shaped isothermal lines in the case of CO2, the performance of transcritical cycle depends on optimization of gascooler pressure and temperature.

Table 2: Mathematical heat balance models for the components

Component	**Equations**	**Input**	**Output**
Compressor	$m^{\cdot}{}_{\downarrow}(r\ ) = V\ \rho_{\downarrow}r\ \ N\ (_{\downarrow}v$ $W_{\downarrow}actual = m^{\cdot}{}_{\downarrow}(r\ )\ (\boldsymbol{h}_{\downarrow}(2, ise) - \boldsymbol{h}$ $[\boldsymbol{h}_{2,act} = \boldsymbol{h}_{2,sat} + C_{p_r}(T]_{r,2} - T_{2,sa}$ $S_{1,ise} = S_{2,ise}$	VV N ρ_r $(_{\downarrow}v$ $(_{\downarrow}ise$ $\boldsymbol{h}_2$ N ρ_r	$\dot{m}\ r$ W_{actual} $T_{r,2}$
Evaporator	$Q_a = \dot{m}_a . C_{p_a}\left(T_{a_o} - T_{a_i}\right)$ $Q_a = \dot{m}_r\ (\boldsymbol{h}_6 - \boldsymbol{h}_5)$ $Q_{max} = \ \dot{m}_r\ C_{p_r}\ \left(T_{r_i} - T_{a_i}\right)$ $Q_{max} = {Q_a}/{\varepsilon}$ $\varepsilon\varepsilon = \mathbf{1} - \exp(-NTU)$ $U_oA_o = \left(\dot{m}C_p\right)_{min}\ \ NTU$	U_oA_o $\dot{m}_r$ T_{r_i} $\dot{m}_a$	Q_a $\boldsymbol{Q}_a$ $\boldsymbol{h}_6$ $\boldsymbol{T}_6$
Suction Line Heat exchanger	$\varepsilon_{SX}\ (\boldsymbol{h}_3\boldsymbol{h}_6) = (\boldsymbol{h}_1 - \ \boldsymbol{h}_6) = \ (\boldsymbol{h}_3 -$	ε_{SX} $\boldsymbol{h}_3$ $\boldsymbol{h}_6$	$\boldsymbol{h}_1$ $\boldsymbol{h}_4$
Gascooler	$Q_a = \dot{m}_a . C_{p_a}\left(T_{a_o} - T_{a_i}\right)$ $Q_a\ Q_{max} = {Q_a}/{\varepsilon}$ $\varepsilon = 1 - exp\left[\frac{NTU^{0.22}}{C_r}\left\{exp\left(-C_r.NTU^{0.78}\right) - 1\right\}\right.$	U_oA_o U_oA_o $\dot{m}_r$ T_{r_i} $\dot{m}_a\ \ T_{r_i}$ $\dot{m}_a$	Q_a Q_a $\boldsymbol{h}_3$ T_3

	$C_r = \frac{\left(\dot{m}C_p\right)_{min}}{\left(\dot{m}C_p\right)_{max}}$ $U_oA_o = \left(\dot{m}C_p\right)_{min} \quad NTU$		
Expansion valve	$h_4 = h_5$	h_4	h_5
Cycle	$COP = \frac{Q_a}{W_{actual}}$ $COP_{carnot} = \frac{T_{r,evap,sat}}{T_{r,gc,sat} - T_{r,eva,sat}}$	Q_a W_{actual}	COP COP_{carnot}

3. Results and Discussion

The results of the cycle simulation with and without suction line heat exchanger are presented in the form of figures.

3.1 Coefficient of Performance (COP)

Figure 3 shows variation of COP with gascooler exit pressure at various ambient temperatures. For ambient temperature more than 35°C, the use of SLHX improves the COP of the cycle by 2 to 4% as compared to cycle without SLHX. The improvement in COP is because of more specific heat rejection in the SLHX hence more refrigeration capacity. The use of SLHX effects in higher specific power consumption. However, the specific power consumption is less than specific heat rejection in the cycle using SLHX.

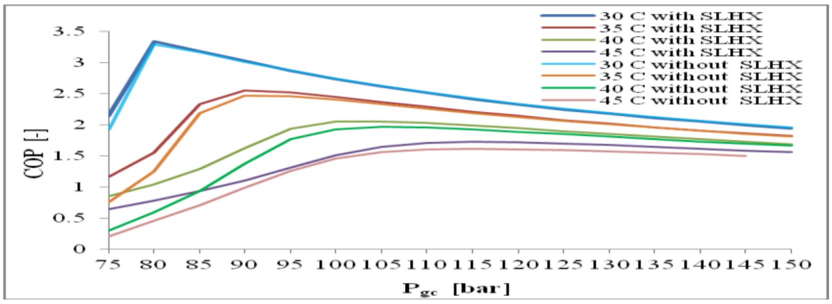

Fig. 3 Effect of gascooler pressure on the COP for different ambient conditions

Figure 4 depicts that use of SLHX improves the COP in the range 53.6% to 55.2% for rise in evaporator saturation temperature from -10°C to 5°C at ambient temperature 35°C. For ambient temperature 45°C, this change is from 40.40% to 41.38%. At 35°C ambient and 5°C evaporation temperature, COP with SLHX increases by 3.24% than that of without SLHX. For ambient 45°C, it increases by 56.30%.

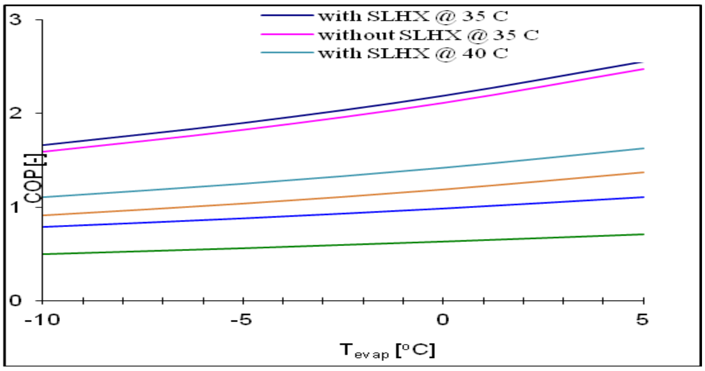

Fig. 4: Effect of evaporator temperature on COP for different ambient conditions

3.2 Gascooler Capacity

Figure 5 shows the variation of gascooler capacity for a range of evaporating temperatures. In case of CO_2transcritical system, the heat rejection capacity of the gascooler is very much sensitive to evaporation temperature and ambient temperature. For ambient temperatures 45°C and 35°C, the heat rejection capacity in the gascooler with SLHX is 29.28% and 1.56% lower than that of without SLHX respectively.

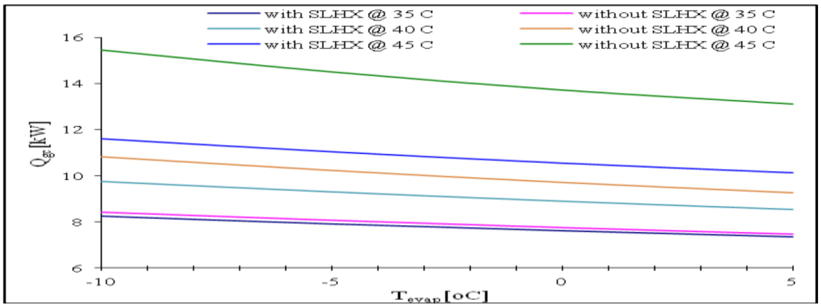

Fig. 5: Evaporator temperature versus actual heat rejection in the gascooler

3.3 Power Consumption

Figure 6 shows the power consumption for a range of evaporating temperature. The power consumption decreases with rise in evaporation temperature at all cases. For ambient 45°C, the power consumption decreases in the range 63.5% to 93% with SLHX than that of without SLHX for -10°C to 5°C change in evaporator saturation temperature. In the case of ambient 35°C, this change is in the range 11% to 22.7%. This indicates that use of SLHX has more advantage at higher ambient as compared to lower ambient.

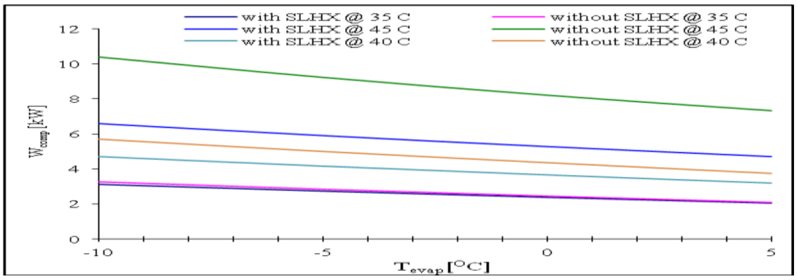

Fig. 6: Effect of evaporator temperature on actual compressor power consumption

3.4 Dryness fraction

Figure 7 shows the effect of SLHX on the inlet quality of refrigerant at evaporator. The dryness fraction of refrigerant decreases in the range of 30% to 27% with SLHX than that of without SLHX for -10°C to 5°C change in evaporator saturation temperature for ambient temperature 35°C.

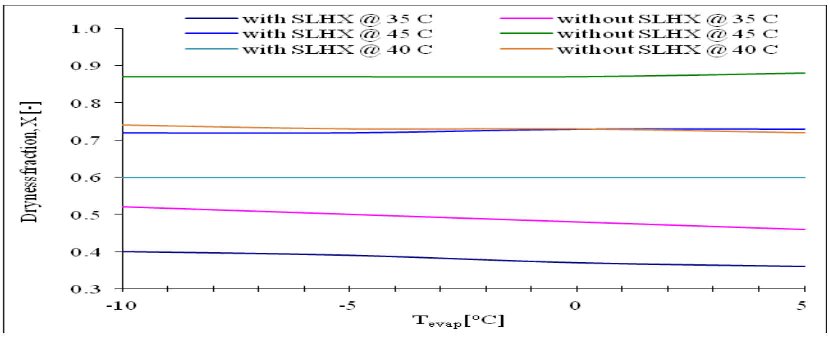

Fig. 7: Evaporator temperature versus dryness fraction for different ambient conditions

4. Conclusions

The performance of transcritical CO_2 refrigeration cycle depends on the ambient temperature and its corresponding optimized gascooler pressure. The use of suction line heat exchanger improves the COP in the range 2 to 4% for ambient temperature more than 35°C. For the evaporator temperature range -10°C to 5°C, the use of SLHX improves the COP in the range 50 to 55% for ambient temperature range 35°C to 45°C. The compressor power consumption decreases in the range of 60 to 95% at ambient temperature 45°C and 11 to 22%

at ambient temperature 35^{o}C with SLHX. The share of liquid at evaporator inlet improves by 27 to 30% due to extra cooling of the gas at the gascooler outlet. The use of SLHX in the transcritical CO2 system operating at high ambient conditions has major benefits in terms of the performance parameters of the system.

5. Nomenclature

Q	Actual heat flow [kW]	*Greek letter*	
A	Area [m^2]	å	Heat exchanger effectiveness [-]
COP	Coefficient of Performance [-]	*Subscripts*	
W	Compressor power [kW]	r	refrigerant
Cr	Heat capacity ratio [-]	v	volumetric
m	Mass flow of refrigerant [kg/s]	ise	Isentropic process
NTU	Number of transfer units	a	air
U_o	Overall heat transfer coefficient [W/m^{2}K]	max	maximum
h	Specific enthalpy [kJ/kg]	SX	Suction line heat exchanger
S	Specific entropy [kJ/kg]	sat	Saturation condition
C_p	Specific heat of refrigerant [kJ/kg K]	gc	gascooler
N	Speed of compressor [rps]	evap	evaporator
T	Temperature of refrigerant [^{o}C]	i	Inlet
V	Volume flow capacity of compressor [m^3]	o	Outlet

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