EXPERIMENTAL INVESTIGATION OF TRANSCRITICAL CO_2 HEAT PUMP FOR SIMULTANEOUS WATER COOLING AND HEATING

by

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This paper presents the experimental performances of a transcritical CO_2 heat pump prototype for simultaneous water cooling and heating applications. System behavior and performances such as cooling capacity, heating capacity, and system coefficient of performance (COP) have been studied experimentally for various operating parameters such as water mass flow rate, water inlet temperature for both evaporator and gas cooler, and expansion valve opening. Performance is also compared with previous test data. Test indicates that the expansion valve opening has a significant effect as well near the full valve closing condition (up to 20°). Study shows that both the water mass flow rate and inlet temperature have significant effect on system performances. Test results show that, at gas cooler pressure of 90 bar, the effect of evaporator water mass flow rate on the system performances is more pronounced (COP increases 0.6 for 1 kg/min.) compared to the gas cooler water mass flow rate (COP increases 0.4 for 1 kg/min.) and the effect of gas cooler water inlet temperature is more significant (COP decreases 0.48 for given ranges) compared to the evaporator water inlet temperature (COP increases 0.43 for given ranges).

Key words: *CO_2 heat pump, transcritical, gas cooler, water cooling and heating, valve opening*

Introduction

Recently, natural working fluid CO_2 has became a promising alternative particularly in heat pump applications due to its performance related various advantages and eco-friendliness. Some of the current theoretical and experimental investigations on several heat pump applications of transcritical CO_2 cycle have been presented by Neksa [1] and Kim *et al.* [2].

Neksa *et al.* [3] and White *et al.* [4] experimentally investigated the effects of discharge pressure, water inlet and outlet temperatures on the heat pump water heater perfor-

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mances. Hwang *et al.* [5] experimentally showed that the actual CO_2 cycle performed similar to the R-22 cycle when the same outside volume of the heat exchanger was employed for both refrigerants. Yarral *et al.* [6] experimentally investigated to effect of discharge pressure on CO_2 heat pump performance for simultaneous production of refrigeration and water heating to 90 °C for the food processing industry. Adriansyah [7] experimentally studied the effect of discharge pressure for simultaneous air-conditioning and water heating. Stene [8] presented the effect of water inlet temperature for combination of space and water heating. Cho *et al.* [9] studied the performance of the CO_2 heat pump by varying the refrigerant charge amount at standard cooling condition to show the importance of refrigerant charge to achieve better performance. Kim *et al.* [10] have done experimental study on CO_2 heat pump to study the effect of internal heat exchanger using water as secondary fluid for both sides with emphasizes only on heating. Cabello *et al.* [11] experimentally evaluated the energy efficiency and optimal gas-cooler pressures of a single-stage transcritical refrigerating plant working with carbon dioxide and showed that Sarkar *et al.* [12] correlation matches best with the test data of optimal gas-cooler pressures. However, effects of expansion valve opening and water mass flow rate on system behaviours are scarce.

In the present investigation, experimental results on the working prototype of a transcritical CO_2 heat pump system for simultaneous water cooling and heating are presented. The cooling and heating capacities and system coefficient of performance (COP) have been studied for various operating conditions (water mass flow rates and water inlet temperatures of both evaporator and gas cooler) and expansion valve openings. Performance is also compared with previous test data [10].

Experiments of a transcritical CO_2 heat pump

Description of test setup

The working prototype of transcritical CO_2 heat pump system for simultaneous water cooling and heating has been developed based on numerical simulation [13] for a rated cooling capacity of 3.5 kW. Test facility layout of transcritical CO_2 heat pump for simultaneous water cooling and heating with instrumental positions is shown in fig. 1. Stainless steel was chosen as the material for all system components. A Dorin CO_2 compressor (model TCS113: displacement of 2.2 m^3 per hour and rated speed of 2900 rpm) was chosen for the experimental investigation. On the basis of minimum and maximum pressure ratios of 80/50 and 120/26 bar/bar, respectively, a Swagelok integral bonnet needle

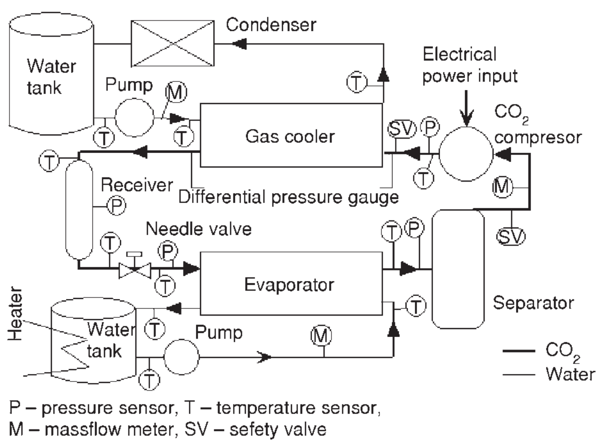

Figure 1. Test facility layout of the transcritical CO_2 heat pump

valve (model SS-1RS4) was used as the expansion device, which can be used regulate flow rate and degree of superheat. The separator and receiver were designed for a total volumetric capacity of 8 L and 2 L, respectively. A fan-coil cooling unit including a fan and a storage tank was employed for a heat transfer rate of 6 kW to cool the warm water to its initial temperature at the inlet to the gas cooler. A water bath with heater and pump was incorporated in the evaporator to supply water at constant temperature and flow rate. The evaporator and the gas cooler are counter-flow tube-in-tube heat exchangers, where CO_2 flows in the inner tube and water in the outer annulus (tab. 1). Measuring ranges of instruments with uncertainties are listed in tab. 2.

Table 1. Dimensions of gas cooler and evaporator

Heat exchangers	Gas cooler	Evaporator
Configuration	Coaxial, single pass, 14 rows	Coaxial, single pass, 9 rows
Inner tube OD/outer tube OD	6.35/12 mm	9.5/16 mm
Total length of tubing	14 m	7.2 m

Table 2. Ranges and uncertainties of measuring instruments

Parameters	Measuring instruments	Ranges	Accuracy
Pressure	Dial pressure gauge	0-160 bar	±1.5% of full range
Pressure loss	Differential pressure gauge	0-4 bar	±1.5% of full range
CO_2 mass flow rate	Coriolis mass flow meter	0.2-10 kg/min.	±0.1% of full range
Water mass flow rate	Mass flow meter	0.5-20 kg/min.	±0.5% of full range
Temperature	Thermocouples (T-type, K-type)	Calibrated range: 0-150 °C	±0.5 °C
Power input	Power meter	–	±0.1 kWh

Test procedure and test conditions

In the experimental study, the effects of water inlet temperature and mass flow rate in gas cooler, and water inlet temperature and mass flow rate in evaporator were investigated by varying them using heat dissipation unit for the gas cooler and heating unit for the evaporator. Suction pressure and discharge pressure were set at required level by simultaneous control of the total mass of CO_2 (refrigerant charge) in the system and degree of opening of the expansion device before varying the external parameters. The total refrigerant mass in the system was controlled by adding CO_2 from a high pressure cylinder or by venting it through the safety valve. For certain test conditions, constant water flow rates for both evaporator and gas cooler were maintained by pumps, inlet water temperature to gas cooler was maintained by controlling fan speed and water inlet temperature to evaporator was maintained by heater control. The compressor power input was measured by using a power meter, the refrigerant mass flow rate was measured by a Coriolis effect flow meter, the pressure of the refrigerant were monitored by using pressure transducers, pressure drop in the heat exchangers was measured by differential pressure transducer and refrigerant and water temperatures at all required locations were measured by using T-type and K-type thermocouples. All the measurements have been done at steady-state condition. The principal system performance parameters under steady-state, namely, power input to the compressor, cooling capacity, heating capacity, and system COP have been computed from the measured data. The uncertainties of cooling capacity, heating capacity, and system COP, estimated by error analysis, are approximately ±5%, ±5%, and ±6%, respectively. Repeatability tests were conducted of various sets of operating parameters (one such test result is shown in fig. 2) and showed that most of the data points for system COP are within the uncertainty ranges ( ) of the test loop measurements in most cases [14]. Figure 3 shows the variations of cooling capacities in refrigerant and water side with discharge pressure at suction pressure of 35 bar. Mass flow rate of water and its inlet temperature in gas cooler and evaporator are kept constant at 1.5 and 2 kg/min. and 33 and 29 °C, respectively. The comparision between cooling capacities in refrigerant and water show that the maximum deviation is 11%, which is within the maximum heat gain in the evaporator of 15%.

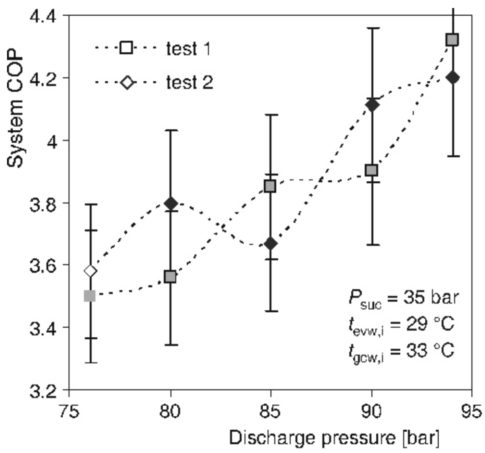

Figure 2. Repeatability analysis for suction pressure of 35 bar

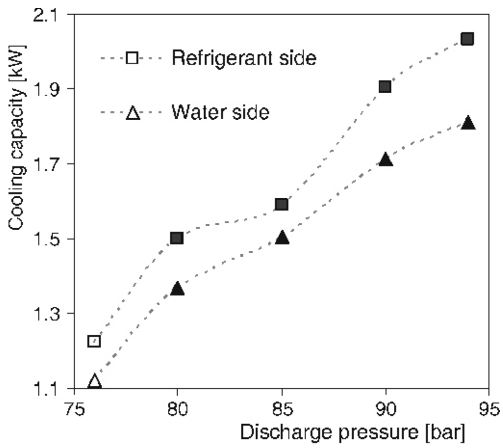

Figure 3. Comparison between cooling capacity in refrigerant and water

Results and discussion

The performance of the CO_2 heat pump system in terms of cooling or heating capacity and system COP (cooling + heating outputs divided by compressor power) considering both cooling and heating as useful outputs are studied for various valve opening, water inlet temperatures and mass flow rates. Maximum measured cooling and heating capacities (not necessarily simultaneous) of the system have been recorded as 3 kW and 5 kW, respectively. The trend for experimental data of this study matches fairly well with a previously reported experimental data [10], although, absolute COP values are relatively lower (by about 10%); this could be attributed to use of internal heat exchanger in previous case and the prevailing difference in test conditions and the test facilities. Steady-state performance of the system is presented in terms of system COP.

Variation of suction and discharge pressures, and refrigerant mass flow rate for certain refrigerant charge with various valve openings of the needle valve at steady-state are shown in fig. 4. Mass flow rate of water and its inlet temperature in gas cooler and evaporator are kept constant at 1.5 and 2 kg/min. and 32.3 and 32.9 °C, respectively. At smaller valve openings, the pressure difference across the compressor is much larger due to accumulation of CO_2 in the gas cooler and the corresponding reduction of CO_2 in the evaporator. As the valve opens from the completely closed position (measured in angle turned from the closed position), initially refrigerant mass flow rate increases rapidly due to large reduction in pressure ratio which further slows down consequently refrigerant mass flow as well. Valve opening plays a strong role at a nearly closed position (nearly up to 20°). The variation of other parameters with valve opening (refrigerant charge and other constant parameters are similar as in fig. 4) as shown in fig. 5 shows that the system gives maximum heating outlet temperature and minimum cooling outlet temperature for some optimum valve opening and the optimal values can vary between 20 to 40° depending on the operating conditions.

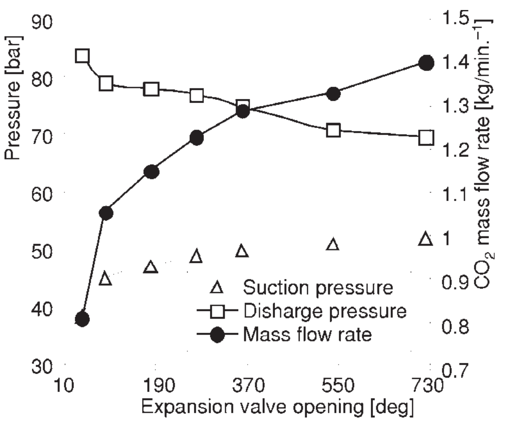

Figure 4. Variation of suction and discharge pressure and mass flow rate with expansion valve opening

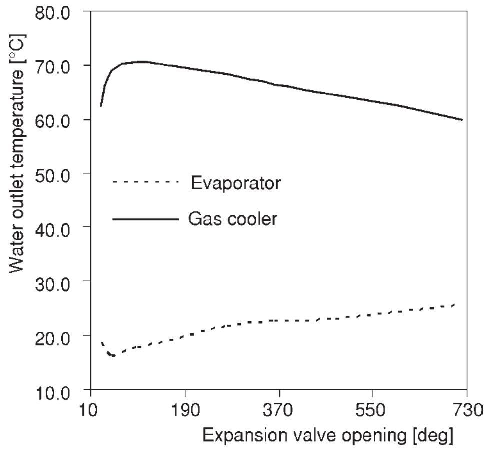

Figure 5. Variation of water outlet temperatures with expansion valve opening

Effects of water mass flow rate to evaporator on the performances are shown in fig. 6, for suction pressure of 40 bar, discharge pressure of 90 bar, evaporator water inlet temperature of 29 °C, gas cooler water inlet temperature of 33 °C, and water mass flow rate of 1 kg/min. With increase in water mass flow rate to evaporator, the cooling capacity increases due to increase in water side heat transfer coefficient and both the heating capacity and compressor work increase modestly due to minor increase in the suction temperature (increase in degree of superheat from 15 to 16.2 °C) and also discharge temperature from 124 to 126.5 °C. Water outlet temperatures of both evaporator and gas cooler increase with increase in water mass flow rate to evaporator.

Effects of water mass flow rate to gas cooler on the performances are shown in fig. 7, for suction pressure of 40 bar, discharge pressure of 90 bar, gas cooler water inlet temperature of

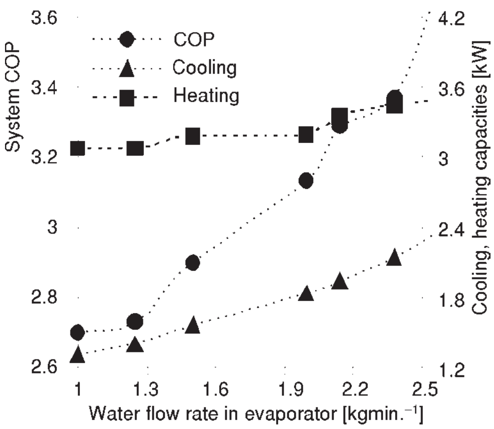

Figure 6. System performance variation with evaporator water flow rate

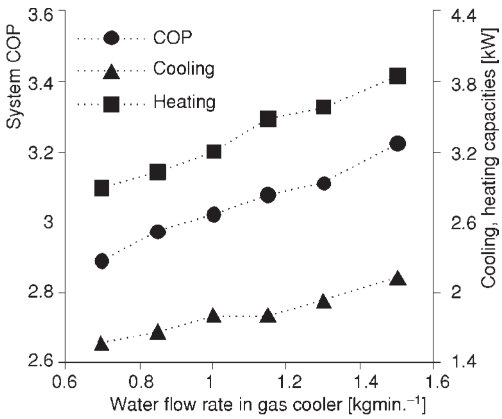

Figure 7. Variation of system performance with gas cooler water flow rate

33 °C, evaporator water inlet temperature of 29 °C, and water mass flow rate of 1.5 kg/min. With increase in water mass flow rate in gas cooler, heating output increases modestly due to increase in water side heat transfer coefficient and refrigerant mass flow rate increases, which enhances the cooling capacity, although the effect on compressor work is insignificant and hence the system COP increases. With increase in water mass flow rate to gas cooler, water outlet temperatures of both evaporator and gas cooler decrease. Results show that the effect of water flow rate in evaporator is more pronounced on the performances (COP increases about 0.6 per 1 kg/min.) compared to the effect of water mass flow rate in gas cooler (COP increases about 0.4 per 1 kg/min.).

Figure 8 exhibit the variations of performances with evaporator water inlet temperature for suction pressure of 40 bar, discharge pressure of 90 bar, mass flow rates of water in gas cooler and evaporator of 1 kg/min. and 1.5 kg/min., respectively, and water inlet temperature to gas cooler of 33 °C. With the increase in water inlet temperature to evaporator, suction temperature as well as discharge temperature increases, which enhance both cooling and heating capacities; however compressor work increases insignificantly due to decrease in refrigerant mass flow rate and hence system COP increases. Water outlet temperature of evaporator increases due to dual effect of increase in cooling capacity and water inlet temperature; whereas water outlet temperature of gas cooler increases due to increase in heating capacity.

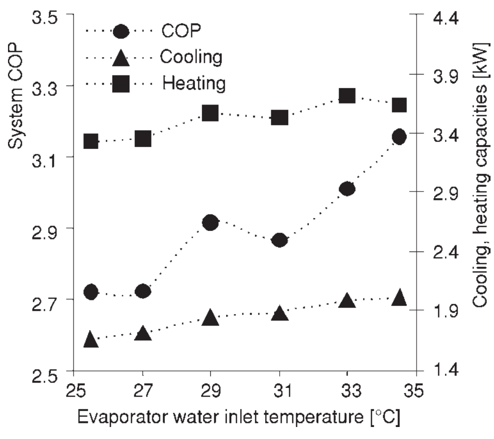

Figure 8. Variation of system performance with evaporator water inlet temperature

Figure 9 exhibits the variation of performances with water inlet temperature to gas cooler for suction pressure of 40 bar, discharge pressure of 90 bar, mass flow rates of water in gas cooler and evaporator of 1 kg/min. and 1.5 kg/min., respectively, and water inlet temperature to evaporator of 29 °C. As the water inlet temperature to gas cooler increases, the heating capacity decreases due to deterioration in heat transfer properties of CO_2 in gas cooler and the refrigerant exit temperature in gas cooler increases, which increases the vapor quality in evaporator inlet and hence cooling output also decreases, however, compressor work is nearly invariant due to negligible change in refrigerant mass flow rate and degree of superheat and hence system COP decreases significantly. With increase in water inlet temperature to gas cooler, water outlet temperatures of both evaporator and gas cooler increase. Results show that the effect of water inlet temperature to gas cooler on the system performances is more significant (COP decreases about 0.48 per 10 °C) compared to the effect of water inlet temperature to evaporator (COP increases about 0.43 per 10 °C water inlet temperature).

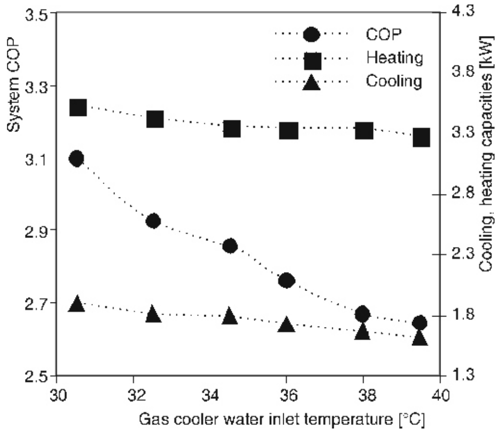

Figure 9. Variation of system performance with gas cooler water inlet temperature

Conclusions

An experimental study on a transcritical CO_2 heat pump system for simultaneous water cooling and heating has been carried out to study the system behaviour and performance for various operating conditions. Measurements for various expansion valve openings show that valve opening has a significant effect near the fully closed position and result shows some optimum valve opening to give maximum performance. Comparison with the previous test data shows reasonably good agreement with a maximum deviation of 10% and the trends are fairly similar. The cooling and heating capacities and COP increase with increase in water mass flow rate to both evaporator and gas cooler; however, effect is more pronounce in case of evaporator mass flow rate (COP increases by 0.6 per 1 kg/min., whereas 0.4 for gas cooler). Water outlet temperatures of evaporator and gas cooler increase with increase in evaporator mass flow rate and decrease with increase in evaporator mass flow rate. Cooling and heating capacities and COP increase with increase in water inlet temperature to evaporator, whereas trends are opposite for gas cooler water inlet temperature; effect of gas cooler water inlet temperature is more pronounce (COP increases by 0.48 per 10 °C, whereas 0.43 for gas cooler). With the increase in water inlet temperatures to both evaporator and gas cooler, water outlet temperatures of both evaporator and gas cooler increase.

Nomenclature

$t_{evw,i}$ – evaporator water inlet temperature, [°C]
$t_{gcw,i}$ – gas cooler water inlet temperature, [°C]
P_{suc} – compressor suction pressure, [bar]

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