EFFECT OF GEOMETRY ON THE PERFORMANCE OF CO_2 GAS COOLER/CONDENSER AND ITS ASSOCIATED REFRIGERATION SYSTEM

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ABSTRACT

As a natural working fluid, CO_2 is increasingly being used in refrigeration systems in which an air cooled CO_2 gas cooler or condenser is employed on the system to reject heat to the ambient. Depending on the application and design, the CO_2 refrigeration system can be a single-stage or multi-stage. Whatever the design, the CO_2 gas cooler or condenser plays an important role in maximising system performance. To understand and facilitate the effect of CO_2 gas cooler or condenser on refrigeration system performance and controls, two CO_2 finned-tube gas coolers/condensers with different geometric designs were built, instrumented, connected to an existing CO_2 booster refrigeration system and tested extensively. Models of the finned-tube CO_2 gas coolers/condensers have been developed and validated with the measurements. The models are used to investigate the effect of geometric parameters on the performance of the CO_2 heat exchangers and associated refrigeration system.

1. INTRODUCTION

As an environmentally friendly working fluid with superb thermophysical properties, CO_2 has been readily applied in refrigeration and heat pump systems. Air cooled finned-tube condensers used in conventional refrigeration systems have also been greatly exploited in CO_2 systems with cascade arrangements or all-CO_2 transcritical arrangements (Schiesaro and Kruse , 2002) of which the CO_2 heat exchangers operate as either condensers or gas coolers, depending on ambient conditions and 'head' pressure controls. Therefore, it is demonstrable that due to the low critical temperature and very high critical pressure of the CO_2 fluid, a CO_2 refrigeration system can periodically operate between high performance subcritical cycles and less efficient transcritical cycles. However, this operating efficiency can be significantly improved through the use of an expansion turbine, a liquid-line/suction-line heat exchanger (llsl-hx), and more efficient system equipment such as a compressor, evaporator or gas cooler/condenser (Kim et al., 2004), as well as optimal controls of refrigerant high-side pressures (Ge and Tassou , 2009). The feasibilities of such strategies can be substantiated through system experiment and modelling.

An experimental investigation was carried out on a two-stage CO_2 transcritical refrigeration system with external intercooling (Cavallini et al. 2005). In the test rig, air-cooled finned tube gas coolers with different structures and circuits were installed in the high pressure side. The test results showed that an optimal head pressure did exist to maximize the system COP which was necessarily controlled in actual operations. Alternatively, it would be beneficial for a direct staging CO_2 transcritical system such as the CO_2 booster refrigeration system to be studied experimentally. To understand the performance of a CO_2 air cooled gas cooler, a series of tests were conducted at different operating conditions using a purposely designed test facility (Hwang et al., 2005). The effects of air and refrigerant side flow parameters on the heat exchanger heat transfer and hydraulic behaviours were examined. In addition, the temperature profiles along the heat exchanger circuit pipes were measured. Further investigation, including a model development, will be implemented to predict these effects on the performance of the associated system. Apart from the overall performance investigations of the CO_2 gas coolers, the in-tube cooling processes of CO_2 supercritical flow were extensively tested and correlated (Srinivas et al., 2002; Yoon et al., 2003; Son and Park , 2006) , which is helpful for the model development of a CO_2 gas cooler.

In terms of the theoretical analysis of CO_2 gas coolers, two modelling methods can be used, ε-NTU or LMTD i.e. lumped method and distributed method (Ge and Cropper , 2008, 2009) . The lumped method is simpler, requires less computation time and is thus suitable for simulating a complex system with an integrated heat exchanger. However, such a modelling method can only provide a reasonable prediction accuracy if appropriate correlations are applied. The rapid change of CO_2 thermophysical properties with temperature during an isobaric gas cooling process means that it is not practical to use ε-NTU or LMTD method to simulate gas coolers if detailed information of refrigerant temperature profile and localised heat transfer rate along circuit pipes are required. The detailed model, however, will take a much longer computation time especially when it involves a whole-system simulation. Although the performances of CO_2 finned-tube gas coolers or condensers have been investigated extensively using both experimental and theoretical methods, research on the effect of their integration with associated systems is still rather limited (Chang and Kim, 2007) . To some extent, this could lead to inaccurate design and mismatching of the heat exchanger size and control when applied to a real system. The combined analysis of the heat exchanger can also contribute towards the selection and design of other matched components and appropriate system controls.

In this paper, the performance of the CO_2 gas cooler or condenser has been investigated experimentally in a purposely-built CO_2 gas cooler test rig, which is connected to a CO_2 transcritical booster refrigeration test facility. Models of the finned-tube CO_2 gas cooler and condenser have been developed using lumped methods which have been validated against test results. The model was then integrated with other component models to establish an overall system model. The effect of heat exchanger sizes and pipe circuitry arrangements on system performance and controls at different operating states has been investigated.

2. EXPERIMENTAL FACILITIES

To examine experimentally the performance of CO_2 gas coolers or condensers with different sizes and operating conditions, a test rig has been purposely built, as shown in Figure 1. The CO_2 heat exchanger is suspended tightly between two upright metal frames. A propeller air fan with variable speed control is installed above the heat exchanger to maintain a fixed airflow. Above it are a number of smaller air fans installed in opposition along the direction of pipe length that will switch on if the air on temperature is controlled to be higher than ambient. As such, part of the hot exhaust air will flow back through the return air tunnels, then return air grills, and mix with lower temperature ambient air flow. If the mixed air flow temperature is still lower than the designed air on temperature, an electric air heater installed just beneath the heat exchanger will be switched on to maintain the air on temperature. Consequently, the gas cooler air on parameters, temperature and flow rate, can be well controlled to specified values. The test rig has been comprehensively instrumented to detailed measurement data and overall performance description of the heat exchanger itself and its integrated CO_2 refrigeration system. These include two thermocouple meshes with 24 points each to measure air-on and air-off temperatures; pressure difference of air flow through the heat exchanger to ascertain the air side pressure drop, air flow velocity to obtain the air flow rate. For the refrigerant side, four pressure transducers are installed inside the inlet and outlet headers and one circuit of the heat exchanger to measure the overall and heat exchanger refrigerant side pressure drops. In addition, as shown in Figure 1, a large number of thermocouples are attached on all the pipe bends along the pipes of one circuit to measure refrigerant temperature variation or profile from inlet to outlet. Instead of measuring the refrigerant mass flow rate directly, it is calculated from the heat exchanger heat balance between the air and refrigerant sides.

As shown in Figure 2, two finned-tube CO_2 gas coolers/condensers with different sizes and pipe arrangements were investigated experimentally in the test rig described above. The larger one named coil A has 3 rows, 4 pipe circuits, 96 pipes in total and overall dimension 1.6m×0.066m×0.82m (L×D×H) while the smaller one named coil B has 2 rows, 2 circuits, 64 pipes in total and overall dimension 1.6m×0.044m×0.82m (L×D×H). All other structural parameters are the same for both heat exchangers, including a 6.72mm inner diameter of copper pipe, 0.16mm thickness of aluminium fin and 453 fins/m fin density.

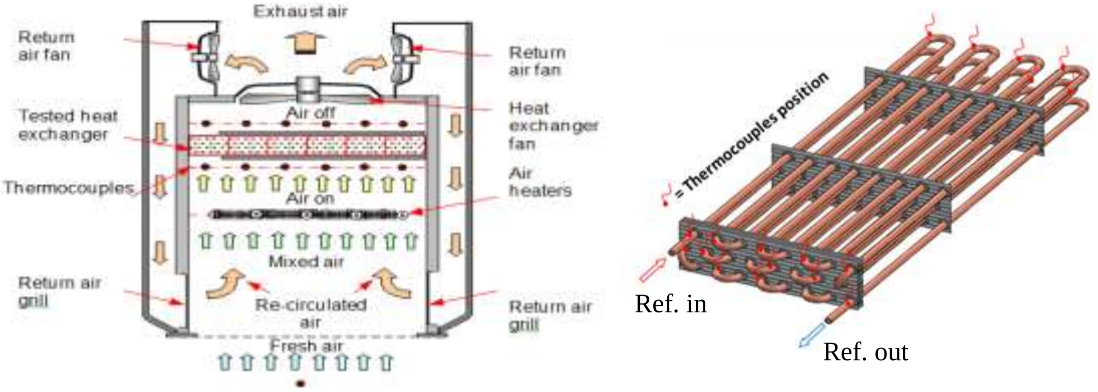
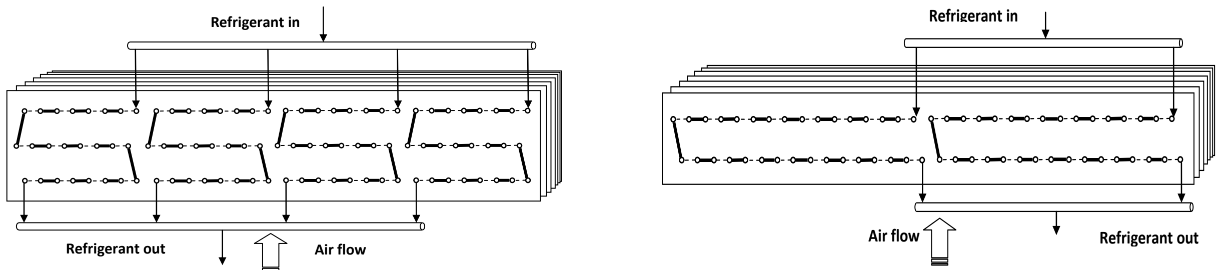


Figure 1. Test rig of CO_2 gas cooler & condenser and pipe arrangement and locations in one heat exchanger circuit



(Coil A-3 rows 4 circuits) (Coil B-2 rows 2 circuits)

Figure 2. two tested finned-tube CO_2 gas coolers/condensers

The two tested gas coolers/condensers are connected separately with their integrated refrigeration system, as depicted in Figure 3; this allows it to pass through the refrigerant flow at a constant flow rate within specific superheated parameters. The refrigeration layout is actually part of a CO_2 booster system in which the tested gas cooler/condenser is connected to two parallel high-temperature compressors and a liquid receiver. In the refrigeration system, there are three pressure levels: high, intermedium and medium, which are controlled respectively by the back pressure valve (ICMT) connected after the gas cooler, bypass valve (ICM) and thermostatic expansion valve (AKV-MT). The approach temperatures at the gas cooler and condenser outlets are controlled by the gas cooler fan speed and ICMT while the system cooling capacity is modulated with variable speeds of compressors. In Figure 3, at one test condition, sample measured parameters of temperatures and pressures at each component inlet and outlet are also indicated.

All the sensors in the system were calibrated before the experiments to ensure acceptable accuracy with thermocouples uncertainty less than ±0.5 ^{0}C, pressure transducer ± 0.3% and air velocity meter ±3.0 %.

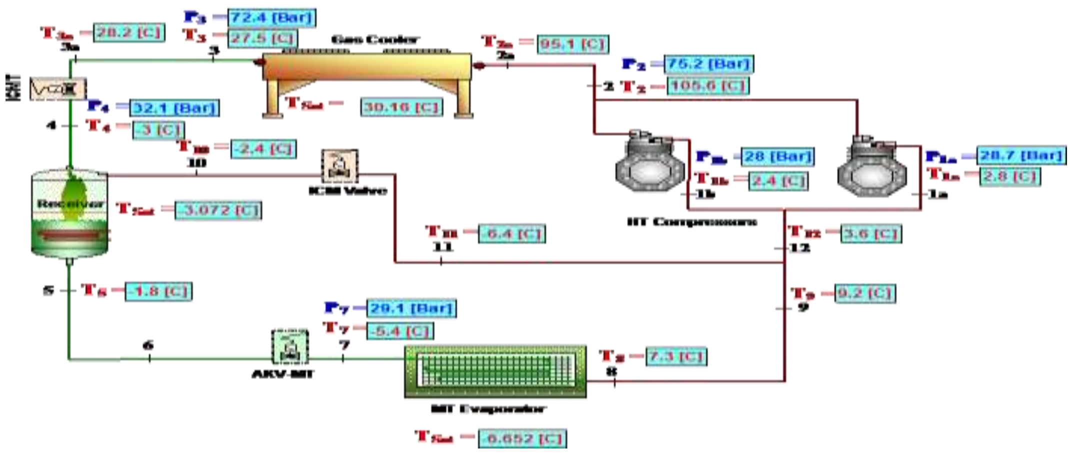


Figure 3. Tested CO_2 gas cooler/condenser and its integrated refrigeration system

3. MATHEMATICAL MODELS AND VALIDATIONS

The CO_2 finned-tube gas cooler/condenser can be modelled using two well-known methods: distributed and lumped. The former is a detailed model and has been developed by the authors (Ge and Cropper, 2009), in which the heat exchanger to be modelled is divided into a number of small segments with specified 3-D coordinates i, j and k along directions of pipe length, longitudinal and transverse. For each piece, conservation equations of mass, momentum and energy need to be derived and applied. The detailed model applies localised correlations of heat and mass transfer coefficients and hydraulic processes such that the local parameter distribution profiles such as temperature, pressure and heat transfer rate, of both hot and cold fluid flowing along the heat exchanger can be accurately predicted. Therefore, the detailed model is believed to be more precise than other modelling methods . However, if more segments are divided for a modelled heat exchanger, it would require a longer computation time. Therefore it is not entirely appropriate for simulating a refrigeration system where the heat exchanger is integrated with a detailed model. Comparatively, the lumped method is a simple model in which the heat exchanger is divided into very limited number of segments and each is described with conservation equations of mass, momentum and energy. The computation time is thus greatly saved and it is more practical to model a system, although prior model validation with test results is strictly required. The lumped method is therefore used in this paper to model the tested CO_2 gas coolers /condensers and the whole system, as shown in Figure 3, to examine their compatibilities with the system and controls. The comprehensive description of the simple model development and validation can be found from our previous work (Ge at al., 2014)

4. MODEL APPLICATIONS

The developed simple CO_2 gas cooler model is then integrated into the existing CO_2 booster system to compare and analyse the effects of heat exchanger sizes (number of pipe rows and circuits) and controls (supercritical pressures) on the system performance under various design specifications. These include a specified evaporating temperature, superheating, compressor swept volume rate and head pressure controls, etc., as listed in Table 1. The structure and pipe arrangement of the CO_2 gas cooler- Coil B shown in Figure 2 is used as the base for the analysis. For a gas cooler with different number of rows such as 3 rows, all other structural parameters are the same apart from the row number. In the modelling simulation, the gas coolers with 2 to 6 pipe circuits are also applied and the pipe numbers are evenly distributed for each circuit.

Table 1. Specification of system component parameters and controls

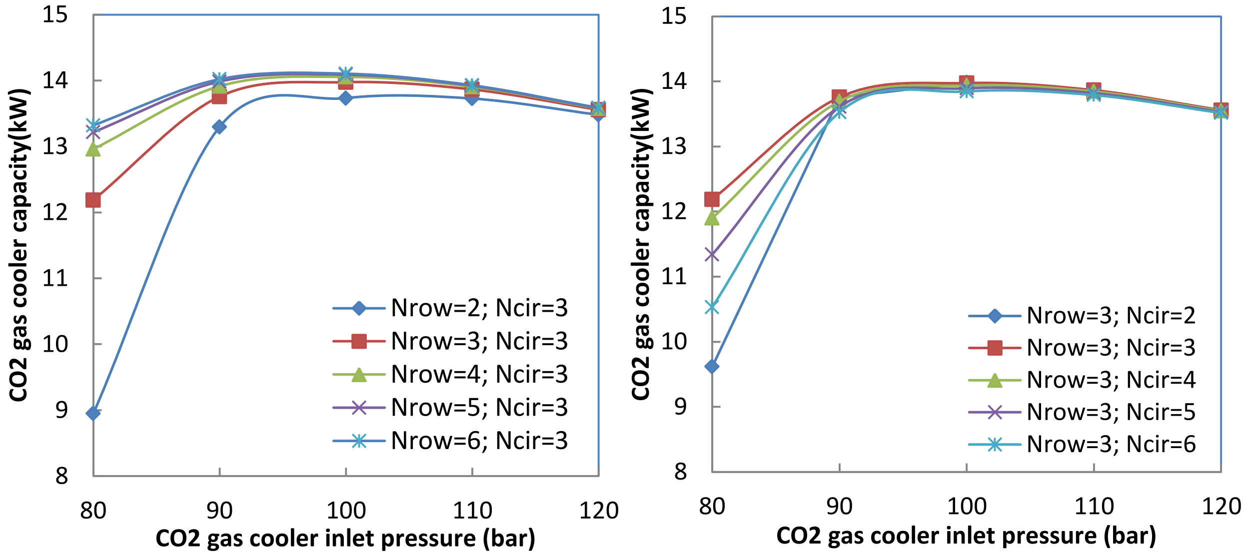
Component	Parameter	Control
Evaporator	Evaporating temperature	-10^0C
	Superheating	10K
Compressor	Swept volume rate	3.64 m^3/h
Gas cooler	Supercritical pressure	Constant 80~120 bar
	Pipe row number in the air flow direction (N_{row})	2~6
	Air flow rate	500 l/s
	Air flow temperature	30^0C
	Pipe circuit number(N_{cir})	2~6

For the booster CO_2 refrigeration system shown in Figure 3, a number of Bock CO_2 semi-hermetic reciprocating compressors of the same size and type (RKX26/31-4 CO_2 T) are to be installed in parallel for the simulation. The total swept volume rate of these compressors is assumed to be 3.64 m^3/h. Based on the manufacture catalogue, the compressor isentropic and volumetric efficiencies in transcritical cycles can be correlated and used in the compressor model:

$$\eta_{is} = 0.74443 - 0.050539 R_p \quad (1)$$

$$\eta_v = 1.2733 - 0.1406 R_p \quad (2)$$

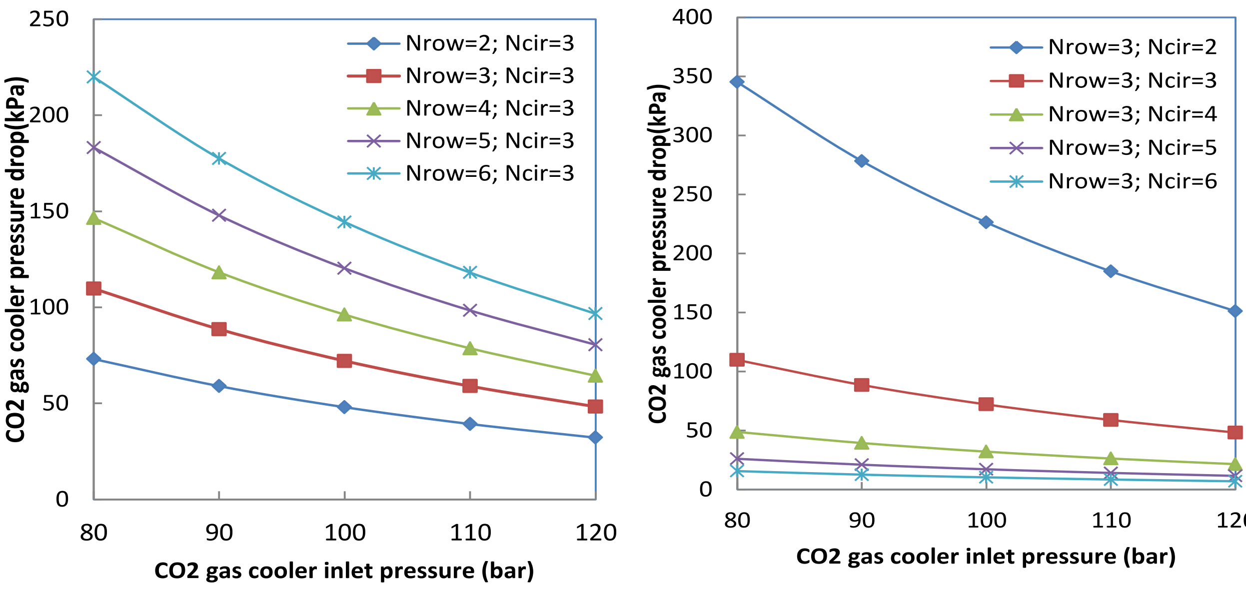
where η_{is}=isentropic efficiency, η_v=volumetric efficiency, R_p=pressure ratio.



(a) (b)

Figure 4. Variation of CO_2 gas cooler capacity with supercritical gas cooler pressure and numbers of pipe rows (a) and circuits (b)

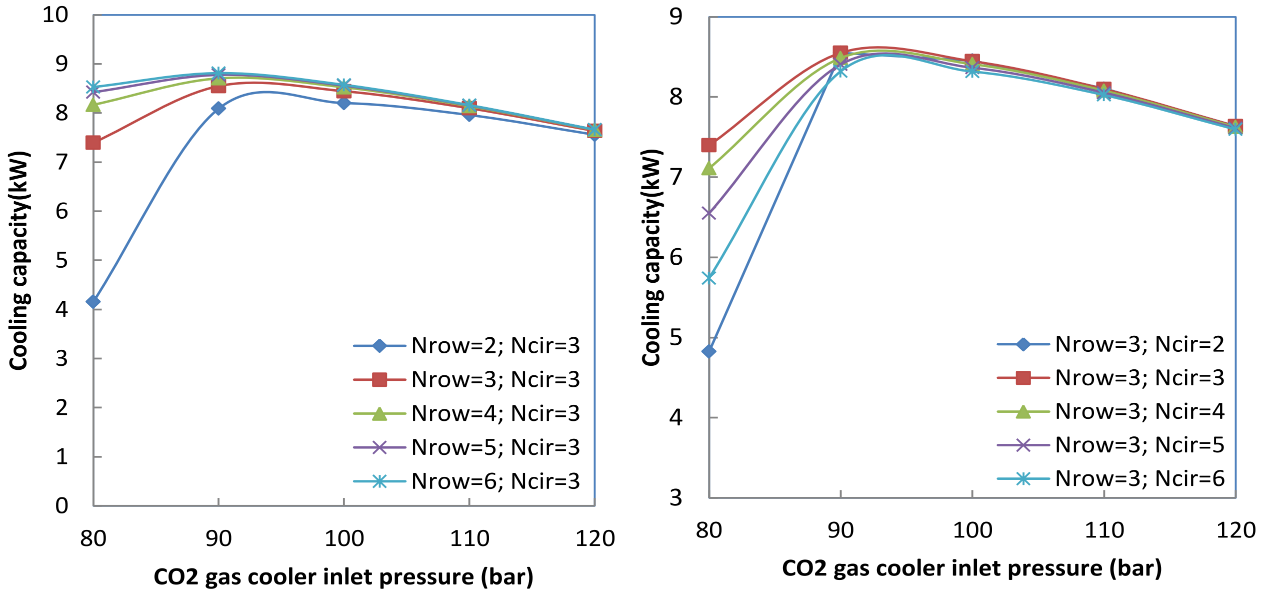
The variations of CO_2 gas cooler heat capacity with supercritical gas cooler pressure and numbers of pipe rows (a) and circuits (b) are predicted by the model and depicted in Figure 4. It is indicated that at a fixed circuit number and coil inlet pressure, the heat capacity increases when the coil row number increase. This is understandable that the more number of pipe rows, the larger of the heat exchanger size and heat transfer area and thus more heat capacity. However, the increase rate is not constant but larger for the coils with less number of rows. When the rows increase above five, the capacity increase rate can be neglected. In addition, the increase rates are insignificant if the coil inlet pressure is above 90 bar. On the other hand, for a gas cooler with unchanged pipe rows such as three rows, the capacity decreases with higher number of circuits due to the reduction of refrigerant mass flow rate, as shown in Figure 4(b). Nevertheless, if the circuit number is as less as two in this case, the higher circuit refrigerant mass flow rate can lead to higher pressure drop which can conversely decrease the heat capacity. Similarly, the effect of circuit number on the capacity is insignificant when the CO_2 inlet pressure is higher than 90 bar.



(a) (b)

Figure 5. Variation of CO_2 gas cooler pressure drop with supercritical gas cooler pressure and numbers of pipe rows(a) and circuits(b)

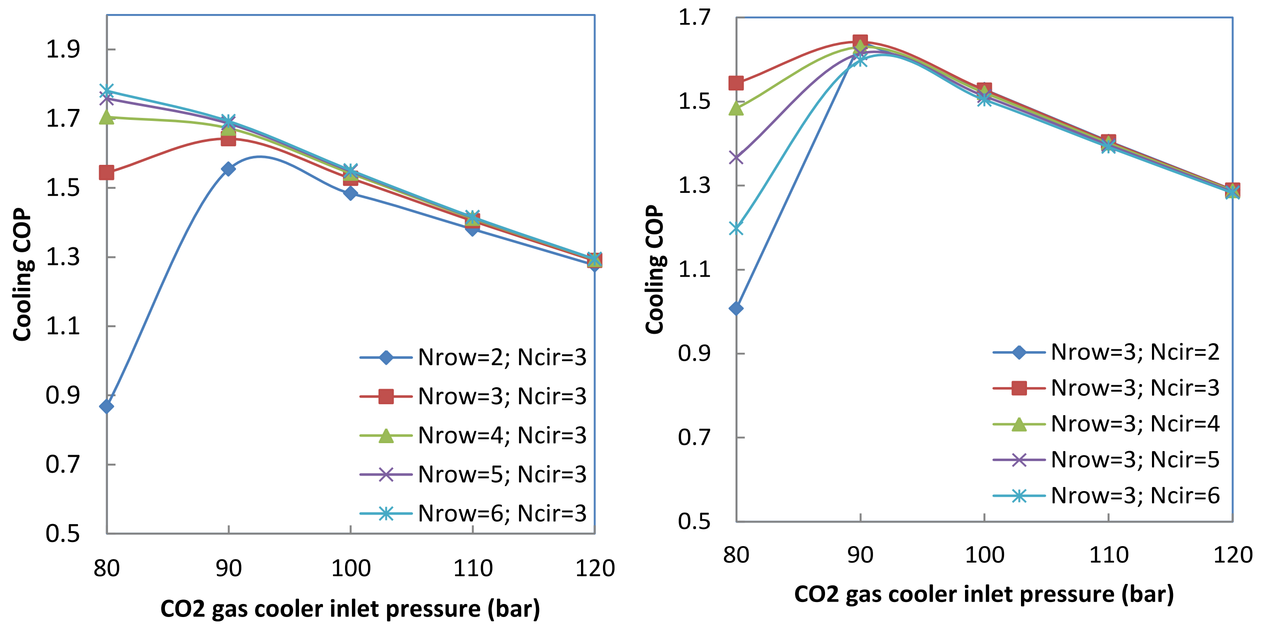
The pressure drops for the gas cooler with different rows and circuits are therefore calculated by the model and shown in Figure 5. At a fix CO_2 inlet pressure and circuit number, the coil with more rows representing larger heat exchanger will definitely cause extra refrigerant side pressure drop. For a selected gas cooler, the increase of circuit number will reduce circuit refrigerant flow rate and thus pressure drop. It is also noted that for the same gas cooler with a specified circuit number, the pressure drop in each circuit can be less if the system operates at a higher supercritical pressure. This is due to the abrupt changes of the CO_2 thermodynamic properties with the supercritical pressure.



(a) (b)

Figure 6. Variation of cooling capacity with supercritical gas cooler pressure and numbers of pipe rows (a) and circuits(b)

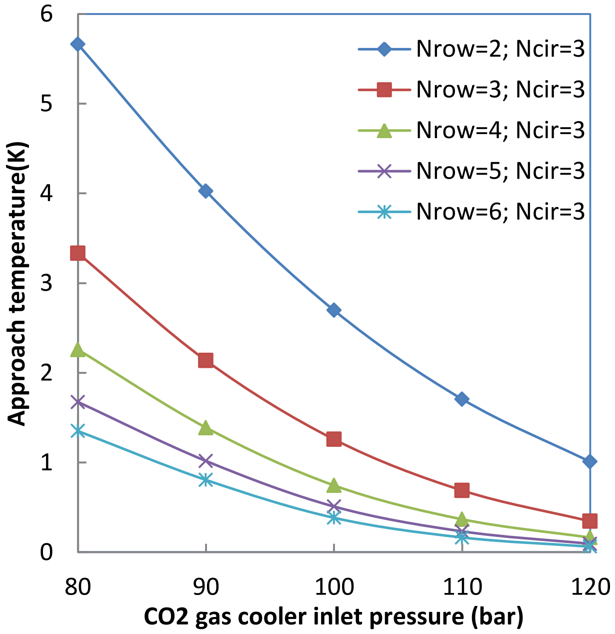
Corersponding to the heat capacity of gas cooler, the higher heat capacity indicates less refrigerant enthalpy at the gas cooler outlet or the evaporator inlet. The cooling effect or cooling capacity will thus be higher. Therefore, as shown in Figure 6, similar results to the gas cooler capacity can be predicted for the cooling capacity at different high-side supercritical pressure and number of pipe rows and circuits of the gas cooler.



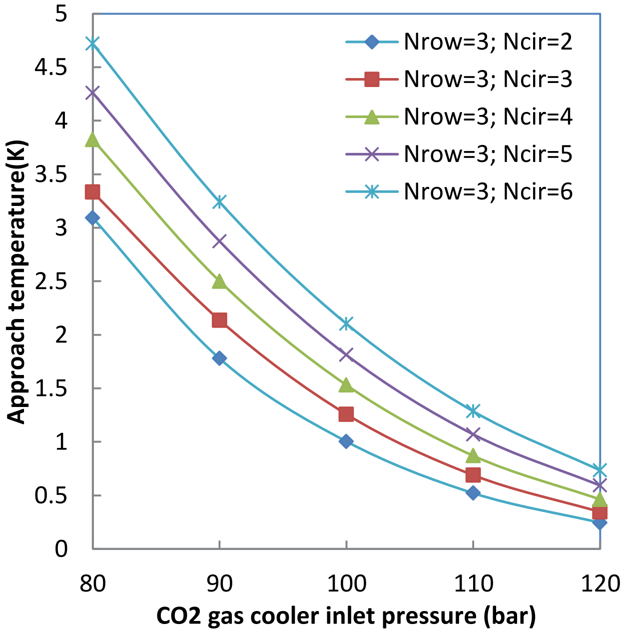
(a) (b)

Figure 7. Variation of cooling COP with supercritical gas cooler pressure and numbers of pipe rows (a) and circuits(b)

When refrigerant state at the evaporator outlet or the compressor inlet is fixed, the compressor power consumption is only determined by the pressure ratio or its discharge pressure at the gas cooler inlet. Since the cooling COP is calculated as the ratio of cooling capacity and compressor power consumption, it is predicable that similar variations will be presented for the cooling COP with the discharge pressure and the number of pipe rows and circuits of the gas cooler, as shown in Figure 7.



(a)



(b)

Figure 8. Variation of approach temperature with supercritical gas cooler pressure and numbers of pipe rows (a) and circuits(b)

The approach temperature is defined as the temperature difference between CO_2 gas cooler outlet and gas cooler incoming air. It is determined mainly by the heat exchanger size and heat transfer behaviours at both hot and cold fluid sides, which can be demonstrated by the simulation results shown in Figure 8. For a fixed pipe circuit number and constant CO_2 inlet pressure, the higher number of rows indicates increased coil size for each circuit and thus leading to smaller approach temperature. However, the reduction rate is gradually approaching to zero with larger coil size. Simultaneously, at a fixed gas cooler size and CO_2 pressure, the less number of circuit is better for the approach temperature reduction due to the increase of circuit refrigerant mass flow rate which is thus enhance the refrigerant side heat transfer. In addition, the higher CO_2 inlet pressure can also present better refrigerant side heat transfer behaviours and subsequently obtain smaller approach temperature.

The simulation results demonstrate that the CO_2 gas cooler designs and controls are quite important to the performance of both the heat exchanger and its associated CO_2 refrigeration systems and component costs. Considering of these issues, the 3-row and 3-circuit gas cooler may be the optimised design for this case study. In addition, at this specified operating state, there is an optimal supercritical pressure (90 bar) to be controlled to achieve the best performance for CO_2 refrigeration system and its main components including gas cooler, evaporator and compressor.

5. CONCLUSIONS

In varying ambient air temperatures, the high pressure CO_2 heat exchanger in a CO_2 refrigeration system will operate as either a gas cooler or condenser. The heat exchanger needs to be well designed and controlled so as to maximize the performance of its integrated system. Accordingly, two different sized and structural CO_2 fined-tube heat exchangers are manufactured and installed in a test rig of a CO_2 booster system. Extensive experiments have been carried out in the test rig at different operating states and controls. In the meantime, the CO_2 heat exchanger models have been developed using two methods : distributed and lumped. The former needs to be utilised if detailed profiles of temperature and heat transfer rate along a pipe circuit are required. However, the latter method is suitable for a system simulation when the heat exchanger model is integrated into an overall system. The simple model have been validated with corresponding experiment measurements and the simple model is used to predict the effects of the heat exchanger sizes and controls on the system performance.

The simulation results show that the CO_2 gas cooler designs in terms of size (pipe rows) and circuit arrangements and inlet pressure controls can affect greatly to the performance of the heat exchanger and its associated system. The larger gas cooler size or row number can improve the system and coil efficiencies but there is a limitation considering the consequence of cost, refrigerant side pressure drop and reduced increase rate for the system performance. Similarly, the lower circuit number is better to the system and heat exchanger performance but it cannot be too low considering the caused larger pressure drop. In addition, the high-side supercritical pressure controls have a significant impact to the system operation and an optimal pressure should be controlled for a better system performance.

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