MODELLING AND ANALYSIS OF CO_2 GAS COOLERS FOR COMMERCIAL REFRIGERATION APPLICATIONS

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ABSTRACT

CO_2 is an environmentally benign refrigerant which is increasingly being used in commercial refrigeration applications. CO_2 refrigeration systems can operate subcritically during periods of low ambient temperature or transcritically when the ambient temperature is above 25 °C or so, depending on the design of the heat rejection heat exchanger. Optimisation of the heat transfer performance of the heat rejection heat exchanger can extent the temperature range in which the CO2 refrigeration system can operate in the subcritical region and this will improve the seasonal efficiency of the system. This paper presents a model which has been developed to simulate the performance of CO_2 heat rejection heat exchangers. The model has been validated against both sub-critical and transcritical data with the view to establishing its suitability for use as a design and selection tool for CO2 heat rejection heat exchangers over a wide range of conditions.

Keywords: gas cooler, commercial refrigeration, optimisation of gas cooler

1. INTRODUCTION

The use of CO_2 (R-744) as a refrigerant has been increasing in recent years due to its good thermodynamic properties zero Ozone Depletion Potential and negligible Global Warming Potential. CO_2, however, suffers from a relatively low critical temperature of 31°C and at a pressure of 73 bar, so CO_2 systems will operate in the transcritical region (above the critical point), when the ambient temperature is below 31 °C. The transition point between sub-critical and transcritical operation will depend on the design of the heat rejection equipment and operation and control strategy employed, but in the majority of cases transition takes place at ambient temperatures in the range 23 °C and 26 °C. A single heat exchanger is normally used for heat rejection both during transcritical and subcritical operation of the refrigeration system. In transcritical operation the heat exchanger rejects heat from the superheated refrigerant gas to the ambient air without condensation of the CO_2 gas taking place (single phase heat transfer). In this case the heat exchanger is known as a *gas cooler*. In subcritical operation the heat exchanger behaves in a very similar way to a standard condenser, condensing CO_2 refrigerant gas by rejecting heat to the ambient air, and is referred to as a *condenser*.

In recent years significant numbers of investigations have been performed on the design and control optimisation of CO_2 refrigeration systems (Neksa, 2004; Colombo *et al* . 2011) amongst many others. Gupta *et al*.(2010) indicated that three major factors which affect the performance of transcritical CO_2 refrigeration systems are; the design of the gas cooler, the gas cooler pressure and the gas cooler outlet temperature. The gas cooler outlet temperature will depend on the ambient conditions. The approach temperature is also an important parameter in the design of gas coolers. To reduce the thermodynamic losses the refrigerant exit temperature from the gas cooler should approach the coolant inlet temperature. Thermodynamic analyses of CO_2 refrigeration systems indicated that whilst the system operates in the transcritical mode, there is an optimum discharge pressure that maximises the system COP. Ge and Tassou (2009) investigated the variation of this optimum pressure with ambient air temperature using a fixed approach temperature of 3 K, a constant evaporating temperature of -10° C and 10 K superheat at the evaporator outlet.

Waltrich *et al*. (2010) developed a model to investigate the thermo-hydraulic behaviour of compact fan-cooled tube-fin heat exchangers used as gas coolers in CO_2 refrigeration systems. The model was based on the mass, momentum and energy conservation equations applied to both the refrigerant and air streams. The model predictions for the temperature profiles, heat transfer rate and the air-side pressure drops were compared with experimental data obtained using a purpose-built test facility. It was observed that the model is able to predict the overall performance of heat exchangers in terms of heat transfer rate and pressure drop with errors within 10% and 15% bands, respectively. Ge and Cropper (2009) presented a distributed model for the steady state simulation of finned-tube air-cooled gas coolers. The model was validated against data published in the literature and was used to investigate the effect of different pipe circuit arrangements on the performance of the gas coolers.

2. MODEL DESCRIPTION

Depending on the ambient air temperature and head pressure controls, the same high-side air cooled heat exchanger in a CO_2 refrigeration system can function as either a gas cooler or condenser. It is therefore important to be able to simulate the performance of the heat exchanger for both modes of operation.

In this paper, the standard effectiveness-NTU method is employed to model the CO_2 heat exchanger in the gas cooler and condenser modes of operation. The heat exchanger is normally arranged in a number of parallel pipe circuits to ensure appropriate refrigerant mass flux and equalised pressure drop for each circuit. To illustrate the heat exchanger model in the condenser mode, a sample single circuit pipe arrangement is depicted in Figure 1. The circuit has four tubes parallel to air flow and six tubes in each row perpendicular to the air flow.

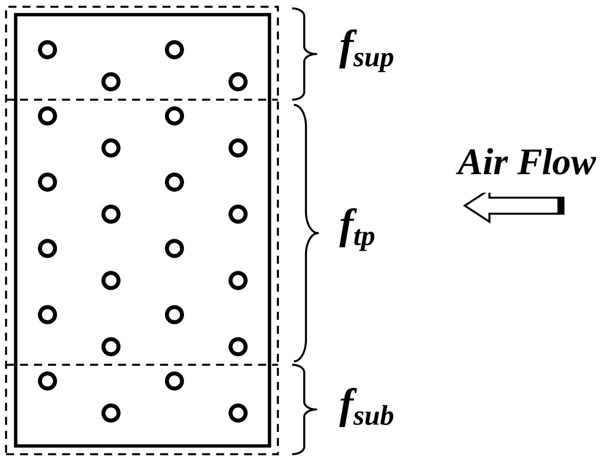

Figure 1 A sample single circuit fined-tube heat exchanger

For such a heat exchanger in the condenser mode, it is assumed that the refrigerant flows downward such that it passes consecutively through the desuperheating, two-phase and subcooling regions with the heat transfer area fractions of f_{sup} , f_{tp} and f_{sub} respectively. It should be noted that the outlet of the superheating section or inlet to the two-phase section is at saturated vapour state, and the outlet of the two-phase section also the inlet to the subcooling section is saturated liquid. A small degree of subcooling make take place in the sub-cooling section before the refrigerant liquid exits the condenser. The accurate determination of the fractions of the heat exchanger used for desuperheating, condensing and subcooling are critical for the prediction accuracy of the model. These fractions are determined from the application of the conservation equations of mass, momentum and energy for each corresponding section in the heat exchanger.

The condenser model is based on the following assumptions:

- flow conditions are in steady state;
- heat conductions along the pipe axis are neglected;

- air flow is distributed homogeneously through each section;
- contact resistance between pipe and fin is ignored;
- refrigerant flow at each cross section is in thermal equilibrium.

For the refrigerant side of each section, the mass conservation is automatically satisfied at the steady state. The momentum equation can be represented by the calculation of pressure drop along the pipe while the energy equations for each section can be calculated as below.

The heat transfer from the refrigerant side is :

$$Q = \dot{m}_r(h_{in} - h_{out}) \tag{1}$$

Where, $\dot{m}_r$ is refrigerant mass flow rate along each section in the same circuit and h_{in} and h_{out} are refrigerant enthalpies at a section inlet and outlet respectively.

As to the air side, similarly, the mass balance is automatically satisfied at the steady state and the pressure drop calculation is applied for the momentum equation in this side. There is an heat balance between refrigerant and air flows for each section:

$$Q = \dot{m}_a C_{pa}(T_{ain} - T_{aout}) = \varepsilon(G_c)_{min}(T_{hin} - T_{cin}) \tag{2}$$

Where the dry coil effectiveness ε can be calculated as:

$$\varepsilon = \begin{cases} \dfrac{\mathbf{1-exp}(-(\mathbf{1-exp}(-Ntu))(G_c)_{min}/(G_c)_{max})}{(G_c)_{min}/(G_c)_{max}} & \text{for refrigerant single} - \text{phase section} \\ 1 - \exp(-Ntu) & \text{for refrigerant two} - \text{phase section} \end{cases} \tag{3}$$

and,

$$Ntu = \frac{(\boldsymbol{UA})_{\boldsymbol{tot}}}{(G_c)_a} \tag{4}$$

where,

$$(\boldsymbol{UA})_{\boldsymbol{tot}} = \frac{\mathbf{1}}{\frac{\mathbf{1}}{(\boldsymbol{UA})_{\boldsymbol{o}}} + \frac{\mathbf{1}}{(\boldsymbol{UA})_{\boldsymbol{i}}}} = \frac{\mathbf{1}}{\frac{\mathbf{1}}{\boldsymbol{\eta_f \alpha_o A_o}} + \frac{\mathbf{1}}{\boldsymbol{U_i A_i}}} \tag{5}$$

For the gas cooler model, the refrigerant state is always be supercritical gas and the heat transfer fractions shown in Figure 1 will be set to 1, 0, 0 for f_{sup} , f_{tp} and f_{sub} respectively. The above equations applicable for single vapour gas phase can be applied directly for the calculations of the gas cooler model .

The overall heat transfer conductance for 'dry' heat exchangers is computed in terms of the fin efficiency as outlined in Threlkeld (1970) and heat transfer coefficients from both air and refrigerant sides. For air flow over finned coil surface, correlations developed by Wang and Chi (2000) are utilized for determining air-side heat transfer coefficient and pressure drop. The refrigerant side single phase heat transfer coefficient and pressure drop are determined using standard turbulent flow relations. For the refrigerant two-phase flow, correlations from Dobson and Chato(1998) , and Müller-Steinhagen and Heck(1986) are used to calculate the heat transfer coefficient and pressure drop respectively. In addition, the refrigerant properties are calculated from the NIST properties software REFPROP 8 (2007).

3. TEST PROCEDURE AND RESULTS

3.1 Experimental Facilities and Test Conditions

For the validation of the model, test facilities were employed which include a CO_2 refrigeration system and a specially designed gas cooler test rig. The gas cooler rig is shown schematically in Fig. 2(a). It allows for the variation of air flow rate and air temperature on the gas cooler.

The finned tube gas cooler/condenser coil used for the tests is detailed in Table 1. A schematic of a single section indicating position of thermocouples on the tubes is shown in Fig 2(b).

Table 1. Specification of Gas Cooler

Number of circuits	4	Tube thickness	0.640 mm
Tube configuration	Staggered (Equilateral)	Tube vertical spacing	22 mm
Number of tubes per row	32	Tube horizontal spacing	25.4 mm
Number of rows deep	3 rows	Fins per inch	14 in^{-1}
Tube length	1600 mm	Fin thickness	0.16 mm
Tube OD	8 mm	Coil face air velocity	Variable

Measurement systems were used during the tests including, pressure, temperature and mass flow rate on the R-744-side and velocity, pressure drop and temperature on the air side.

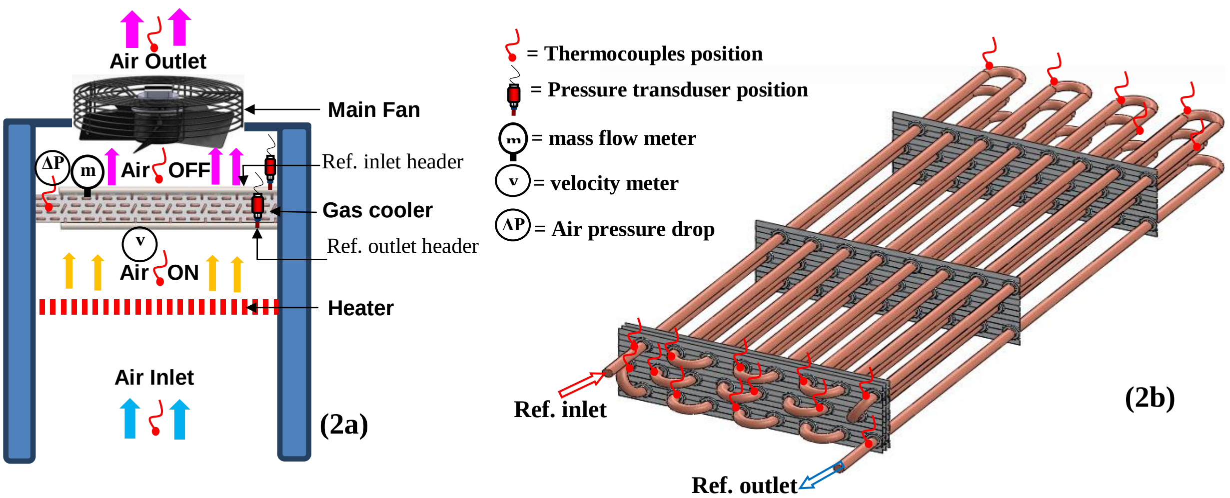

Fig.2a Scematic of Gas cooler testing unit (Fig.2b)Position of themocouples on coil tubes

On the tube side temperatures were measured at every bend (30 points in all). The air temperature entering the gas cooler/condenser was measured at 24 points along the face of the coil and at 12 points after the coil. The refrigerant pressure measured at the inlet and outlet of the coil and the air pressure drop was measured with a differential pressure transducer. The refrigerant mass flow rate was measured with a coriolis mass flow meter located on the refrigerant liquid line after the liquid reveiver.

The thermocouples used had an uncertainty narrower than ± 0.5°C, the pressure transducers (uncertainty ± 0.3%), mass flow meter (uncertainty ± 0.035%) and air velocity meter with uncertainty ± 3 %. To enable the information to be read and recorded for system analyses and evaluation, the instrumentation was connected to a data logging system (Labtech software and Datascan modules).

3.2 Test Results

Testing was done with air flow rates across the coil of 2000 ls^{-1}, 2400 ls^{-1}, 2800 ls^{-1}, or air velocity 1.6 ms^{-1}, 2 ms^{-1}, 2.5 ms^{-1}, respectively. Air-ON temperatures were varied by regulating the air recirculation rate and modulating the air-on heaters of the gas cooler test rig. The air temperature onto the coil was varied between 20°C and 35°C. The results for the condenser mode (CD1 to CD9) are detailed in Table 2 and for the gas cooler mode (GC1 to GC9) in Table 3. The performance of each parallel circuit of the coil was found to be very similar so temperature data of only one circuit are presented in this paper.

Figs.3(a) and 3(b) show the variation of the refrigerant temperature at 30 points along the length of the heat exchanger circuit, inlet to outlet. The temperatures were measured at each bend of the refrigerant piping. It can be seen that for both the condenser and gas cooler mode, most of the heat transfer between the refrigerant and air takes place in the first tube of the circuit. For the condenser mode, 80% of the temperature drop on the refrigerant side takes place in the first tube (1.6 m length) whereas for the gas cooler the first tube is responsible for 77% of the temperature drop. This indicates that the gas cooler was oversized for the refrigeration capacity of the refrigeration system. However, the results could be used for the validation of the model.

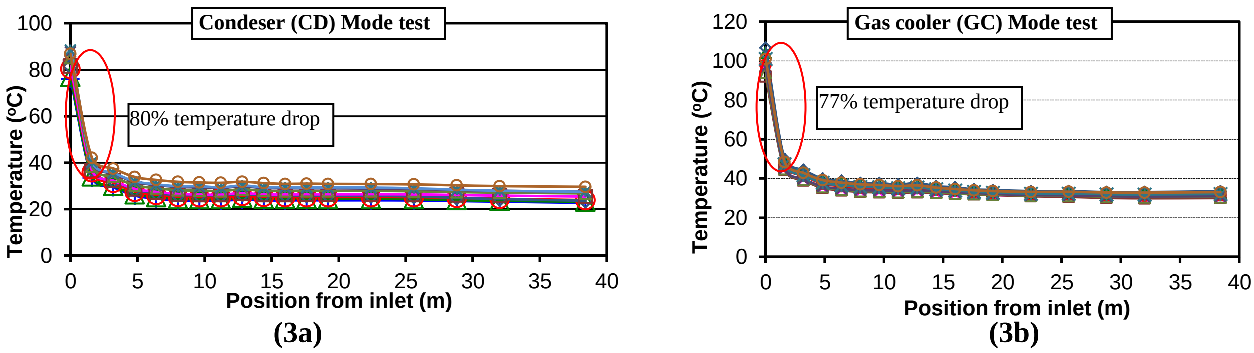

Fig.3 Coil tube temperature at circuit 2 (a) Condenser mode test (b) Gas cooler mode test

4. MODEL VALIDATION AND DISCUSSION

Tables 2 and 3 provide a comparison between the test and model results for the condenser and gas cooler modes of operation for all conditions tested. For heat exchanger air on temperature, air flow rate and heat exchanger refrigerant inlet pressure, the tables present test and simulation results for air temperature rise, air pressure drop and refrigerant pressure drop across the heat exchanger, refrigerant temperature at exit of the heat exchanger and degree of subcooling and heat rejection from the heat exchanger.

It can be seen that the model is able to predict the performance of the heat exchanger in both the condenser and gas cooler model reasonably well. The maximum error in the prediction of heat rejection was ± 6% for the condenser mode and ± 2.7% for the gas cooler mode. This is also reflected in the good accuracy of prediction of the air temperature rise across the heat exchanger. It can be seen that the air side pressure drop increases with increasing gas cooler/condenser air flow rate, as expected. The error in the prediction of air pressure drop shown in Fig. 4, is within ± 20%. The differences are mainly due to fluctuations in measurements.

Table 2. Model validation for condenser mode

Refer to Test	AMT air-ON (°C)	P_{CO2} (bar-a)	Air flow rate (L/s)	DT_{air}(°C)		ΔP_{air} (Pa)		ΔP_{CO2} (kPa)		T_{CO2-o} (°C)		$\Delta T_{subcool}$ (°K)		Heat rejection (kW)	
				T	M	T	M	T	M	T	M	T	M	T	M
CD1	22	65	2000	3.8	3.7	25.2	24.0	44	43.7	24.1	23.9	1.5	1.7	9.1	9.0
CD2	25	68	2000	3.8	3.3	20.2	23.4	47	48.7	26.3	27.5	1.3	0.0	9.0	8.1
CD3	29	73	2000	3.6	3.3	18.9	22.7	48	47.9	29.9	30.5	1.7	0.2	8.7	7.8
CD4	22	64	2400	3.2	3.1	31.9	32.0	47	47.0	23.8	24.3	1.3	0.7	9.3	9.2
CD5	24	66	2400	3.1	3.0	31.3	31.6	52	45.6	25.3	26	1.1	0.4	9.1	8.9
CD6	27	73	2400	3.0	2.8	36.4	30.6	40	43.6	28.4	29.9	2.3	0.7	8.7	8.2
CD7	19	60	2800	2.6	2.4	36.9	42.0	43	43.5	20.4	21.8	1.3	0.0	9.0	8.4
CD8	24	67	2800	2.7	2.7	38.0	40.2	48	44.4	25.7	25.5	1.2	1.4	9.2	9.2
CD9	27	71	2800	2.6	2.5	38.0	39.1	49	43.8	27.6	28.5	1.8	0.9	8.8	8.5
Error				**± 0.14 K**		**± 20.6%**		**± 6.4%**		**± 1.15 K**		**± 0.89 K**		**± 6%**	

AMT = arithmetic mean temperature; CD = test for condenser mode; T=Test; M=Model

Maximum error in the prediction of refrigerant pressure drop is ±6.4% for the condenser and ±3.9% for the condenser. The measurements and predictions include the pressure drop in the headers of the heat exchanger. Future investigations will consider the measurement and prediction of pressure drop in the tubes and headers separately to ensure that errors in the prediction of the two separate components are independent of each other.

Table 3. Model validation for gas cooler mode

Refer to Test	AMT air-ON (°C)	P_{CO2} (bar-a)	Air flow rate (L/s)	DT_{air}(°C)		ΔP_{air} (Pa)		ΔP_{CO2} (kPa)		T_{CO2-o} (°C)		$\Delta T_{subcool}$ (°C)		Heat rejection (kW)	
				T	M	T	M	T	M	T	M	T	M	T	M
GC1	30	76	2000	3.9	3.9	20.7	22.6	51	53.4	29.8	30.3	-	-	9.3	9.2
GC2	31	79	2000	3.9	4.0	23.5	22.4	52	53.9	31.1	31.4	-	-	9.4	9.3
GC3	33	85	2000	4.1	4.1	26.6	21.9	51	52.5	33.2	33.7	-	-	9.7	9.4
GC4	29	76	2400	3.1	3.2	32.6	30.1	48	49.8	29.5	29.8	-	-	8.9	8.9
GC5	31	78	2400	3.1	3.1	30.7	29.7	49	48.0	31.6	31.7	-	-	8.7	8.8
GC6	33	84	2400	3.1	3.1	34.2	29.2	49	47.4	32.8	33.4	-	-	8.9	8.6
GC7	29	76	2800	2.3	2.7	42.0	38.5	45	45.8	29.6	29.6	-	-	8.8	8.8
GC8	31	82	2800	2.7	2.8	38.2	37.7	48	46.4	31.8	32.0	-	-	9.1	8.9
GC9	32	85	2800	2.8	3.7	39.5	37.4	45	43.2	32.8	32.9	-	-	9.0	8.8
Error				**± 0.64 K**		**± 10%**		**± 3.9%**		**± 0.73 K**		-		**± 2.7%**	

AMT = arithmetic mean temperature; GC = test for gas cooler mode; T=Test; M=Model

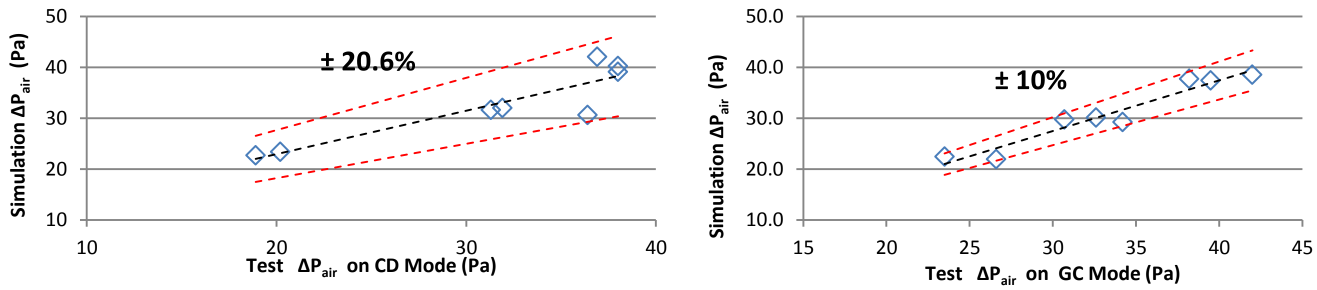

Fig. 4 Comparison of simulation and test result for CD air pressure drop and GC air pressure drop

5. CONCLUSION

CO_2 has several advantages as a natural refrigerant over conventional hydrofluorocarbon refrigerants and has been attracting increasing interest in applications such as commercial refrigeration, heat pumps and air conditioning. The gas cooler/condenser is a key component of CO_2 refrigeration systems and its appropriate design or selection for a particular application can have a significant bearing on the overall system efficiency and reliability. Mathematical models can be employed for quick sizing and/or selection of the most appropriate gas cooler for a specific application. This paper presented a lump parameter based model of a CO_2 condenser/gas cooler. The model has been shown to provide reasonably accurate results in both the condenser and gas cooler mode of operation when compared with data from tests conducted on a gas cooler in the laboratory. Having been validated over a wide range of conditions, the model can be used to investigate the influence of key design parameters on the performance of the gas cooler.

NOMENCLATURE

A	Area (m^2)
Air OFF	Air-side outlet heat exchanger (^oC)
Air ON	Air-side inlet heat exchanger (^oC)
AMT	Arithmatic mean temperature (^oC)
Bar-a	pressure-absolute (Bar)
C_p	Constant pressure specific heat (J/kg.K)
DP	Pressure different
DT	Temperature different
f	fraction of overall heat transfer area (0-1)
GC	Gas cooler state
G_c	Heat capacity (W/K)
h	Enthalpy(J/kg)
HX	Heat exchanger
Ntu	Overall number of transfer units (-)
ODP	Ozone depletion potential
Q	Heat transfer rate (W)
RH	Relative humidity (%)
T	Temperature of brine (K)
UA	Overall heat conductance (W/K)
U	Overall heat transfer coefficient ($W/m^2.K$)

Greek symbols

α	heat transfer coefficient ($W/m^2.K$)
ε	heat transfer effectiveness (-)
η	efficiency (-)

Subscripts

a	air
ain	air inlet
cin	cold fluid in
aout	air outlet
f	fin
hin	hot fluid in
hout	hot fluid out
i	inner
max	maximum
min	minimum
o	outer
out	outlet
r	refrigerant
sub	subcooling
sup	superheating
tp	two-phase

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