Experimental Investigation of CO2 Gas Cooler/Condenser in a Refrigeration System

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Abstract: Natural refrigerants including CO2 have been recognized as the most promising working fluids and have been widely applied in refrigeration systems over the last decade. Owing to its attractive thermo-physical properties and negligible environmental impact, the CO2 refrigerant can be used as a replacement for convectional HFC working fluids. Normally, CO2 refrigeration systems can be classified into three different groups as indirect, cascade and all CO2 transcritical booster structures. The CO2 booster system has some advantageous over the others in terms of functions and sizes etc. However, the performance of such system still requires further investigation and improvement.

This study focused on the experimental investigation into the performance comparison of two CO_2 finned-tube gas coolers/condensers with different design structures and their effects on the overall performance of a CO_2 booster refrigeration system. The integrated CO_2 booster refrigeration system consisted of two variable speed semi-hermetic compressors, a gas cooler/condenser, a liquid receiver, electrically operated expansion valves, a medium temperature display refrigeration cabinet and an additional water/glycol load. The refrigeration system and especially the CO2 gas cooler/condenser had been comprehensively instrumented to enable detailed monitoring of the system and the heat exchanger at different operating states. Results for the system performance were obtained and analysed for different CO2 gas coolers/condensers. The results include the effect of heat exchanger designs and fan operations on the system performance. In addition, the controls of supercritical and subcritical pressures and cooling capacity are described.

Keywords: heat exchanger, CO_2 booster refrigeration system, COP, optimal designs, control strategies

1. INTRODUCTION

Refrigeration technology is an essential utility used in the food retail sector to ensure proper merchandising and safety of the food products by enabling the long-term storage for frozen foods and short-term storage for chilled foods. Two temperature levels are required in the food retail sector for frozen and chilled products -18°C to -35°C for frozen and -1°C to 7°C for chilled. Due to the huge amount of the refrigerant fluids used to cover the needs of food storage, the vapour compression refrigeration systems are responsible for greenhouse gas emissions (GHG) to the environment. The GHG emissions from refrigeration systems can be split into two main categories: direct and indirect emissions. The direct emissions are created by leakage of refrigerant with high global warming potentials (GWP) such as fluorocarbons including the HFC's. On the other hand, the indirect emissions produced from the energy required to operate the refrigeration systems (Tassou, 2011). Therefore, reducing both direct and indirect emissions is become an important objective for the researchers and public companies around the world. Potential actions to reduce the indirect emissions are reported including the new guidance for buying new refrigeration equipment with low cost indirect emissions, more sophisticated performance information for new large systems and preparation of guidance on indirect emissions reduction opportunities for existing systems (DEFRA-SKM Enviros, 2011). The actions can be taken to reduce the direct emissions divided into two sections for existing and new system installations. The suggested actions for the existing systems include the regularly maintenance of the equipment used. For the new installation system the use of more environmental friendly refrigerants such as CO_2 is recommended (DEFRA-SKM Enviros, 2011). CO2 is an ozone friendly substance with zero Ozone Depletion Potential (ODP), negligible direct Global Warming Potential (GWP<1), non-toxic and non-flammable. Along with the excellent environmental characteristics, the CO2 has favourable thermo-physical properties in terms of higher density, specific heat, volumetric cooling capacity, latent heat and thermal conductivity. The use of CO2 as a refrigerant create a great opportunities to eliminate entirely the direct GHG emissions to the atmosphere and diminish potentially the indirect effects by optimising the system designs and operations. The CO2 has 28 times higher density than the ammonia and 7 times higher than R-134a at the same temperature. The high gas density of CO2 enables a greater cooling capacity to be achieved comparing with other refrigerants for the same volume flow. This allows the use of smaller suction pipes and smaller compressor size. The use of smaller compressor decrease indirect emissions of the system (Pearson, 2014).

In the application of CO_2 as a refrigerant to food retail refrigeration systems, a number of different design approaches can be adopted that fall into three major categories: indirect/secondary, subcritical cascade systems and all-CO2 system. For an indirect system, the CO2 is commonly used as a two-phase secondary coolant while a HFC refrigerant such as R404A is conventionally charged in the primary side. The performance characteristics of the R404A indirect system using CO2 as a secondary refrigerant has been analysed experimentally and the optimum design parameters for this particular system was discuss (Yoon, 2014). In a cascade layout, the system constitutes of two single stage sub-systems, integrated by a cascade heat exchanger. The CO2 is applied in the low cascade side while a working fluid such as R404A, NH3 or hydrocarbon is operated in the high cascade side. The performance of subcritical cascade systems in supermarket applications has been reported by a number of investigators (Hinde, 2009). The performance of an ammonia/CO2 cascade refrigeration system for a supermarket application has been theoretically and experimentally investigated. The authors have reported that the cascade arrangement could provide better COP compared to a conventional R-404A system (Sawalha, 2006). All-CO2 system, the CO2 refrigerant is the only working fluid employed without need of a second refrigerant on the high pressure side. The main advantages of the CO2 booster system over other ones include the simpler design requirements, less installation cost and less environmental impacts from the refrigerant leakage comparing with cascade systems use a HFC's as a secondary fluid. However, the CO2 as a working fluid has a quite low critical temperature of 31.1°C and very high critical pressure of 73.8 bar. Consequently, the CO2 operates in relatively high pressure comparing with other refrigerants. In case the ambient temperature is higher than the critical temperature, the CO2 is not condensate in the high pressure-side of the system. The high pressure-side in the system can thus act as either a condenser or gas cooler depending on the ambient conditions. This necessitates high pressures which can lead to high power consumption. Therefore, the pressure of the gas cooler becomes very important operating parameters which need to be controlled in order to obtain best performance of the system. To understand the performance of a CO2 air cooled gas cooler, a series of tests were conducted at different operating conditions using a purposely designed test facility (Hwang, 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. Apart from the overall performance investigations of the CO2 gas coolers, the in-tube cooling processes of CO2 supercritical flow were extensively tested and correlated, which is helpful to understand the CO2 heat transfer and hydraulic behaviours in the heat exchangers (Srinivas, 2002; Yoon, 2003). The CO2 finned-tube gas cooler/condenser performances have been investigated extensively. Experimental and theoretical methods used to investigate the effect of their integration with associated systems are still rather limited (Chang, 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.

In this experimental research, two different sizes and designs of CO2 finned-tube gas cooler/condensers are purposely selected in order to be integrated with the existed CO2 booster system with specially designed measurement facilities. Extensive measurements based on different air volume flow rates have been carried out and some meaningful effects on the system performance have been obtained from the analysis of the test data.

These data include heat exchanger designs, air flow rates, supercritical and subcritical pressure controls and cooling capacity controls.

2. TEST FACILITY

The tested gas cooler/condenser and integrated CO2 booster system shown in Fig. 1. CO2 booster system consists of the gas cooler/condenser which located at the top of the refrigeration plant; inside the plant room an ICMT motorized valve and ICM by-pass valve, a liquid receiver, a CO2 accumulator and two semi-hermetic compressors have been installed. A medium temperature (MT) direct expansion (DX) open vertical multi-deck refrigerated display was used for the investigations and the tests were carried out at controlled conditions in an environmental chamber. An additional CO2/Brine load has been installed at the top of the plant room in order to balance the capacity of the compressor.

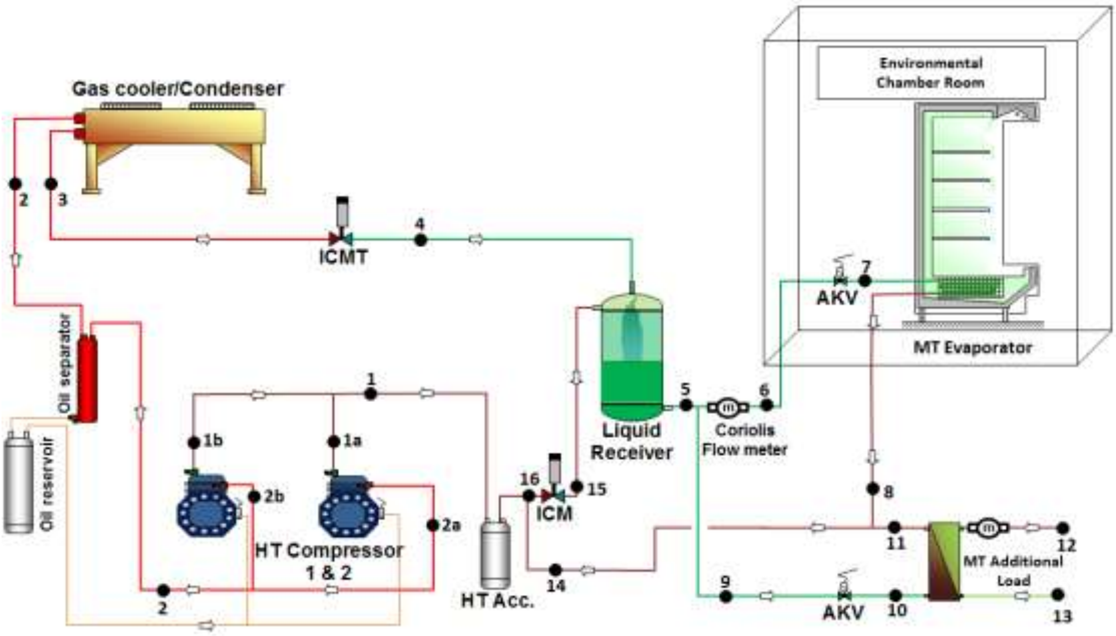

Figure 1 - CO2 Booster system test rig

The system is divided into three pressure sections, high, intermedium and medium; where the high pressure section begins at the outlet of the compressor and continues through the gas cooler/condenser and inlet line of ICMT motorized valve. Intermediate pressure section exists at the outlet of the ICMT valve and through the receiver where the flow is divided into gas and liquid. The gas phase returns back to the compressor suction line through the by-pass ICM valve. The liquid flows to the expansion valve where the medium pressure level begins. The liquid is expanded prior to MT display cabinet. The gas outlet from the MT flows back and mixing with the gas by-pass from the receiver before enter to compressor suction line and complete the circuit. The test facility is equipped with Danfoss controls and was comprehensively instrumented with pressure transducers, thermocouples and coriolis refrigerant flow meter to enable the detailed experimental investigation of the system and individual component under different operating conditions. Fig. 2 shows the relative installation position of the main components in the system.

The thermodynamic analysis and control optimisation of a similar CO2 booster system have been explained in more details by the authors (Ge, 2009; Ge, 2011). The overall system performance has been investigated experimentally at different operating conditions and controls when two different sized and designed gas coolers/condensers are installed separately. A purposely test rig has been built in order to examine extensively the performance of CO2 gas cooler/condenser with different structure designs. This purposely test rig allows to investigate the system under various operating conditions without affect from the ambient temperature.

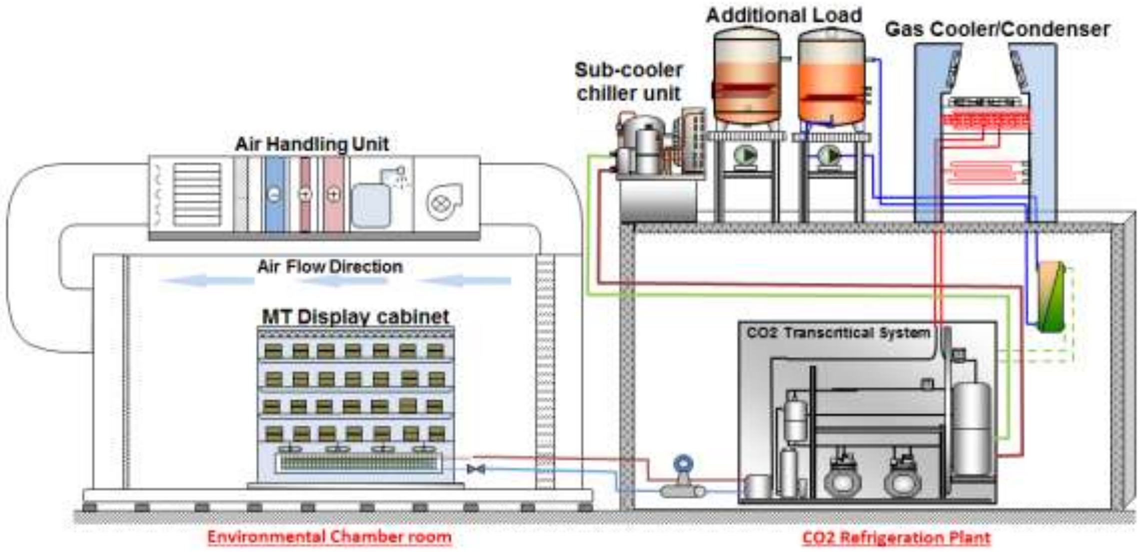

Figure 2 - Simplified diagram of the main component installation

This purposely gas cooler/condenser test unit shown in Fig. 3. The CO_2 heat exchanger is suspended tightly between two upright metal frames. Two propeller air fans are installed above the heat exchanger and controlled by variable speed control to maintain the requirement fixed air flow passing through. There are a number of smaller air fans installed oppositely along the direction of the pipe length, which will be switched on if the air inlet temperature is required to be higher than the ambient. The hot exhaust air from the coil will flow back through the return air tunnels then return air grills and mix with lower temperature ambient air flow. In the case the mixed air flow temperature is still lower than the required set point, an electric air heater which is installed at the air inlet section just below the heat exchanger will be controlled on to maintain the air inlet temperature at the designed temperature. With the variable controlled main air fans, recirculate air fans and air inlet heaters the gas cooler air on parameters, temperature and flow rate, can be well controlled to specified values.

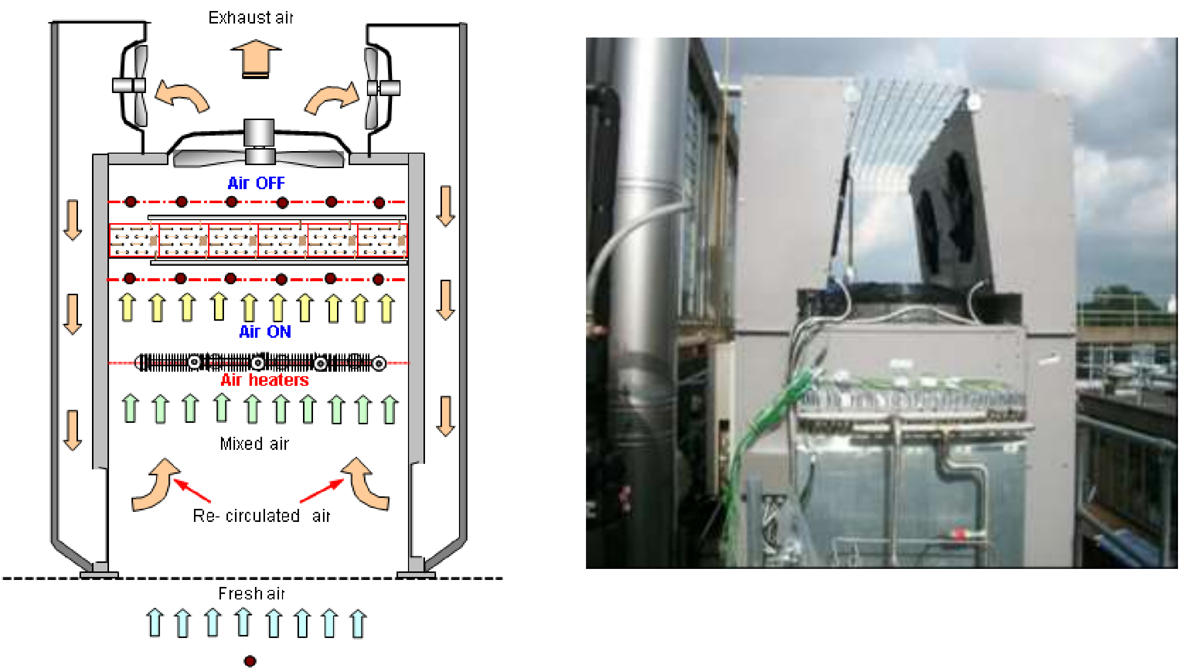

Figure 3 – Gas cooler/condenser test unit

The test rig has been instrumented comprehensively to detailed measurement data and overall performance description of the heat exchanger itself and its integrated CO_2 refrigeration system. The instrumentation devices used on the test unit are temperature and pressure measurements, air flow velocity and air pressure drop sensors. T-type thermocouples measure the air-inlet and air-outlet temperatures in 24 points respectively. For the refrigerant side, two pressure transducers are installed inside inlet and outlet headers of the heat exchanger to measure overall and heat exchanger refrigerant side pressure drops. A large amount of T-type thermocouples are attached and proper insulated on all the pipe bends along the pipes of one circuit to obtain the refrigerant temperature variation from the inlet to outlet. The refrigerant mass flow rate passing through the gas cooler/condenser it is calculated as the proportional of the heat balance air and refrigerant sides.

3. EXPERIMENTAL PROCEDURE

Fig. 4 shows the two different CO2 gas coolers/condensers which have been purposely designed and manufactured with different pipe arrangements and circuit numbers in order to experimentally examine the effects on the overall system. The coil number 1 is a three rows coil with four pipe circuits and total pipe number of 96. The overall dimension is 1.6m×0.066m×0.82m (L×D×H). Correspondingly, the coil number 2 has two rows and two circuits. The total number of pipe is 64 and overall dimensions 1.6m×0.044m×0.82m (L×D×H). Both coils use a copper pipe with 8mm outer diameter, aluminium fins with 0.16mm thickness and fin density of 453fins/meter. These two heat exchangers were installed in the CO2 system separately and tested at the same operating stages including intermedium and medium temperatures and pressures and air flow parameters. The tests conditions including air flow temperatures were varied from 19°C to 36°C and flow rates were modulated from 2000l/s to 2800 l/s through fan speed controls. Therefore, both heat exchangers operated at both subcritical and transcritical modes. In order to investigate the transient performance, the system was monitored from start up to the point it achieved steady state conditions. Investigations on steady state performance were carried out after the test plant reached the specified steady state conditions. Measured performance parameters from the instrumentation devices such as temperature, pressure and flow rate were logged by data logging every 20 seconds.

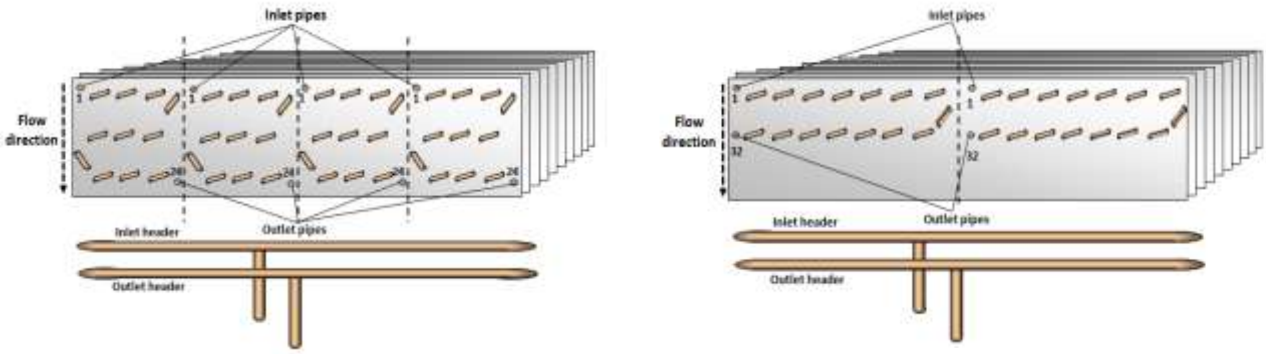

Figure 4 – CO_2 heat exchangers

To ensure the precise measurement, all measurement devices were calibrated carefully and the measurement uncertainty values are ±0.5°C, ±5%, ±0.3% and ±0.035% for temperature sensors, air velocity meter, pressure transmitters and refrigerant mass flow rate respectively. Table 1 illustrate the different operating conditions for the experimental investigations of both heat exchangers.

Table 1 - Experimental test conditions

Operating Mode	Coil Type	Air Temp. inlet °C	Main Fan Speed %	Air Volume l/s (x 1000)	Refrigerant Temp. inlet °C	Refrigerant pressure inlet bar
Condenser	3 Rows	19 - 27	50 – 60 - 70	20-24-28	69 - 85	60 - 73
Condenser	2 Rows	19 - 27	50 – 60 - 70	20-24-28	69 - 85	60 - 73
Gas Cooler	3 Rows	28 - 34	50 – 60 - 70	20-24-28	86 - 115	75 - 90
Gas Cooler	2 Rows	28 - 34	50 – 60 - 70	20-24-28	86 - 115	75 - 90

4. THERMODYNAMIC ALALYSIS OF THE SYSTEM

Energy and mass balance equations applied for individual components of the CO_2 system. The refrigeration capacity of the evaporator of the MT display cabinet was calculated using the enthalpy difference across the coil and the mass flow rate of the refrigerant. The enthalpy of the refrigerant liquid entering the evaporator and the refrigerant vapour leaving the evaporator were determined from the measurements of temperature and pressure of the refrigerant inlet of the expansion valve and outlet of the evaporator coil respectively. The thermo-physical properties of the CO2 used for first law of thermodynamic analysis in this paper were calculated by using Engineering Equation Solver (EES ®, 2015).

Equation 1: Mass and Energy balance of the MT evaporator. (Numbers refer to Fig. 1.)

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_{MT}$$

$$\dot{Q}_{MT} = \dot{m}_{MT}\,(\Delta h)_{ref} = \dot{m}_{MT,air}\,c_p\,(\Delta T)_{air}$$

Where:

- $\dot{m}$ = refrigerant mass flow rate (kg/s)
- Q = refrigeration capacity (kW)
- c_p = Specific heat capacity (kJ/kg K)
- Δh = enthalpy difference (kJ/kg)
- ΔT = temperature difference (K)

The additional refrigeration capacity load was calculated from the energy balance between brine and refrigerant side assuming adiabatic heat transfer. The brine mixture properties were determined by using the equations form M. Conde Engineering (2011).

Equation 2: Mass and Energy balance of the Additional Load evaporator. (Numbers refer to Fig. 1.)

$$\dot{m}_{12} = \dot{m}_{13} = \dot{m}_{AL,brine}$$

$$\dot{m}_{10} = \dot{m}_{11} = \dot{m}_{AL,ref}$$

$$\dot{Q}_{AL} = \dot{m}_{11}\,(\Delta h)_{ref} = \dot{m}_{12}\,c_p\,(\Delta T)_{brine}$$

Where:

- $\dot{m}$ = mass flow rate (kg/s)
- Q = refrigeration capacity (kW)
- c_p = Specific heat capacity (kJ/kg K)
- Δh = enthalpy difference (kJ/kg)
- ΔT = temperature difference (K)

The total refrigerant mass flow rate passing through the heat exchanger is calculated by the energy balance of air and refrigerant sides and is assumed to be the same passing through the compressor, ICMT motorized valve and at the inlet of the receiver where the mass flow is divided into gas and liquid form.

Equation 3: Calculation of the refrigerant mass flow rate.

$$\dot{m}_{ref}\,(\Delta h)_{ref} = \dot{m}_{air}\,c_{p,air}\,(\Delta T)_{air}$$

In order to calculate the compressor power consumption, a mathematical model based on the manufacture data (GEA BOCK Compressors, 2009) has been created. The model is divided into subcritical and transcritical operations. However, the former is a function of condensing and evaporating temperatures while the later one is dependent on the compressor suction and discharge pressures. The coefficients from these equations are correlated and listed in Table 2.

Equation 4: Compressor power consumption – Subcritical operation

$$W_{com,sub} = c_1 T_{ev}^2 + c_2 T_{ev} + c_3 T_{cd}^2 + c_4 T_{cd} + c_5 T_{ev} T_{cd} + c_6$$

Equation 5: Compressor power consumption – Transcritical operation

$$W_{com,trans} = c_1 P_{suc}^2 + c_2 P_{suc} + c_3 P_{dis}^2 + c_4 P_{dis} + c_5 P_{suc} P_{dis} + c_6$$

Where:

- T_{ev} = evaporating temperature (K)
- T_{cd} = condensation temperature (K)
- P_{suc} = Suction pressure (Bar),
- P_{dis} = Discharge pressure (Bar)

It is noted that the manufacture data of compressor power consumption was measured at constant superheat 10 K at compressor inlet. If the actual suction superheat is different from 10 K, the following correlation factor c_{suc} is applied:

Equation 6: Correction factor

$$c_{suc}= \frac{v_{exp}}{v_{10K}}$$

Where:

- v_{exp} = experimental specific volume at compressor inlet (m^3/kg)
- v_{10K} = specific volume at 10K superheat (m^3/kg).

Table 2 - Correlated coefficients for the mathematical model

Subcritical	c1	c2	c3	c4	c5	c6
	-1.61804	303.794	-0.23315	-299.358	1.955376	-5696.67
Transcritical	c1	c2	c3	c4	c5	c6
	-1.79591	40.16467	-0.12544	21.38474	1.473649	2520.822

5. TEST RESULTS & DISCUSSION

The COP of the CO_2 refrigeration system is defined as Eq. 7. The refrigeration capacity of the MT cabinet and the additional load can be determined using the thermodynamic relationships from Eq. 1 and 2. The work done by the CO_2 compressor can be calculated by Eq. 4 and 5 for subcritical and transcritical respectively. The power consumption of the extraction fans was determined by recording the power by using a power quality analyser.

Equation 7: System COP

$$COP = \frac{Q_{AL}+Q_{MT}}{W_{com}+W_{fan}}$$

Where:

- $\dot{Q}_{AL}$ = cooling capacity of additional load (kW)
- $\dot{Q}_{MT}$ = cooling capacity of refrigeration cabinet (kW)
-W_{com} = compressor power consumption (kW)
- W_{fan} = extraction fans power consumption (kW)

Fig. 5 (a) shows the extraction fan power consumption with respect of different air volumetric flow rates. Apparently, the fan power consumption increases with higher air volumetric flow rate and the increase rate is much higher for a higher air flow rate, which effect on the final COP calculation of the system. The influence of the air volume flow rate on the air side pressure drop for both heat exchangers is illustrated in Fig. 5 (b). From the graph, it is seen that the air side pressure drop increases with higher air flow rate and more pressure drop in the 3-row coil due to the higher row number. On the other hand, the higher air flow rate will enhance the air side heat transfer and subsequently improve the system performance.

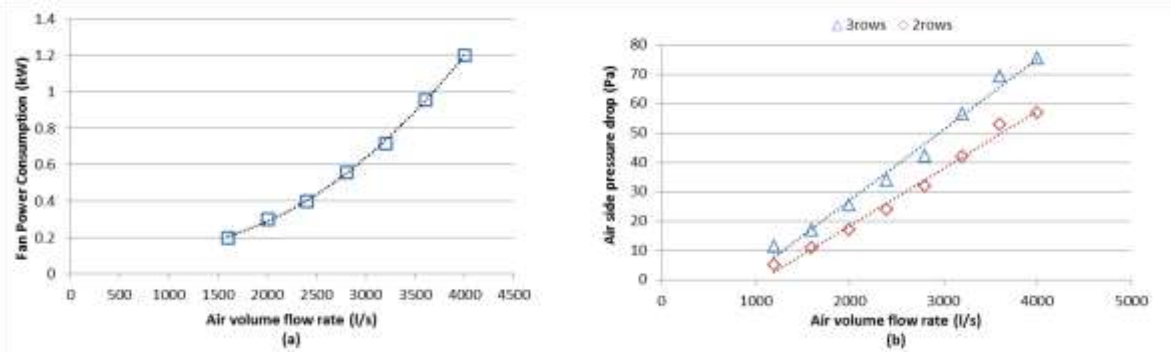

Figure 5 – (a) Fan power consumption for different air volume flow rates

(b) Air side pressure drop with air volume flow rates

The intermediate pressure is effect on the cooling COP of the system. The intermediate pressure is controlled by the ICMT valve and ICM by-pass valve; by adjusting both we can control the liquid level inside the receiver. With lower intermediate pressure the refrigerant quality will diminish at the inlet of the receiver and the enthalpy values will be lower at the inlet of MT evaporator (Ge, 2011). In such circumstances, the COP of the system will be increased. Therefore, during the experimental tests the intermediate pressure controlled to be between 30 to 31 bar, slightly higher than the MT evaporating pressure. Moving to the higher pressure side of the system is easily to define that is an optimal high side refrigerant pressure for each ambient temperature in which the COP reaches the maximum values. The optimum high side pressure is operate in irrespective from the intermediate pressure side and is only proportional of the air-inlet temperature and coil structural design. Fig. 6 illustrates the relationship of heat exchanger discharge pressure and air-on temperature at three different fan speeds of 50%, 60% and 70% of the total corresponding to 2000l/s, 2400l/s and 2800 l/s respectively.

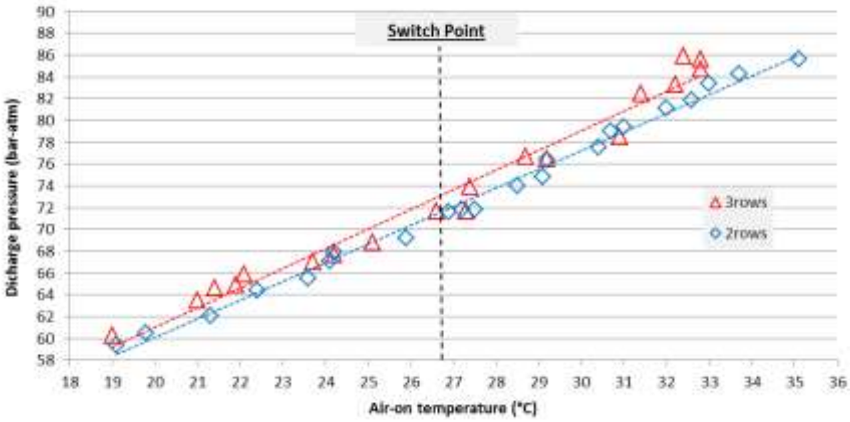

Figure 6 - Relationship between air-inlet temperature and heat exchanger discharge pressure

It is clear from the linear relationship that there is no any effect from the fan speed on the discharge pressure of the system. In transcritical mode the control strategies involve a PID controller which record the air-on temperature on the gas cooler/condenser and tries to maintain the optimal pressure in the gas cooler. On the other hand, the temperature difference (ΔT) subcooling controls can be used for the subcritical operation. The refrigerant temperature outlet of the condenser used to regulate subcooling degree based on the reference pressure. The condensing temperature of the system varies between 20°C to 28.4°C which is lower than the critical point of the CO_2 refrigerant. The variation of the system cooling capacity and compressor power consumption with respect of varies air-inlet temperatures is shows in Fig. 7(a). The graph illustrates the results for 70% extraction fan speed at 2800l/s. Similar trends can be obtained for 50% and 60% fan speeds. The cooling capacity for the 3-row gas cooler/condenser is higher while the compressor power consumptions are nearly the same for both coils. The higher cooling capacity is the proportional of the bigger refrigerant pressure drop across the coil for the 3-rows coil. This will lead to a higher COP for the system with the 3-row CO_2 coil and different fan speeds, as shown in Fig. 8. M, the COP for both circumstances decreases with higher air-on temperature. The higher COP was found on the 60% main fan speed, corresponding to 2400 l/s of air passing through the coil, and is a proportional to the lower power consumption of the fan when compared with 70% of fixed fan speed.

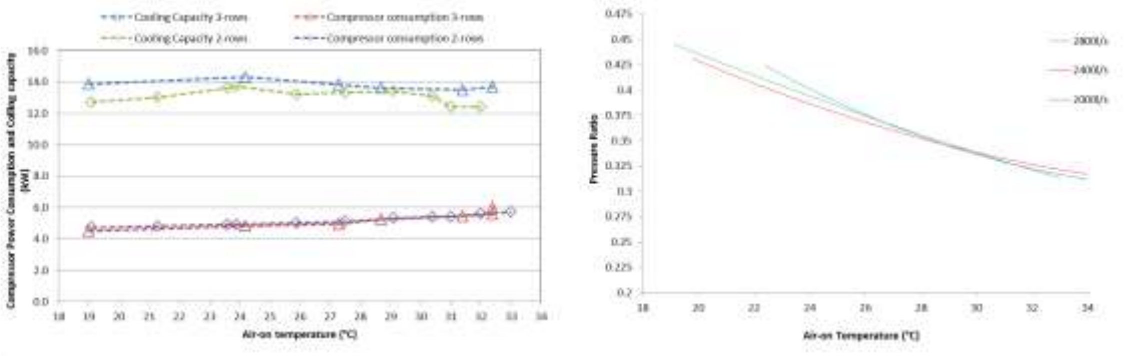

Figure 7 – (a) Effect of air-inlet temperature in compressor power consumption & system cooling capacity

(b) Compressor pressure ratio in different air-inlet temperatures

The system control strategies for this experimental investigation involve a compressor variable speed control system. Fig. 7 (b) shows the pressure ratio with respect of air-inlet temperature for three different air extraction fan speeds. The graphs show that the air fan speed is not effect to the compressor power consumption. On the other hand the power consumption of the compressor it is dependent on the air-inlet condition. As the air-inlet temperature increased the system trying to decrease the pressure ratio and maintain the isentropic efficiency of the compressor in order to keep the power consumption in lower levels.

Test results based on a fixed air extraction fan speed, constant subcooling at 2K for subcritical cycles and constant approach temperature in a transcritical mode. The refrigerant mass flow rate is fairly constant and changed only when the system move from subcritical to supercritical operation and the evaporating temperature was - 8°C (± 1°C).

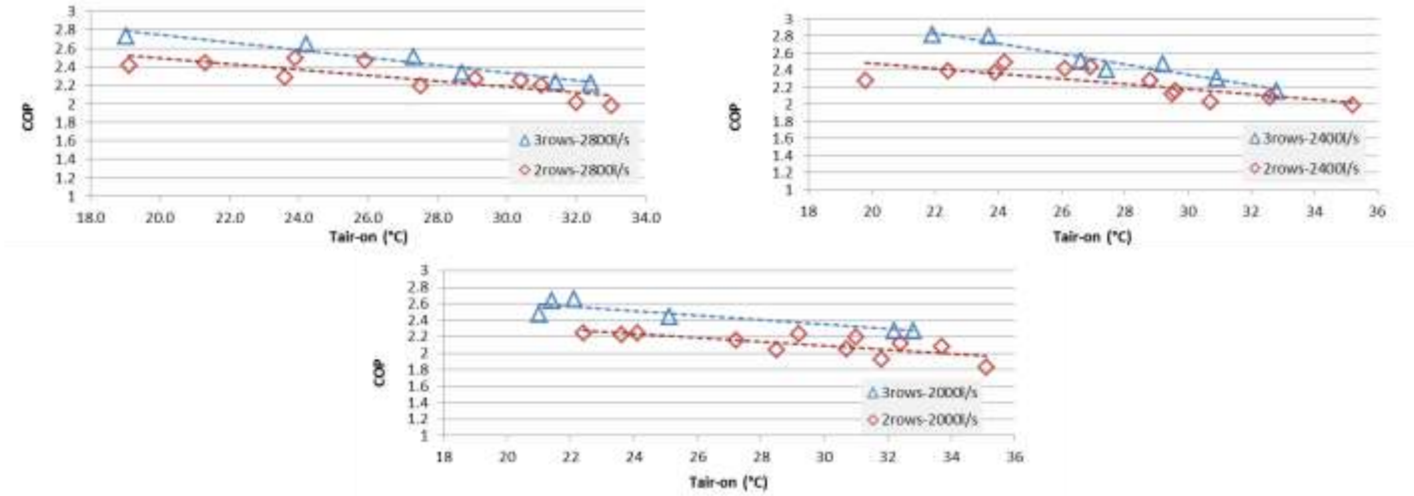

Figure 7 - Variation COP with air-inlet temperature

Take in account the uncertainty of the experimental measured variables which include air mass flow rate and air temperature, refrigerant temperatures, refrigerant pressures, refrigerant mass flowrate and assuming that the individual measurements are uncorrelated and random, the uncertainty in the calculation of COP was determined, using the Engineering Equation Solver (EES) software, to be between ± 5.52% and ± 6.1% for both 3-row and 2-row heat exchanger respectively. The display cabinet evaporator cooling capacity uncertainty calculation was found to be ± 0.77% and the additional load cooling capacity uncertainty calculation is relatively high because the calculation involved the brine side of the system and it's found to be ± 11.22%.

6. CONCLUSION

The CO_2 booster refrigeration system has been demonstrated as a promising solution for use in supermarket applications. This experimental investigation focuses on the effect of different designs of CO2 gas coolers/condensers on the overall performance of a CO2 booster refrigeration system. To achieve this target, two different heat exchangers were designed, manufactured and used in the booster refrigeration system. A number of experiments were carried out with different operating conditions and control strategies of the system. The experimental results show that the system COP and cooling capacity have been improved with the larger CO_2 heat exchanger. On the other hand, the air flow rate and air-side pressure drop have also been investigated. Finally, to achieve the maximum COP of the system a compromise balance has to be considered in the coil design.

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