DESIGN AND EXPERIMENTAL ANALYSIS OF A CARBON DIOXIDE TRANSCRITICAL CHILLER FOR COMMERCIAL REFRIGERATION

M. BERNABEI (b), L. CECCHINATO (a), M. CHIARELLO (a), E. FORNASIERI (a)

(a) Dipartimento di Fisica Tecnica, Università degli Studi di Padova, via Venezia, 1
Padova, I-35131, Italy
Fax: +390498276896, ceck@unipd.it
(b) SCM frigo S.r.l., Strada Zona Industriale, 10
Sant’Angelo di Piove di Sacco (PD), I-35020, Italy
Fax: +390499704947, mirko.bernabei@scmfrigo.com

ABSTRACT

The development of refrigerating units operating on natural working fluids is a topical challenge for refrigeration companies. Carbon dioxide is an interesting solution for commercial refrigeration and in perspective for air-conditioning systems. Because of its low critical temperature, CO$_2$ main drawback is its low cycle energy efficiency. Thus the refrigerating machines operate according to a transcritical cycle, also for relatively low temperature of the hot sink. SCM Frigo carried out a research project concerning the development of a carbon dioxide transcritical chiller for refrigerating propylene glycol down to -8 °C supply temperature. The aim of the project was at optimising the cycle energy efficiency while assuring reliable operation and simple management of the unit. To develop a highly effective unit, outstanding component manufacturers were involved. The whole project was coordinated by the University of Padova. Several experimental tests were carried out, testing the chiller at external temperatures ranging from 16 to 35°C.

1. INTRODUCTION

Carbon dioxide is one of the oldest refrigerants, as it was already employed at the end of the nineteenth century, mainly where safety was mandatory. When synthetic refrigerants, in the thirties, came into use, CO$_2$ began its decline, because the new fluids provided best energy efficiencies with cheaper and more reliable equipment. High critical pressure (7.384 MPa) and rather low critical temperature (31.06°C) summarise the main drawbacks of carbon dioxide. Nevertheless today CO$_2$ is gaining more and more favour thanks to its very low environmental impact and safety. In 1994 the Norwegian prof. Lorentzen first proposed again this refrigerant as a working fluid for compression vapour inverse cycles. Since then many other authors have extensively studied such applications. Although CO$_2$ shows poor thermodynamic properties with reference to the energy efficiency of a traditional vapour compression inverse cycle, it is seen as an effective solution to the problem of the global warming of anthropic origin. In fact, its ODP is zero and its GWP is negligible, even zero, if the fluid is recovered from waste of industrial processes.

Because of its low critical temperature, a CO$_2$ system is different from a traditional one and often operates according to a simple transcritical cycle when the temperature of the hot sink exceeds a specific value (15-20 °C for atmospheric air); in this case the upper pressure is higher than the critical one and heat transfer does not involve two phase transformation (condensation) but only gas cooling (consequently the heat exchanger is named gas cooler). A transcritical cycle behaves differently from a traditional one, especially with reference to the function of the throttling valve and its operation. Many Authors have dealt with these topics, but the contributions of Casson et al. (2003), Kim et al. (2003), Cavallini and Zilio (2006), Nekså (2002), Groll (2001), Lorentzen (1994) appear to deserve a mention for accuracy and clearness.

In commercial refrigeration, CO$_2$ may be an important substitute for the HFC refrigerants normally used. Up to now, on a global basis, commercial refrigeration is probably the application where the refrigerant emissions are the highest. In fact, according to IPCC (2005), this application is responsible for 40% of the total annual refrigerant emissions. Annual leakage rates higher than 30% of the refrigerant charge are found when performing a top-down estimate (Palandre et al., 2004). CO$_2$ has no harmful effects if leaked to the atmosphere and is non-flammable and non-toxic, thus it is also safe to the immediate environment. Different systems operating with CO$_2$ as a refrigerant have been designed and installed in the last years. In northern Europe carbon dioxide has become a real alternative for commercial refrigeration plants.

The application of CO$_2$ in commercial refrigeration has been analyzed in detail both theoretically and experimentally (Girotto and Neksa, 2002), (Schiesaro and Kruse, 2002), (Cecchinato et al., 2007) and...
different plant concepts were shown by different authors (Girotto, 2005), (Eggen and Aflekt, 1998). Considering only centralized systems, the possible uses of CO₂ are (i) as secondary refrigerant (ii) as a primary refrigerant in the low temperature stage of a cascade system; (iii) in all-CO₂ centralized systems with the low temperature stage in cascade (iv) in all-CO₂ centralised systems with separated circuits for LT and MT service, both rejecting directly to the environment (or to another hot sink).

2. THE COMPONENTS OF THE SYSTEM

The chiller was planned mainly to serve refrigerating systems operating with secondary fluids for retail refrigeration at temperature above 0°C. It intends to meet mainly the North Europe customers requirements. It also can serve as the upper stage of a cascade unit where the lower stage is a subcritical CO₂ refrigerating circuit for low temperature service. Subcritical cycle operated on carbon dioxide is a rather common choice when environmentally safe equipment is required, but in this case the problem is the upper stage, since the most common options are ammonia or hydrocarbons (not safe), or a synthetic refrigerant (not environmentally compatible). Provided that energy efficiency and reliability be assured, CO₂ is absolutely safe for persons and environment and therefore can be a good candidate.

The temperature of the secondary refrigerant at the evaporator outlet was set at -8°C, and at the evaporator inlet at -4°C; the evaporation temperature is -11°C, with a refrigerating capacity of 67 kW, at 35°C external temperature.

The schematic of the refrigerating circuit is shown in Figure 1.

The evaporator (1) is a plate heat exchanger with 80 plates, operating with 5°C superheating. The plates overall dimensions are 694x304 mm, the total heat transfer area is 16.93 m² and the volumetric flow rate of the secondary refrigerant is 15.77 m³/h.

The superheated carbon dioxide is collected inside an intake manifold of 0.01 m³ internal volume; the compressors (2) are three single-stage, semi hermetic, 4-cylinder reciprocating units; their displacement is 138 cm³ each and their maximum admissible pressure is 130 bar; the rotational speed is about 1450 rev/min.

The heat recovering device (3) is a 40-plate heat exchanger; the plate overall dimension are 376x119 mm, the total heat transfer area is 1.56 m² and the volumetric flow rate of the secondary refrigerant is 3.5 m³/h. Its maximum operating pressure is 140 bar, being the plates reinforced by a frame (two 40 mm thickness steel plates linked by tie-rods). The recovered heat can be utilized for satisfying the possible needs of tap water or floor heating.

The exceeding heat flow is rejected by an air condenser/gas cooler (4). It is a horizontal finned coil, with 2 circuits, 2 rows and 88 tubes per row. The overall dimensions of the aluminium finned pack are 3200x2200x87 mm, the internal diameter of the copper tubes is 6.52 mm, 1 mm thickness, 25 mm tube spacing and 21.65 mm row spacing. The finned coil is cooled by 4 fans and its geometry allows approach of about 3°C between the refrigerant temperature at the outlet and the external air temperature, assuming design conditions without heat recovering. The nominal volumetric flow rate of air is 30000 m³/h and can be controlled by variation of the rotational speed of the fans.

The finned coil is equipped by a water spraying system, made up of 12 nozzles (3 for each fan), placed 300 mm under the finned pack. The inlet nozzle diameter is 0.91 mm, the outlet nozzle diameter is 1.1 mm, with a sprinkling angle of 120°; the water flow rate is 0.355 l/min for each nozzle, under a pressure of 2.5 bar. The spray system is intended for lowering the air temperature when the its temperature is higher than 30°C: in this way a remarkable increase of the refrigerating capacity and the COP is achieved.

The throttling process of the refrigerant downstream the gas cooler is performed in two stage. The carbon dioxide is firstly throttled through a back-pressur valve (V1), having the function, during transcritical operation, of setting the value of the upper cycle pressure, and then is throttled through a thermostatic expansion valve (V2), having the function of properly feeding the evaporator.

After the first stage of throttling, the two-phase fluid is collected into a flash tank (5), with 0.07 m³ internal volume; this tank compensates for the variations in the void fractions inside the heat exchangers, allowing their optimal filling as the operating conditions vary. It also operates as a liquid-vapour separator. The flash vapour is drawn off by a valve (V3) and merged into the vapour at the evaporator outlet, so keeping the pressure inside the flash tank at a safe value. This fluid is then superheated by an internal heat exchanger (6), equipped with 90 plates, operating in counterflow with the saturated liquid drawn out of the flash-tank. The plate overall dimensions are 287x117 mm, the total heat transfer area is 2.46 m² and the higher working pressure is fixed at 65 bar: even in this case, a similar frame as the one of the heat recovering exchanger (3) helps to withstand the maximum admissible pressure.
Heat exchanger (6) provides a double advantage: it both assures a suitable superheating at the compressor inlet (as required by the manufacturer at 10°C, as the minimum value) and provides increased refrigerating capacity, being the thermostatic expansion valve V2 fed with subcooled liquid. Irrespective of their different functions, all the valves are electronic controlled devices, designed to withstand high pressure.

Figure 1: Layout of the refrigerating circuit.

3. THE SYSTEM CONTROL MODES

In a refrigerating system operating on R744 refrigerant, the throttling valve control algorithm is essential for achieving the highest energy efficiency. During transcritical operation the upper cycle pressure is not any longer imposed by heat transfer, but represents an independent variable subjected to optimisation and its value is determined by the flow coefficient of the throttling valve. Moreover, when the temperature of the external cooling medium is rather low, subcritical cycle is the most convenient choice and the control algorithm has to manage the switch from transcritical to subcritical operation, when the back-pressure valve is kept open.

The chiller can work under two different control modes of the throttling devices. They are defined according to the thermodynamic cycles, subcritical or transcritical, which they realize. The goal is to realize the highest COP value, under different working conditions, while complying with the technological constraints. Among them, the most important is the higher allowed pressure in the medium pressure line: it is set at 62 bar, being 65 bar the maximum admissible pressure of the internal heat exchanger. The diagram of the upper pressure against the outlet gas cooler temperature, as established by the valve control algorithm, is plotted in Figure 2. In the mode “back-pressure”, the operation of the chiller is optimised operating in transcritical conditions; the back pressure valve V1 keeps the optimal gas cooler pressure while the working conditions vary, by controlling the upper pressure as a function of gas cooler temperature at its outlet. The correlation between these two variable, shown in Figure 2, was chosen on the basis of the real operation of the refrigerating circuit as determined by a simulation model of the full system, so taking into account the variation of the compressor and heat exchanger performance. The optimised controlling curve can involve at the lowest values of temperature and pressure, even in the “back-pressure” mode, subcritical operation. This means that the back-pressure valve V1 determines the condensing pressure by varying the flooded length of tubes at the
outlet of the finned coil acting as a condenser; only a further drop of the temperature/pressure causes full opening of the valve and the switching to the second control mode. V3 valve is a back-pressure valve that draws out some vapour from the flash tank to the low pressure line, throttling down the flash vapour from the intermediate to the evaporating pressure, in order to keep the pressure inside the flash tank at a safe value.

The second control mode, called “thermostatic”, is intended for optimising the COP of the chiller working under subcritical conditions. In this case, the valve V1 is fully open and the heat exchanger (4) operates as a condenser while its internal pressure depends on the temperature of the cooling agent. The switch from a controlling mode to another is a critical operation. The controller changes from the mode “back pressure” to the mode “thermostatic” when the carbon dioxide enthalpy at the gas cooler outlet becomes lower or equal to the corresponding enthalpy value of the saturated fluid at 54 bar; being the effective pressure at 62 bar, the refrigerant state is subcooled liquid.

In this mode the back-pressure valve V1 is kept open and the pressure of the flash tank is lower than 62 bar; since the carbon dioxide at the flash tank inlet is liquid (saturated or subcooled), there is no need to remove vapour and the valve V3 consequently is kept close.

During the switching from “back-pressure” to “thermostatic” mode, the valve V1 opens gradually, to avoid too high pressure inside the internal heat exchanger, while, at the same time, the flash valve V3 closes. Finally, the controller changes from the “thermostatic” mode to the “back pressure” mode when the upper pressure rises at 62 bar; then the back pressure valve V1 gradually closes and, at the same time, the flash valve V3 opens, so as to optimise the gas cooler pressure and to lower the flash tank pressure below 54 bar.

![Valve control mode](image)

**Figure 2:** Optimal gas cooler pressure vs. outlet gas cooler CO₂ temperature. The solid line shows the pressure curve in the mode “back-pressure”, the dashed line in the mode “thermostatic”.

### 4. EXPERIMENTAL RESULTS

Several experimental tests of the operation of the system were performed in June/July 2007 at the site of the Company at Vigorovea (PD – Italy); the cooling medium of the gas cooler was outside air not controlled in temperature.

The hydraulic circuit used for the tests is shown in Figure 4. The secondary fluid for refrigeration and heat recovery was antifreeze mixture water/propylene glycol; the chilled secondary fluid coming from the evaporator (line 1 in Figure 4) is warmed by a series of finned coil heat exchangers (dry-heater), placed outside the testing room and then collected inside a tank of 2 m³ internal volume, having the function of heat storage. The hot secondary fluid coming from the heat recovery exchanger through the line 2 is introduced into the thermal storage tank where compensates for some amount of the refrigerant effect of the cycle. The
temperatures of both the cool and the hot streams of secondary fluids entering respectively the evaporator and the recovery heat exchanger are controlled by mixing the fluid flows through two three-way valves that provide a way for bypassing, respectively, the external heat exchangers and the thermal storage. The volumetric flow rates of the hot and cool secondary fluid are measured by two magnetic flow meter. The filling system of the whole test rig is represented by the line 5 in Figure 4.

The nominal performances of the pumps (at 2900 rpm) are 30÷54 m$^3$/h volumetric mass flow rate and 32.5÷36.5 m hydraulic head. The external dry-heaters are finned coils, with a face area of 3.9 m$^2$ (2610 x 1500 mm), 360 m$^2$ heat transfer area (referred to outside area), 2.1 mm fin spacing), 4 rows and 50 tubes per row.

The experimental data are compared with the results of the simulations carried out by a software developed by the Dipartimento of Fisica Tecnica of the Università degli Studi di Padova, able to simulate the operation of different kinds of refrigerating installations. This software was utilised for two tasks:
to simulate the operation of the chiller under different conditions in order to develop the valve control algorithm;

to verify the experimental data against the simulation results for testing the accuracy of the measurements carried out in the testing rig.

4.1. Standard operation

The standard operation of the chiller is defined by constant volumetric flow rate of the secondary refrigerant (15.8 m$^3$/h) and constant temperature of the secondary refrigerant at the chiller inlet (-4°C). Under these constraints, during the test period, the refrigerating capacity, the electrical power consumption and the heat recovered have been measured; these data have been compared with the same data calculated through numerical simulation with external temperatures ranging between 16 and 35°C.

![Figure 5: COP vs. external temperature: comparison between experimental data and simulation results](image1)

![Figure 6: Refrigeration capacity vs. external temperature: comparison between experimental data and simulation results](image2)

![Figure 7: Electrical power vs. external temperature: comparison between experimental data and simulation results](image3)

![Figure 8: Heat recovery capacity vs. external temperature: comparison between experimental data and simulation results](image4)

In the diagram of Figure 5, the COP values of the real system and the ones resulting from the simulations are plotted against the external temperature; as can be easily predicted, the chiller efficiency continuously fades as the air temperature increases, but it never lowers below 1.65. The optimisation of the switch between the two valve control modes (back-pressure mode and thermostatic mode) results in a smooth curve of COP, without any discontinuity. The refrigerating capacity and the electrical power are plotted in Figure 6 and 7, respectively; the experimental data agree fairly well with the simulation results. In the case of heat recovery, the actual thermal power recovered by the desuperheating is slightly lower than the predicted one (Figure 8). This is due to the lower value of secondary fluid mass flow rate used in experimental tests with respect to the design value. However it is worth remarking that high thermal power (40÷50 kW) can be recovered with a rather small-size heat exchanger. This amount of thermal energy is free of charge and can be used, for
example, to provide domestic hot water heating or the thermal energy required by the air handling unit used in an all-air conditioning system.

In Figure 9 three representative temperatures of the carbon dioxide, measured or predicted at low pressure side of the circuit, are plotted against the external temperature. It is worth noticing the high value of superheating at the compressor suction; the minimum superheating recommended by the compressor manufacturer was 10°C, but this value was doubled by the designer of the unit for safety reasons, keeping in mind that the compressor was a prototype. In such conditions the regenerative heat exchanger 6 in Figure 1 is essential for effective operation of the evaporator, which otherwise would have operate with a large amount of dry surface. In the actual situation, most of the superheating occurs inside the heat exchanger 6 and the average superheating at the evaporator outlet is roughly 5°C.

![Figure 9: Compressor suction temperature and evaporator inlet and outlet temperature: comparison between experimental data and simulation results](image)

4.2. "Night" operation (partial load)

Finally, some tests were carried out, which simulate the chiller operation during the night or over a period when the normal commercial activity is stopped.

![Figure 10: COP vs. external temperature in night running: comparison between experimental data and simulation results](image)

In such a situation, the heat load is reduced and the refrigeration capacity required to the system is lower, so significant energy saving could be attained if the temperature of the secondary fluid delivered to the display
cabinets is increased; therefore an experimentation was performed changing the secondary refrigerant inlet/outlet temperature from –4/-8 to –1/-4°C. As shown in Figure 10, the average increase in COP is about 22% and a good agreement between experimental and predicted data was obtained.

5. CONCLUSIONS

SCM Frigo s.r.l., in cooperation with the Dipartimento of Fisica Tecnica of the Università degli Studi di Padova, designed and manufactured a carbon dioxide chiller operating according to a transcritical cycle. To verify proper operation of the system under different operating conditions, a wide experimental test campaign was carried out. The chiller has shown good reliability; during the experimental tests, neither breakdown or any serious drawback occurred and the machine was properly working even in long-time tests (up to 15 days of continuously running). In particular, no instability problems occurred during the switch from a valve control mode to the other (back-pressure and thermostatic), according to the developed controlling procedure. The experimental data agree fairly well with the simulations results; the system provides the expected refrigerating capacity and moreover it can supply thermal power for low-temperature heating applications. The measured energy efficiency agrees with the predicted one and the COP ranges from 1.8 to 3.3, as a function of the operating conditions. The prototype was developed using components available on the market; this proves that the current technology is ready for supplying the market with carbon dioxide refrigerating plants in large numbers. This fact, together with the good values of energy efficiency achieved, is a necessary condition to allow a wide diffusion of this type of refrigerating machine, as a substitute for the traditional systems working with halocarbons.

REFERENCES