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A STUDY ON CYCLE DESIGNS FOR LIGHT COMMERCIAL CO_2 REFRIGERATION SYSTEMS

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Abstract: *This paper compares three different cycle designs – capillary tube, thermostatic expansion valve and medium-pressure receiver – by experimentally evaluating the system behavior under variable ambient temperature. To this end, a detailed analysis was carried out focused on the effects of both the cycle design and ambient temperature on the system performance. It is shown that the thermostatic valve and the capillary tube designs have contrasting effects on the evaporator superheating and discharge pressure and that their effects on the overall system performance are quite similar. It is also shown that the medium-pressure receiver design is able to simultaneously control the superheating and the discharge pressure, leading to a performance improvement at low ambient temperatures. At high ambient temperatures no significant differences between the studied designs were observed.*

Keywords: *capillary tube, thermostatic valve, carbon dioxide, cycle design*

1. INTRODUCTION

Environmental concerns allied to restrictions imposed by regulatory agencies have promoted the use of natural fluids as an alternative to the synthetics CFCs, HCFCs and HFCs. In this regard, carbon dioxide (CO_2, R-744) is a promising candidate due to its favorable physical properties, safety aspects and environmental-friendly characteristics. Moreover, CO_2 has a broad application in processes which involve heat pumps and refrigeration (Lorentzen, 1995; Kim *et al.*, 2004). The light commercial refrigeration sector (beverage coolers, display cabinets and chest freezers), in particular, is a potential niche for CO_2 application provided that the environmental footprints of the CO_2-based systems are at least comparable to those of the ordinary refrigerant systems. With the current technology, CO_2 systems working at high ambient temperatures are strongly penalized in this regard since their efficiencies tend to be lower than those employing synthetic refrigerants. This is mainly due to the low critical temperature of this fluid (31.1^oC), which is responsible for the transcritical operation mode even at quite low ambient temperatures. Under such conditions the discharge pressure plays a significant role in the system performance, and a fine tune adjustment of this parameter is thus required in order to maximize the cooling capacity and the coefficient of performance (COP). The optimum discharge pressure varies according to the gas cooler outlet temperature and also with the evaporation pressure (Gosney, 1982; Kauf, 1999; Kim *et al*., 2004; Sarkar *et al.*, 2004; Cabello *et al*., 2008; Aprea and Maiorino, 2009; Cecchinato *et al.*, 2010). Different approaches may be considered to adjust the discharge pressure but, regardless of the method chosen, adjustments must also be made in the expansion process to avoid penalties in terms of the evaporator superheating and cooling capacity (Montagner and Melo, 2010). It should be mentioned that light commercial refrigeration systems are space- and cost-constrained, thus requiring a simple and cost-effective design. The simplest and widely-applied design is the standard capillary tube-based system (Figure 1A), which demonstrates good adaptability regarding variations in the ambient temperature in spite of its simplicity (Cecchinato *et al*., 2007; Agrawal and Bhattacharyya, 2008). The capillary tube offers a certain degree of control of the discharge pressure when the gas cooler outlet temperature diverges from the system design point, but penalties in terms of the evaporator superheating usually occur (Montagner and Melo, 2011). Conversely, a thermostatic expansion valve design (Figure 1B), which has been shown to work satisfactorily in other applications (Aprea and Mastrullo, 2002), is able to perfectly maintain the evaporator superheating, but with penalties related to the discharge pressure (Montagner and Melo, 2011).

In order to obtain a simultaneous control of the discharge pressure and the evaporator superheating, modifications at the cycle level are thus required. Bearing in mind the requirement of simplicity, one option is to install a medium-pressure buffer, as shown in Figure 1C. In the double expansion process, the first valve controls the discharge pressure while the second regulates the refrigerant flow to the low pressure-side of the system and thus the evaporator superheating. After the high pressure valve the refrigerant exists as saturated liquid since the medium-pressure receiver is thermal insulated and there is no removal of flash gas. Such a design, with minor differences, was theoretically analyzed by Cason at al. (2003) and more recently experimentally evaluated by Cabello et al. (2011).

The main focus of this paper is to compare three alternatives of cycle designs – capillary tube, thermostatic valve and medium-pressure receiver – by experimentally evaluating the system performance at variable ambient temperature. To this end, an extensive analysis of the effects of the cycle design and ambient temperature on the system thermodynamic parameters was carried out. The experiments were performed on a purpose-built testing facility, which allows modifications at both the component and cycle level.

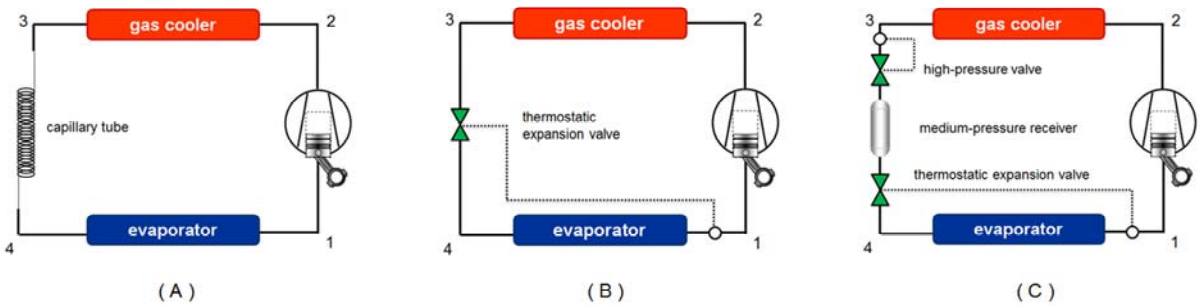

Figure 1. Cycle designs: (A) capillary tube, (B) thermostatic expansion valve and (C) medium-pressure receiver

2. EXPERIMENTAL SETUP AND METHODOLOGY

2.1 Experimental setup

The experimental apparatus is essentially a refrigeration system with a nominal cooling capacity of 600W that can be easily rearranged to study different cycle architectures. Figure 2 shows a schematic representation of the medium-pressure receiver cycle design. The system is driven by a 1.75cm^3 reciprocating-type compressor. The evaporator and the gas cooler are counter-flow tube-in-tube heat exchangers with an inner tube of 6.35mm OD and an outer tube of 14.7mm OD. The heat transfer rates, both in the evaporator and in the gas cooler, are controlled by two secondary heat transfer loops, the former using a solution of ethylene glycol as the heat source and the latter using water as the heat sink. The expansion device is comprised of a needle valve mounted in series with a capillary tube (Figure 3). The needle valve provides a method to easily change the expansion restriction whereas the capillary tube provides stability to the expansion process. Temperature and pressure measurements were made at several points along the circuit. The CO_2 and secondary loop flow rates were measured by Coriolis and turbine-type flow meters, respectively. The compressor power, voltage and current were measured by specific transducers. The transducers, their measurement range and their uncertainties are listed in Table 1. COP and cooling capacity measurements were carried out with an expanded uncertainty of approximately 4%, considering a coverage factor of 2 and a confidence level of 95%. The thermodynamic properties were calculated using the Engineering Equation Solver (EES) software (Klein, 2011).

Each of the cycle designs described in Figure 1 were studied by assembling the “expansion zone” (Figure 2) as follows: (i) a needle valve in series with a capillary tube to mimic a fixed-area restriction-type expansion device (capillary tube) – the valve opening was set and kept fixed during the tests; (ii) a needle valve in series with a capillary tube to mimic a thermostatic expansion valve – the valve opening was varied in order to maintain a certain evaporator superheating; and (iii) an in-line 0.5 cm^3 medium pressure-receiver with two needle valves – the valve openings were varied in order to keep the discharge pressure and the evaporator superheating at certain values.

2.2 Methodology

The experiments aim at investigate the thermodynamic performance of the different cycle designs when exposed to variable ambient temperatures. The ambient temperature has a direct impact on the system performance since it affects the CO_2 temperature at the outlet of the gas cooler. To study such an effect tests were carried out with three cycle designs – capillary tube, thermostatic valve and medium-pressure receiver – by varying the temperature of the cooling water at the inlet of the gas cooler.

The capillary tube baseline design was firstly optimized in order to provide a gas cooler outlet temperature of 38°C, which corresponds to an approach of 6^{o}C. The needle valve opening and the refrigerant charge were both adjusted in order to provide the highest COP under this condition by establishing a low evaporator superheating (3.5°C) and a discharge pressure close to the optimum value (101 bar) (Montagner and Melo, 2011). A baseline cooling capacity of 640 W was thus obtained, with an evaporation temperature of -7°C. Thereafter, the tests consisted of varying the temperature of the cooling water at the gas cooler inlet in order to vary the CO_2 temperature at the gas cooler outlet. The water flow rate was kept constant and equal to the 38^{o}C baseline test. The cooling capacity was maintained at the baseline value of 640 W, by varying the brine flow rate while keeping its temperature at the inlet of the evaporator constant and equal to the baseline value of 12^{o}C.

In the capillary tube design, valve A (Figure 2) was kept fully open while valve B was kept at the opening required by the 38^{o}C baseline condition. In the thermostatic valve design, valve A was also kept fully open while the opening of valve B was varied in order to maintain an evaporator superheating of 3.5^{o}C. Finally, in the medium-pressure receiver design the openings of both valves were varied, valve A to maintain the evaporator superheating at 3.5^{o}C and valve B to maintain the discharge pressure close to the optimum value.

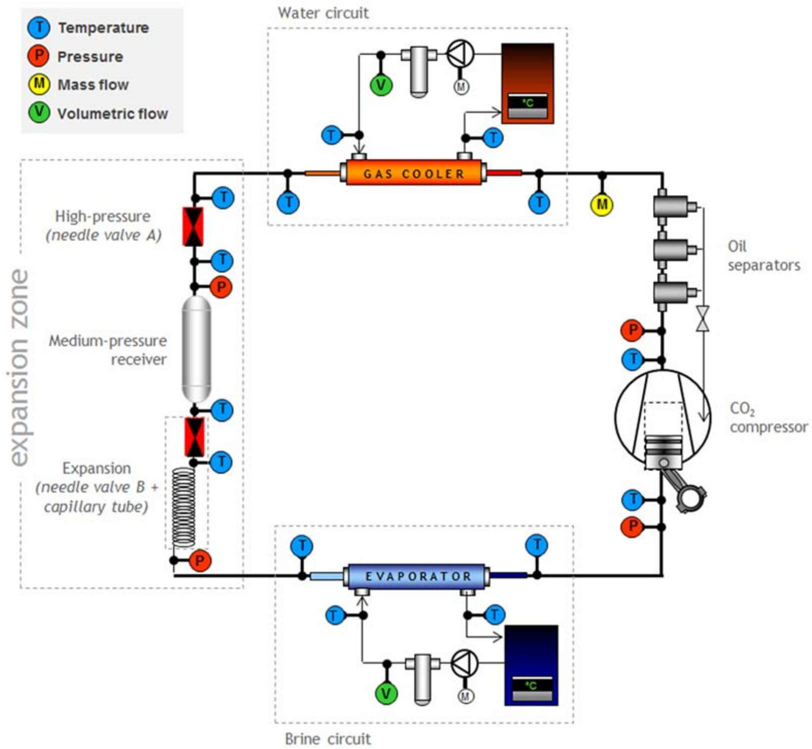

Figure 2. Schematic representation of the medium-pressure receiver cycle design

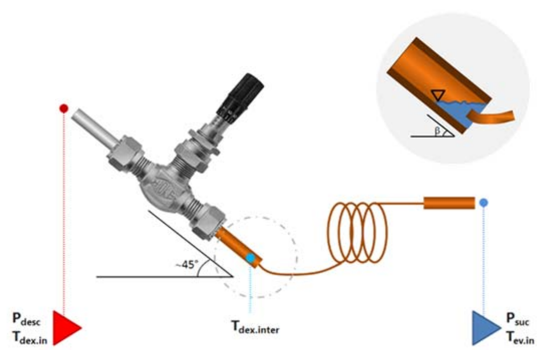

Figure 3. Representation of the expansion device

Table 1. Measurement devices

Parameter	Measurement device	Range	Uncertainty
Temperature (°C)	T-type	-50 – 150	±0.2
Pressure (bar)	Strain gage	0 – 100	±0.3
Pressure (bar)	Strain gage	0 – 200	±0.5
CO_2 flow rate (kg/h)	Coriolis	10 – 50	±0.01
Brine flow rate (m^3/h)	Turbine	0.036 – 0.180	$\pm 2.10^{-5}$
Water flow rate (m^3/h)	Turbine	0.036 – 0.144	$\pm 3.10^{-5}$
Compressor power (W)	Wattmeter	0 – 1000	±3
Compressor current (A)	Amperemeter	0 – 5	±0.01
Compressor voltage (V)	Voltmeter	0 – 220	±0.6
Refrigerant charge (g)	Electronic scale	0 – 5000	±0.1

3. EXPERIMENTAL RESULTS AND DISCUSSION

Figure 4 illustrates the behavior of the evaporator superheating as a function of the CO_2 temperature at the outlet of the gas cooler, for the three cycle designs studied. It is worth noting that the thermostatic valve and the medium-pressure receiver designs are able to maintain the evaporator superheating close to the set-point of 3.5°C, while the capillary tube design allows a superheating variation of approximately 7°C in the temperature range studied. Figure 5 shows that the refrigerant mass flow rate steadily increases in the thermostatic and medium-pressure receiver designs according to the CO_2 temperature at the outlet of the gas cooler. This is so because in both designs the opening of valve B is used as a means of controlling the evaporator superheating. In contrast, the refrigerant mass flow rate in the capillary tube design is kept almost constant, leading to the evaporator starvation at high temperatures at the outlet of the gas cooler, as shown in Figure 4. It should also be noted that the refrigerant mass flow rate in the thermostatic valve design varies from 15.5kg/h to 25.5kg/h while in the medium-pressure receiver design this variation is slightly smaller, mainly due to the differences in the discharge pressure, as explained below.

In a conventional subcritical vapor compression refrigeration cycle the expansion device is almost unable to regulate the system pressures which are set by the evaporation and condensation temperatures, these in turn being governed by the heat exchanger design and by the internal and external air temperatures, respectively. The refrigerant mass flow rate is thus established from a balance between the system pressures and the compressor capacity. Unlike the regular cycle, condensation does not take place in the high-pressure side heat exchanger (gas cooler) of the transcritical cycle. In such a case the discharge pressure is a weak function of the heat exchanger temperature and a strong function of the mass flow rate through the expansion device. Depending on the logic behind the expansion process the discharge pressure may vary in such a way as to affect positively or negatively the system performance. It is precisely this feature of the CO_2-based systems which permits the use of active control techniques to match the actual and optimum discharge pressures.

Operation close to the optimum pressure guarantees that the system is running with the best compromise between the specific refrigerant effect and the specific compression work. Figure 6 shows the behavior of the discharge pressure versus the CO_2 temperature at the outlet of the gas cooler, for

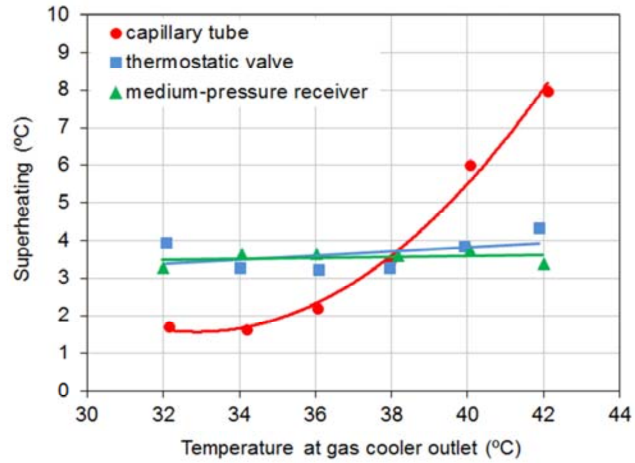

Figure 4. Superheating vs. temperature at gas cooler outlet for three cycle designs

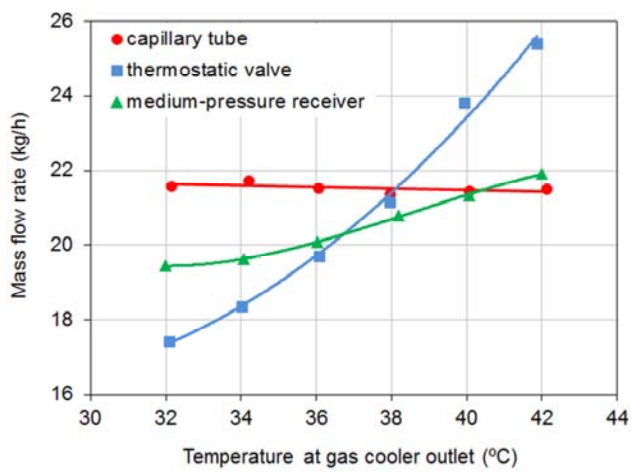

Figure 5. CO_2 mass flow rate vs. temperature at gas cooler outlet for three cycle designs

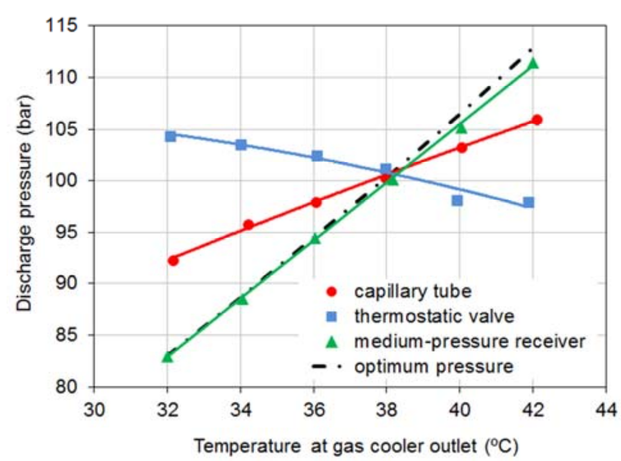

Figure 6. Discharge pressure vs. temperature at gas cooler outlet for three cycle designs

the three cycle designs considered in this work. At temperatures higher than 38°C the thermostatic valve design works with discharge pressures lower than the optimum values due to the progressive opening of valve B to avoid the growth of the evaporator superheating. The opposite behavior is observed at temperatures lower than 38°C, when valve B closes to avoid the reduction of the evaporator superheating. A dissimilar behavior is observed with the capillary tube design, where the discharge pressure follows a trend similar to that of the optimum pressure according to the CO_2 temperature at the outlet of the gas cooler. In such a case, the discharge pressure reflects the variation of the specific volume of the supercritical fluid contained in the high-pressure side of the system. Finally, the medium-pressure receiver design almost reproduces the optimum pressure behavior (Figure 6) while keeping the superheating at the desired set-point (Figure 4), through the appropriate opening of valves A and B.

Table 2 compares the actual and optimum discharge pressures for each of the cycle designs investigated in this study, according to the CO_2 temperature at the outlet of the gas cooler. It can be seen that there is a maximum deviation of 25.3% and 1.4% with the thermostatic valve and medium-pressure receiver designs, respectively. The discharge pressure affects the vapor quality at the evaporator inlet (Figure 7) and the compression power (Figure 8), both of which affect the system performance.

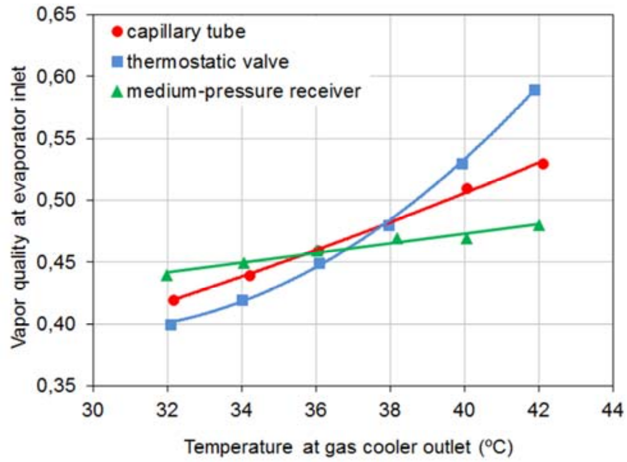

Figure 7. Vapor quality at the evaporator inlet vs. temperature at gas cooler outlet for three cycle designs

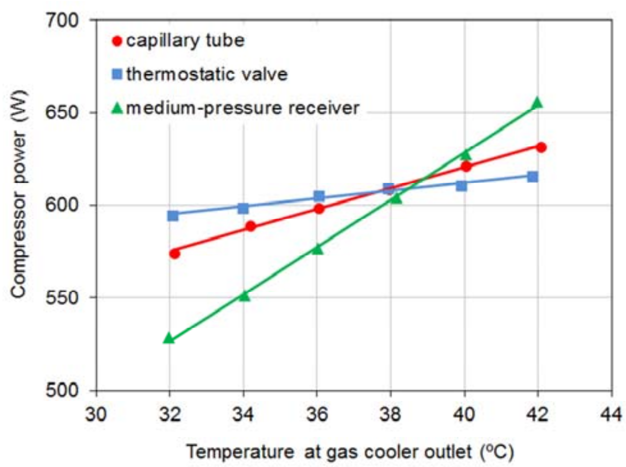

Figure 8. Compressor power vs. temperature at gas cooler outlet for three cycle designs

Figure 7 shows the vapor quality at the evaporator inlet according to the CO_2 temperature at the gas cooler outlet. It can be observed that, independently of the cycle design, the vapor quality increases with an increase in the gas cooler outlet temperature. At gas cooler outlet temperatures higher than 38°C, the medium-pressure receiver design is less penalized in this respect due to the better match between the actual and optimum discharge pressures, as indicated in Figure 6. In contrast, the thermostatic valve design is more penalized in terms of specific refrigerant effect due to the opposite trends of the discharge and optimum pressures. At gas cooler outlet temperatures lower than the baseline value of 38°C this tendency is inverted, with the vapor quality at the evaporator inlet being less affected by the thermostatic valve design, although slight deviations do occur from one design to another. In this region the

Table 2. Experimental discharge pressure and estimated optimum discharge pressure vs. gas cooler outlet temperature

Gas cooler outlet temperature	Optimum pressure	Capillary tube		Thermostatic valve		Medium-pressure receiver	
°C	bar	bar	%	bar	%	bar	%
32	83.0	92.3	*10.7*	104.3	*25.3*	82.9	*-0.1*
34	88.8	95.8	*7.3*	103.5	*16.6*	88.5	*-0.3*
36	94.6	98.0	*3.5*	102.4	*8.1*	94.4	*-0.1*
38	100.9	100.3	*0.1*	101.2	*1.0*	100.1	*-0.7*
40	106.6	103.2	*-2.8*	98.1	*-7.3*	105.1	*-1.4*
42	112.8	105.9	*-5.6*	97.9	*-12.4*	111.4	*-1.2*

compression power plays a significant role, as indicated in Figure 8. It is worth noting that the compressor power decreases with a lowering of the gas cooler outlet temperature and that this trend is also affected by the cycle design, especially at temperatures lower than the baseline value of 38°C. The almost constant compression power associated with the thermostatic valve design results from the offsetting characteristics of the mass flow rate (Figure 5) and the discharge pressure (Figure 6). In contrast, the compressor power of the medium-pressure receiver design reflects the multiplicative effect of these two parameters.

As previously noted, the optimum discharge pressure offers the best compromise between the specific refrigerant effect and the specific compression work and its outcome can be quantified by the coefficient of performance (COP). Figure 9 depicts the COP behavior for the three cycle designs as a function of the CO_2 temperature at the outlet of the gas cooler. It can be seen that the COP values of the capillary tube and the thermostatic valve designs are almost the same, suggesting that the better superheating of the thermostatic valve design is offset by the better discharge pressure of the capillary tube design. It should be mentioned that at 42°C the thermostatic valve design was unable to provide the baseline cooling capacity of 640W, as shown in the inset of Figure 9. The COP of this particular design, at this temperature, is thus penalized. Figure 9 also shows that the medium-pressure receiver design offers better COP values at low gas cooler outlet temperatures than the other designs. It is also interesting to note that in this temperature range the high values of vapor quality at the evaporator inlet (Figure 7) are overcompensated by the low compressor power (Figure 8), for the medium-pressure receiver design. Nevertheless, no differences in the system COP are observed with the three cycle designs studied in this paper, at high gas cooler outlet temperatures.

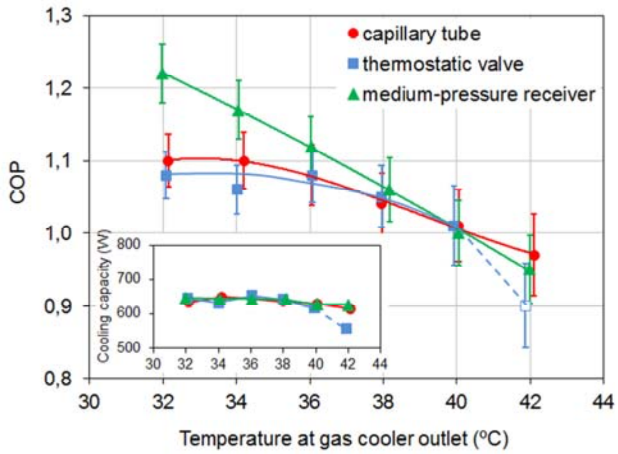

Figure 9. COP and thermal load vs. temperature at gas cooler outlet for three cycle designs

4. CONCLUSIONS

This paper reports part of the results of a research effort carried out to experimentally evaluate the thermodynamic behavior of CO_2 cycles for light commercial refrigeration. Three cycle designs were analyzed in order to compare their influence on the system thermodynamic parameters under variable ambient temperature conditions. It was demonstrated that the thermostatic valve design provides an effective control of the evaporator superheating but leads to a penalty in terms of discharge pressure in the temperature range studied. In contrast, the capillary tube design provides discharge pressures close to the optimum values, but without a proper control of the superheating. The interaction between the opposite behaviors of superheating and discharge pressure results in almost the same performance for both systems. In turn, the medium-pressure receiver design was able to simultaneously control the superheating and the discharge pressure over the temperature range under analysis. This design provides a COP improvement of up to 10% with respect to the other two designs for gas cooler outlet temperatures lower than the baseline value. At temperatures above the baseline value no performance differences were observed among the three designs. In summary, it seems that the capillary tube design is a good alternative for light commercial refrigeration systems, especially at high ambient temperatures, since it offers a certain degree of control of the discharge pressure, in spite of its simplicity and reduced cost.

5. ACKNOWLEDGEMENTS

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