A STUDY ON THE PERFORMANCE CHARACTERISTICS OF CARBON DIOXIDE REFRIGERATING SYSTEMS WITH MULTI-SPEED TWO-STAGE COMPRESSION

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Abstract The light-commercial refrigeration sector considers carbon dioxide (CO_2) as a promising natural candidate to replace the conventional synthetic refrigerants, but performance improvements at the cycle level are still required. The aim of this work is to investigate the effect of the compressor speed on the performance characteristics of CO_2-based refrigerating systems. To this end a variable speed two-stage rotary compressor was installed in an existing experimental apparatus and an energy optimization exercise was carried out. The optimum refrigerant charge and restriction for each compressor speed was experimentally found and compared. It was noted that Coefficient of Performance (COP) for the whole range of compressor speed reaches a maximum at a single combination of charge and restriction. This suggests that a capillary tube may be used for metering the refrigerant mass flow rate in variable capacity carbon dioxide refrigerating systems, even knowing that at low speeds the evaporator heat exchanges will be negatively impacted by the high evaporation pressures.

Keywords carbon dioxide, variable capacity compressor, two-stage compression

1. INTRODUCTION

One of the most relevant aspects to be accounted for during the design phase of a new product is the corresponding lifecycle environmental impact. An increase in consumer awareness, as well as restrictions imposed by regulatory agencies, has pushed most refrigerator manufacturers towards new and innovative products in order to remain competitive in the market. A suitable way of reducing the environmental impact of a refrigerator consists in replacing the conventional synthetic refrigerants by natural substances (Montagner and Melo, 2012). Carbon dioxide is one of the most favorable natural refrigerants especially because it is non-toxic and non-flammable, which makes it particularly attractive for light-commercial refrigeration applications. Other advantages of CO_2 are the high volumic refrigerating effect, low pressure drop and low Global Warming Potential (GWP). But what prevents carbon dioxide from being widely adopted over its synthetic counterparts is its relatively low refrigerating efficiency, especially at high ambient temperatures. To overcome this disadvantage, modifications have to be introduced at the cycle level. Internal heat exchangers (Robinson and Groll, 1998) and two-stage compression (Christen et. al. (2006), for example, are good alternatives of energy saving.

Variable speed compressor research begun in the 1980's and since then this type of compressor has been used in many applications. The compressor capacity is controlled by a frequency inverter and the power consumption is thus decreased due to the reduction of the on/off cycles. In spite of its widespread use there is only a few works reported in the open literature focused on this type of compressor for CO_2-based light-commercial refrigeration systems. The goal of this article is to close this gap by offering a comprehensive experimental investigation on the effect of the compressor speed on the performance characteristics of a CO_2-based refrigerating loop running with a variable capacity two-stage rotary compressor.

2. EXPERIMENTAL SETUP AND METHODOLOGY

2.1 Experimental setup

The experimental apparatus is essentially a refrigeration loop with a cooling capacity of approximately 1250 W. The cycle architecture can be easily rearranged, thus allowing the study of different cycle configurations. Figure 1 shows a schematic of the circuit used in this work. It is driven by a 1.28 cm^3/ 0.84 cm^3 two-stage variable capacity rolling piston compressor with intercooling between stages. Three oil separators are installed in series in the discharge line in order to periodically return the oil to the compressor crankcase. The gas cooler and the evaporator are concentric tube counter-flow heat exchangers, the former cooled by a water loop and the latter heated by an ethylene glycol secondary loop. The expansion device is comprised of a needle valve in series with a 0.82 mm I.D., 600 mm long capillary tube. Table 1 lists some of the measured parameters with their respective transducers, ranges and uncertainties.

The discharge pressure which has a strong effect on the system COP (Kim et. al., 2004; Cabello et. al., 2008) was controlled by the amount of refrigerant contained in the loop. To this end a cell charge – to add and extract refrigerant – was purposely designed, constructed and integrated to the refrigeration loop. It is worth mentioning that an internal heat exchanger will only be added to the analysis in the next phase of this work.

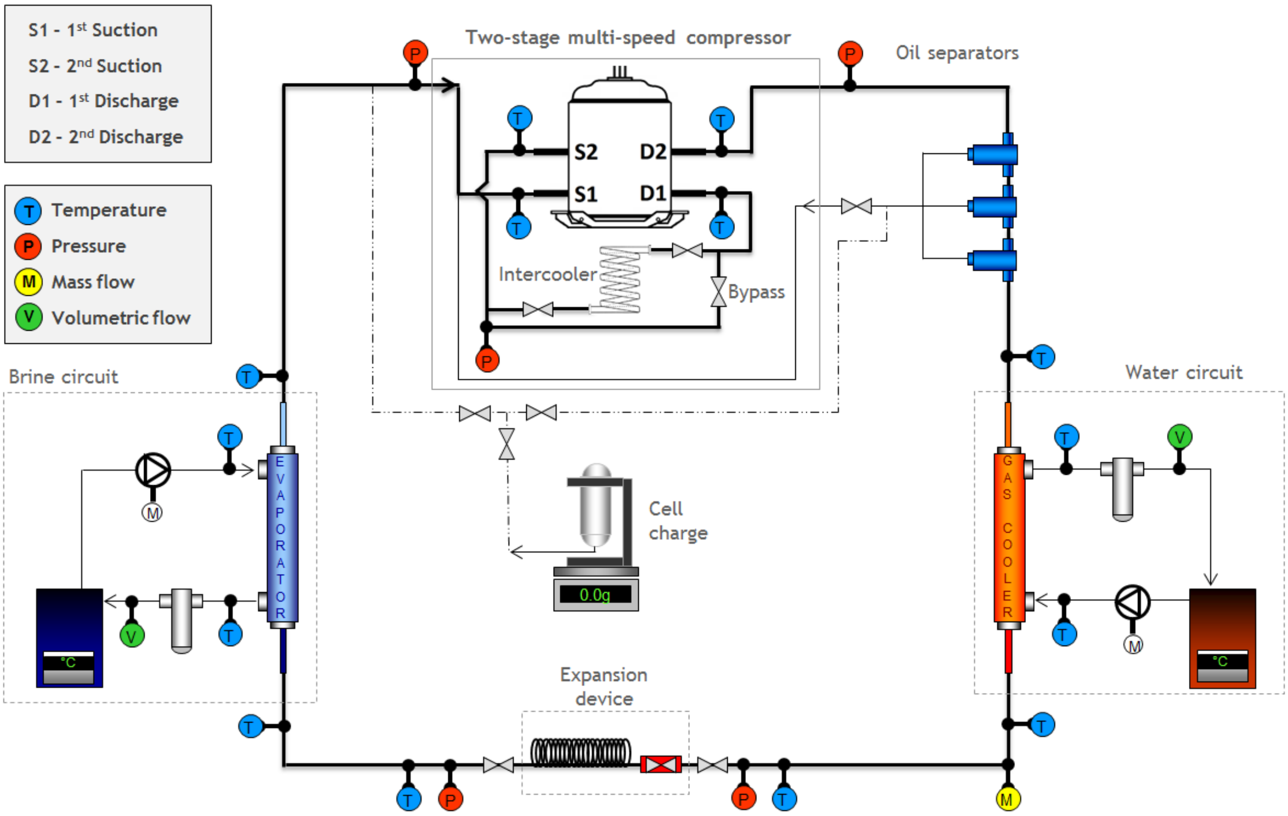

Figure 1. Schematic of the experimental apparatus

2.2 Methodology

Since the refrigerant charge and the corresponding discharge pressure play a significant role on the thermodynamic characteristics of CO_2-based refrigerating systems, the optimum charge was firstly found for each combination of compressor speed and expansion restriction exhibited in Table 2. The expansion restriction was varied through the opening of the metering valve. The valve opening was related to the remaining number of turns required for total closure. The fully-open position corresponds to the 7.5 opening, while the fully-closed position corresponds to the zero opening.

During this procedure the operating conditions were kept at the values exhibited in Table 3. It is worth noting that at 2400 rpm those conditions were only attained with an expansion restriction of 4.5 turns. In total, the experimental database is comprised of a reasonable amount of data – 98 data points – which will be examined in this work.

Table 1. Measurement parameters and devices

Parameter	Transducer	Range	Uncertainty
Temperature (°C)	T-type thermocouple	-50 – 150	± 0.2
Low pressure (bar)	Strain gage	0 – 100	± 0.3
High pressure (bar)	Strain gage	0 – 200	± 0.5
CO_2 mass flow rate (kg/h)	Coriolis	0.1 – 45	± 0.01
Brine volumetric flow rate (m^3/h)	Turbine	0.036 – 0.018	± 2.10^{-5}
Water volumetric flow rate (m^3/h)	Turbine	0.036 – 0.0144	± 3.10^{-5}
Compressor power (W)	Wattmeter	0 – 1000	± 3
Refrigerant charge (g)	Electronic scale	0 – 5000	± 0.1

Table 2. Range of the experimental data

Compressor speed (rpm)	Expansion device restriction (turns)	Charge (g)
2400	4.5	500 – 800 (16 steps of 20g)
3600	4.5, 6.0, 7.5	
4500	4.5, 6.0, 7.5	

Table 3. Operating conditions

System component	Condition
Intercooler	32°C air inlet, 70% effectiveness
Gas cooler	33°C water inlet, 4°C approach
Evaporator	12°C brine inlet, 5°C brine outlet

3. EXPERIMENTAL RESULTS AND DISCUSSION

Figure 2 displays the effect of the refrigerant charge on the discharge pressure for each compressor speed and valve opening. It can be seen that the discharge pressure varies almost linearly with the refrigerant charge with the exception of the data gathered at 45000rpm/4.5 turns, which follows a second-order polynomial behavior.

Figure 3 illustrates the relationship between the refrigerant charge, valve opening and suction pressure. It can be noted that the suction pressure also increases with the refrigerant charge and so does the intermediate pressure. In overall, the compression ratio for both compression stages is almost unaffected by the refrigerant charge. It should be mentioned that the refrigerant charge was limited to 720g during the tests with a valve opening of 7.5 to prevent the liquid slug over to the compressor suction.

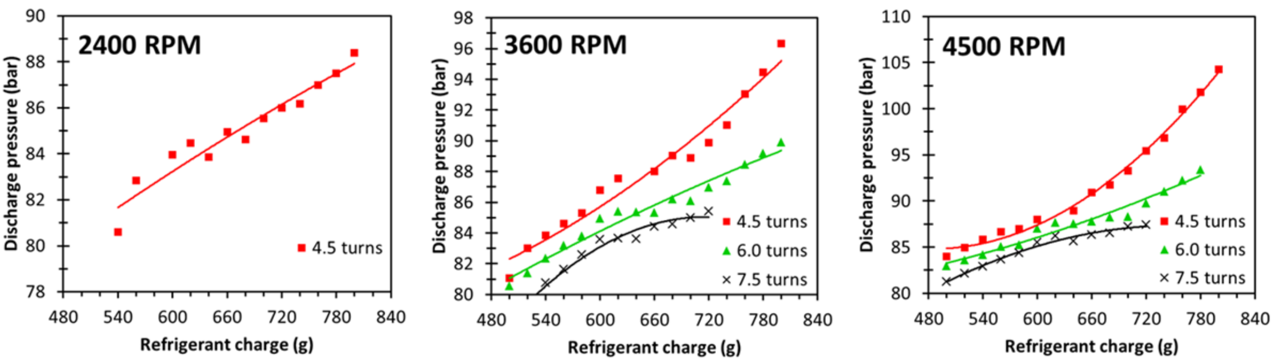

Figure 2. Discharge pressure vs. refrigerant charge for different compressor speeds and valve openings

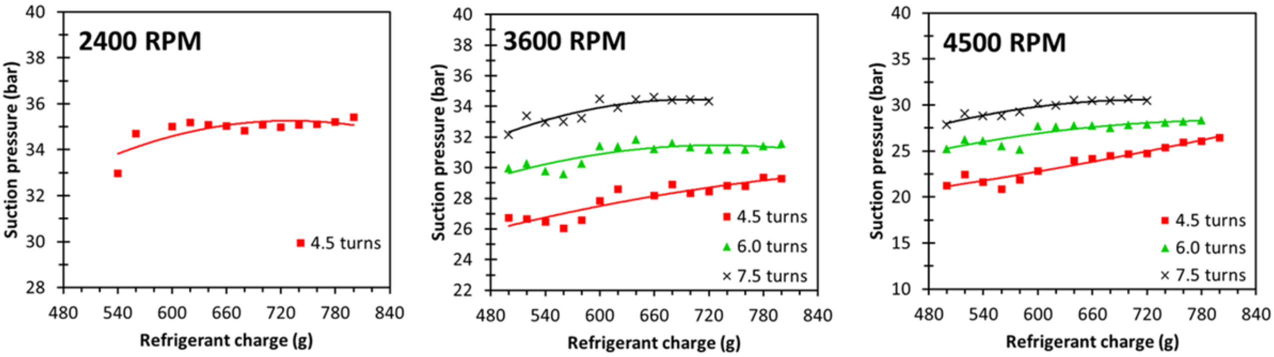

Figure 3. Suction pressure vs. refrigerant charge for different compressor speeds and valve openings

Figure 4 shows that the evaporator superheating decreases almost linearly with the refrigerant charge until a limit amount of charge, from where the behavior turns to be asymptotic. The superheating range is higher at higher compressor speeds, meaning that its effect on the specific volume and mass flow rate is also higher at higher speeds, as shown in Figure 5.

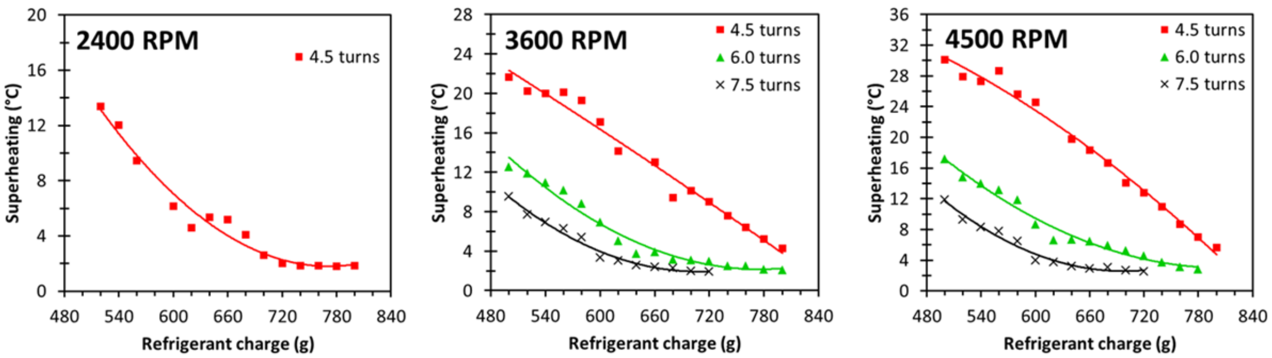

Figure 4. Superheating vs. refrigerant charge for different compressor speeds and valve openings

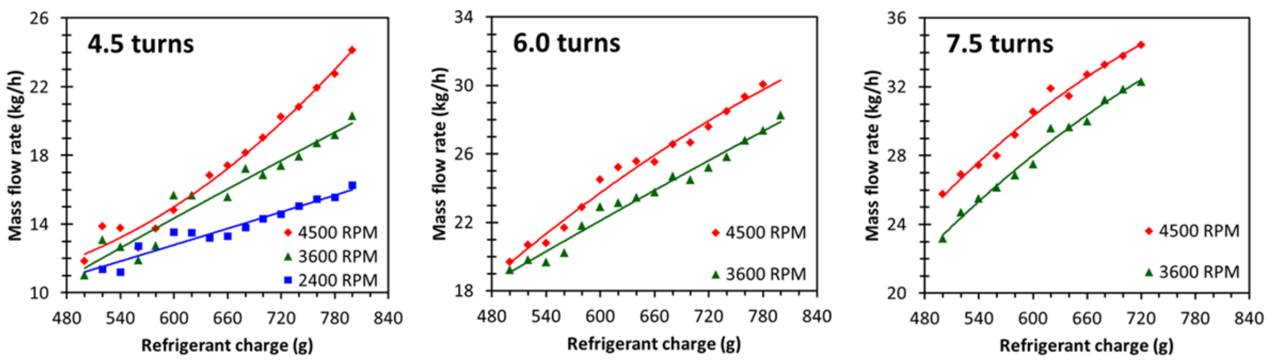

Figure 5. Mass flow rate vs. refrigerant charge for different compressor speeds and valve openings

Figure 6 illustrates the cooling capacity as a function of the refrigerant charge, compressor speed and expansion restriction. It is worth noting that the evaporator latent heat exchanges are increased while the sensible heat exchanges are decreased as more refrigerant is added to the system. This effect, when combined with the higher mass flow rates at higher refrigerant charges increases the cooling capacity until a maximum, when the cooling capacity starts to drop due to the growth of the evaporation temperature. It can also be noted that the influence of the refrigerant charge slowly decreases at higher charges, because after a certain point, when the evaporator is completely filled with liquid, the cooling capacity is mostly affected by the mass flow rate.

Figure 7 shows that the compressor power varies strongly with the refrigerant charge. As previously mentioned the compression ratio is almost constant for each pair of refrigerant charge and expansion restriction and so are the volumetric and isentropic efficiencies. Thus, the compressor power is mostly affected by the specific volume and this is deeply affected by the refrigerant charge, especially at lower suction pressures (higher restrictions and/or higher compressor speeds). The rate of drop of the specific volume with the refrigerant charge, for example, is two times greater at 4500 RPM and 4.5 turns than at 4500 RPM and 7.5 turns.

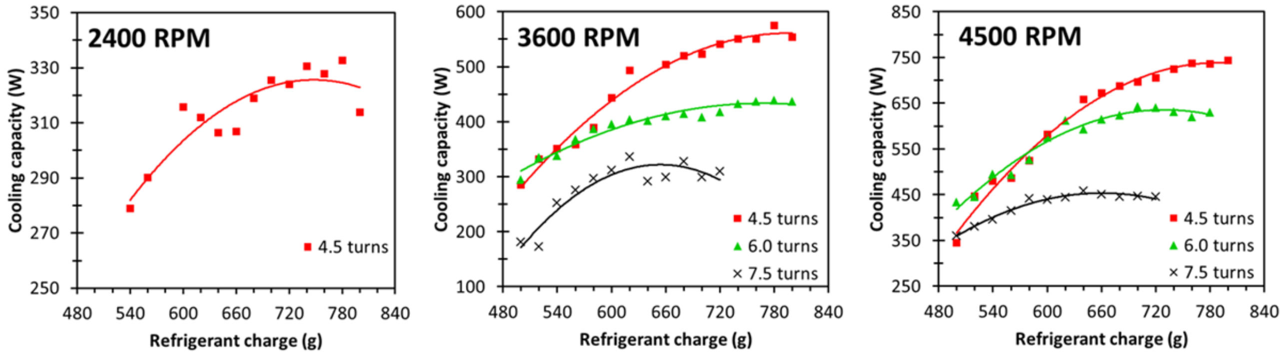

Figure 6. Cooling capacity vs. refrigerant charge for different compressor speeds and valve openings

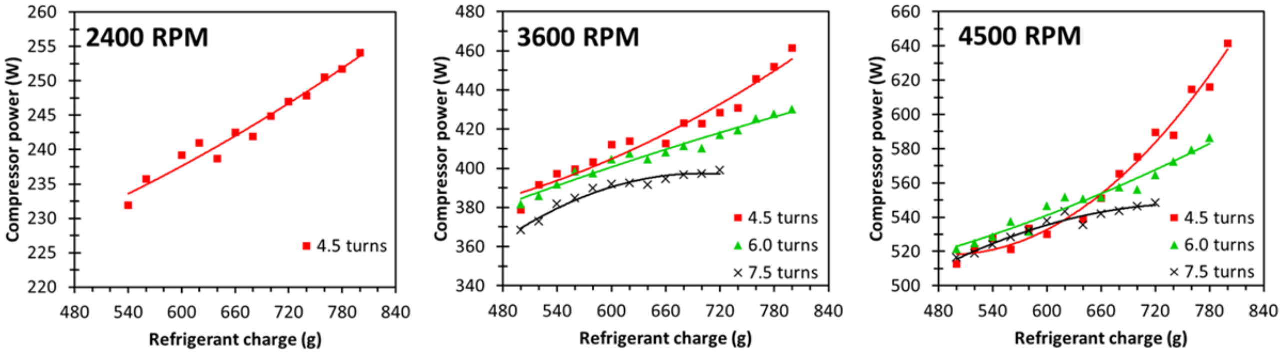

Figure 7. Compressor power vs. refrigerant charge for different compressor speeds and valve openings

Figure 8 shows the COP behavior for each compressor speed and expansion restriction as a function of the refrigerant charge. As expected, for each pair of compressor speed and expansion restriction there is always an optimum refrigerant charge. It is worth noting that a maximum COP is reached at the same valve opening of 4.5 turns and refrigerant charge of 740g, independently of the compressor speed. This behavior reflects the cooling capacity and compressor power behaviors shown in Figures 6 and 7, respectively. Table 4 shows the optimum refrigerant charge in terms of cooling capacity and COP. It can be seen that the system reaches the optimum COP with a refrigerant charge lower than that corresponding to the point of optimum cooling capacity. This is also explained by the cooling capacity and power behaviors, illustrated in Figures 6 and 7, respectively.

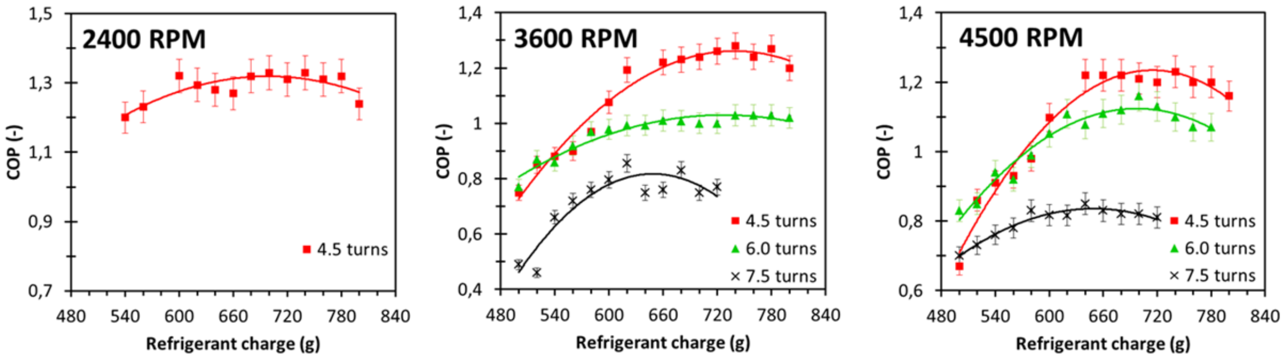

Figure 8. COP vs. refrigerant charge for different compressor speeds and valve openings

Table 4. Refrigerant charges for optimal cooling capacity and COP at 4.5 turns expansion valve opening

	Refrigerant charge (g) at 4.5 turns expansion valve opening		
	2400 RPM	**3600 RPM**	**4500 RPM**
Optimal cooling capacity	780 g	780g	800 g
Optimal COP	740 g	740 g	740 g

Table 5 shows the most relevant thermodynamic parameters at the optimum operating point of each compressor speed. It can be seen that the compressor power, cooling capacity and mass flow rate all increase significantly with the compressor speed, while the system COP is only slightly reduced. It can also be noted that the cooling capacity increases more than the mass flow rate due to the drop of the suction pressure with increasing compressor speed (Chen and Gu, 2004). Although it might be possible to obtain an even higher COP for each compressor speed with a distinct combination of restriction and charge, a single pair that can be used with any compressor speed offers a good compromise between performance and flexibility.

Table 5. Thermodynamic parameters at the optimum operating point

PARAMETER	2400 RPM	3600 RPM	4500 RPM
COP (-)	1.33	1.28 (-3.8%)	1.23 (-7.5%)
Compressor power (W)	248	431 (+73.8%)	588 (+137.1%)
Cooling capacity (W)	354	551 (+55.6%)	724 (+104.5%)
Suction pressure (bar)	35.2	29.2 (-17.4%)	26.1 (-25.8%)
Intermediate pressure (bar)	59.7	54.6 (-8.5%)	48.4 (-18.9%)
Discharge pressure (bar)	86.2	91.0 (+5.6%)	96.8 (+12.3%)
Discharge temperature (^oC)	52.5	67.0 (+14.5^oC)	79.1 (+26.6^oC)
Evaporator inlet temperature (^oC)	0.7	-6.1 (-6.8^oC)	-10.1 (-10.8^oC)
Evaporator superheating (^oC)	1.9	7.6 (+5.7^oC)	10.9 (+9.0^oC)
Mass flow rate (kg/h)	15.0	17.9 (+19.3%)	20.8 (+38.5%)
Vapor quality at evaporator inlet (-)	0.52	0.49 (-5.8%)	0.48 (-7.7%)

4. CONCLUSIONS

The effect of the compressor speed on the thermodynamic performance of CO_2-based refrigerating systems was studied herein. To this end the optimum refrigerant charge for each pair of compressor speed and expansion restriction was experimentally found. It was shown that the discharge pressure varies almost linearly with the refrigerant charge, giving room for a performance optimization process based only on the amount of refrigerant contained in the system.

It was found that a single pair of expansion restriction (4.5 turns) and refrigerant charge (740g) provides the maximum COP, independently of the compressor speed. These finding, which is in line with those reported by Montagner (2013), is of utmost importance for variable capacity CO_2-based refrigeration systems, since it indicates that a properly sized fixed restriction expansion device, such a capillary tube, can be used to meter the refrigerant flow. The drawback is a penalty in cooling capacity since the suction pressure increases at lower speeds.

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REFERENCES

1. Cabello, J.B., Sánchez, D., Llopis, R., Torrela, E., 2007. "Experimental evaluation of the energy efficiency of a CO_2 refrigerating plant working in transcritical conditions". Applied Thermal Engineering, Vol. 28, pp. 1596-1604.

2. Chen, Y., Gu, J., 2005. "Non-adiabatic capillary tube flow of carbon dioxide in a novel refrigeration cycle". Applied Thermal Engineering, Vol. 25, pp. 1670-1683.

3. Christen, T., Hubacher, B., Bertsch, S.S., Groll, E.A, 2006. "Experimental performance of prototype carbon dioxide compressors". In Proceedings of the International Compressor Enginnering Conference at Purdue, West Lafayette, Indiana-IL, USA.

4. Kim, M., Pettersen, J., Bullard, C.W., 2004. "Fundamental process and system design issues in CO_2 vapor compression systems". Progress in Energy and Combustion Science, Vol. 30, pp. 119-174.

5. Montagner, G.P., 2013. A Study on the Application of Transcritical CO_2 Cycles in Light-Commercial Refrigerating Systems. Ph.D. thesis, Federal University of Santa Catarina, Florianópolis, SC, Brazil. (in portuguese)

6. Montagner, G.P., Melo, C., 2012. "Study on cycle designs for light commercial CO_2 refrigeration systems". In Proceedings of the 14^{th} Brazilian Congress of Thermal Sciences and Engineering. Rio de Janeiro, RJ, Brazil.

7. Robinson, D.M., Groll, E. A., 1998. "Efficiencies of transcritical CO_2 cycles with and without an expansion turbine". International Journal of Refrigeration, Vol. 21, No. 7, pp. 577-589.