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DESIGN AND CONSTRUCTION OF AN AUTOMATED HOT-GAS CYCLE CALORIMETER FOR CARBON DIOXIDE COMPRESSORS

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Abstract. *The present work proposes a novel design of an automated hot-gas cycle calorimeter for CO_2 compressors. The proposed refrigeration cycle, with three pressure levels and two valves coupled with stepper motors, provided an automatic control of the discharge and suction pressures and made the calorimeter much more agile and stable for a wide range of operating conditions. For this end, a specific control system was studied, and a 1.28cm^3 variable-speed rolling-piston compressor was tested. A set of 72 experimental points could be measured within only 2 weeks. The compressor speed was varied from 35 to 65Hz, the suction pressure from 20 to 50bar and the discharge pressure from 75 to 105bar. The ambient and suction line temperatures, on the other hand, were held at 32ºC. In addition, an empirical and a physical-based model were analyzed in order to generate the compressor efficiency curves. The results indicated that the physical-based model was the one that better represented the behavior of the compressor.*

Keywords: *calorimeter, variable capacity compressor, carbon dioxide, refrigeration, hot-gas*

1. INTRODUCTION

The growing concern with the environment and the pressure imposed by regulatory agencies have pushed the refrigerators manufacturers to use natural refrigerants as an alternative to current synthetic fluids. Due to its safety aspects and environmentally friendly characteristics, carbon dioxide stands out as a promising candidate (Montagner, 2013). However, to be considered a viable substitute for the existing refrigerants, the CO_2 system should be as or more effective than the current systems in a wide range of ambient temperatures. Therefore, changes should be introduced in the cycle to ensure a better use of the CO_2 characteristics, including the improvement of the compressor.

In order to improve the compressor it is necessary to understand its thermodynamic behavior in several operating conditions. The most common way to do it is to test the compressor in a calorimeter. However, depending on the calorimeter design the tests could be very time-consuming. In this context, Dirlea *et al.* (1996), Joffily (2007) and Duarte *et al.* (2014) proposed more agile calorimeters, with a low refrigerant charge and a reduced thermal inertia. Silveira (2012) proposed an automated hot-cycle calorimeter for microcompressors that counted with two valves coupled to stepper motors to control the suction and discharge pressures, and an intermediate reservoir which was used to accommodate the excess of refrigerant.

The present work proposes a novel hot-cycle calorimeter design based on that of Silveira (2012), but adapted to work with CO_2 compressors. To validate the experimental apparatus a variable-speed compressor was tested and its efficiency curves were obtained with an empirical and a physical-based model.

2. EXPERIMENTAL WORK

2.1 Experimental Apparatus

The experimental apparatus was designed in a modular fashion in order to comprise not only the refrigeration circuit but also the DAQ system, the electric board and the test section (see Fig. 1). To measure the refrigerant mass flow rate a Coriolis meter was used. The discharge and suction pressures were controlled by two valves with internally-coupled stepper motors (see Fig. 3). A stainless steel intermediate reservoir, with an inner volume of 2.25 liters, was installed between the suction and discharge lines in order to accommodate the excess of refrigerant and to allow the calorimeter to operate in a wide range of conditions with only one refrigerant charge. All temperature measurements were performed with T-type thermocouples. A schematic view of the refrigeration loop is shown in Fig. 2.

The suction line temperature was controlled with a PID-driven flexible heater attached to the tube combined with a forced-convection tube-and-fin heat exchanger. Due to the high heat transfer rate through the compressor shell, an auxiliary refrigeration system was designed to control the test section temperature. The test section air velocity, in turn, was controlled with the aid of a variable-speed radial fan.

Figure 1. Experimental apparatus

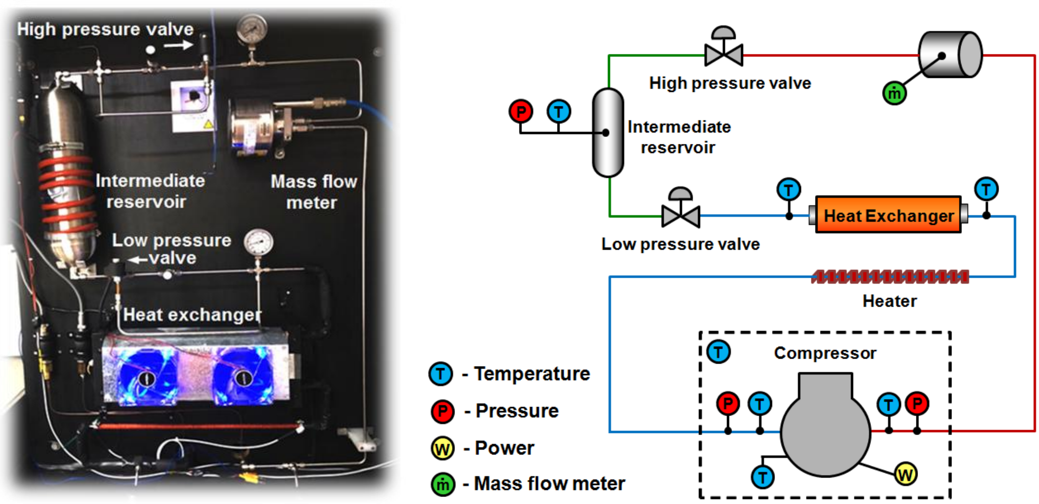

Figure 2. Schematic view of the calorimeter refrigeration loop

Figure 3. Automatic valve used to control the discharge and suction pressures

2.2 System identification and control strategy

To control the pressures two PI (proportional-integral) controllers were developed. A mathematical model for the process (see Fig. 4) was obtained using a method based on the step response (Åstrom and Hägglund, 1995). The following first-order transfer function with transport delay was adopted:

$$G(s) = \frac{k_p}{\tau s + 1} e^{-\theta s} \tag{1}$$

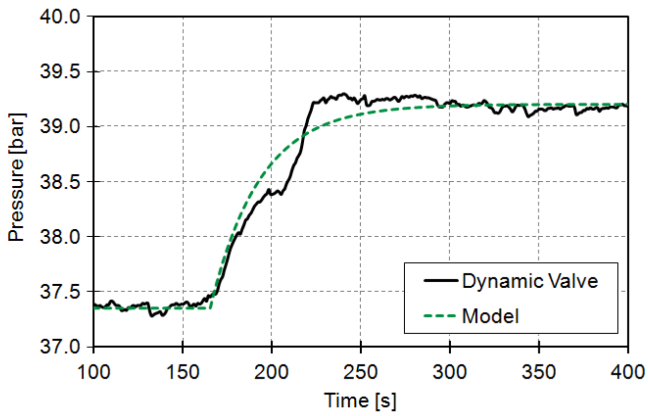

Figure 4. System identification

where K_p is the static gain, τ is the time constant, s is the complex variable and θ is the transport delay. The controller parameters were tuned using the root-locus method (Ogata, 2002), in order to guarantee a balance between performance and robustness. Equation (2) shows the PI controller transfer function.

$$C(s) = k_e + \frac{k_i}{s} \tag{2}$$

where k_e is the proportional gain and k_i is the integral gain.

Figure 5 illustrates the closed-loop diagram for the process and Tab. 1 shows the controller parameters for each process variable.

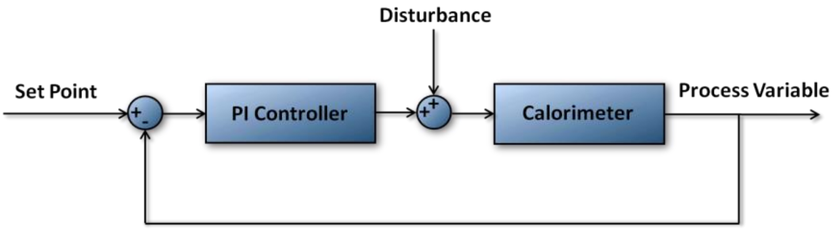

Figure 5. Closed loop control system

Table 1. Controller parameters

Process Variable	k_e	k_i
Suction pressure	4.75	0.1187
Discharge pressure	-4.70	-0.1215

2.3 Methodology

Prior to the test beginning the intermediate and discharge lines were heated up to 70ºC by flexible heaters in order to guarantee that there was only superheated vapor throughout the cycle. After that, the test section air temperature and velocity were controlled and held at the standardized values of 32ºC and 3m/s, respectively. Finally, the compressor was turned on and the suction and discharge pressures were set to the desired values.

In order to validate the calorimeter 72 tests were carried out with a 1.28cm^3 rolling-piston variable-speed compressor. Three different speeds were evaluated (35, 50 and 65Hz) and 24 tests were performed for each one. Six evaporating temperatures were evaluated from -15 to 10ºC, while four discharge pressures (from 75 to 105bar) were considered. The full test matrix is shown in Fig. 6a. In all tests the suction temperature was kept as close as possible to the ambient temperature, as recommended by the standard ANSI/ASHRAE 23 (2005).

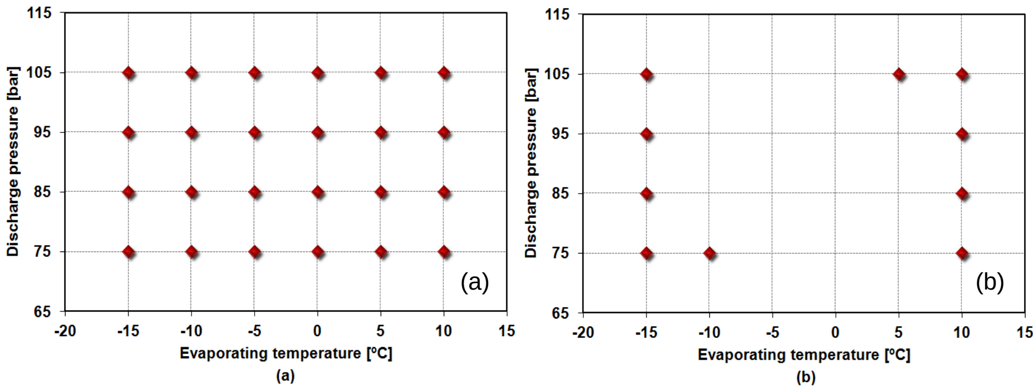

Figure 6. Full test matrix (a) and reduced test matrix (b), for each speed

3. EMPIRICAL AND PHYSICAL-BASED COMPRESSOR MODELS

The compressor experimental data were fitted through the least-square method using two different models: the AHRI 571 (2012) standard empirical model and the physical-based model proposed by Li (2012). It is worth noting that only 10 points (out of the 24 points obtained for each speed) were used to obtain the curves coefficients (see Fig. 6b).

3.1 Empirical model

This model is purely empirical and consists of fitting the AHRI 571 (2012) bi-cubic polynomials as a function of the discharge and suction pressures:

$$X = c_1 + c_2 P_{suc} + c_3 P_{disc} + c_4 P_{suc}^2 + c_5 P_{suc} P_{disc} + c_6 P_{disc}^2 + c_7 P_{suc}^3 + c_8 P_{suc} P_{disc}^2 + c_9 P_{suc}^2 P_{disc} + c_{10} P_{disc}^3 \quad (3)$$

where P_{disc} is the discharge pressure, P_{suc} is the suction pressure, and X can be either the cooling capacity or the power.

Table 2 shows the coeficients c_1 to c_{10} fitted through the least-square method for 50Hz.

Table 2. AHRI 571 (2012) polynomials coefficients

Coefficients	Power [W]	Cooling Capacity [W]
c_1	50.48084	64997.77
c_2	-1.507468	-3705.478
c_3	5.905405	3559.3
c_4	0.061052	351.5915
c_5	0.004969	-163.6382
c_6	-0.033631	-11.41492
c_7	-0.003472	-5.704322
c_8	-0.000454	-0.029883
c_9	0.002054	2.495667
c_{10}	0.000143	0.043304

3.2 Physical-based model

The Li (2012) model is considered to be physical because it is based on the fundamental equations of the compressor theory. It basically consists of an equation for the volumetric efficiency and another for the power consumption. Li (2012) adapted the model proposed by Jähnig *et al.* (2000) to calculate the volumetric efficiency by using two coefficients related the compressor clearance volume (b_1 and b_2) and one coefficient (dp) to take into account the effect of the suction pressure drop:

$$\eta_v = b_1 + b_2 \left[\frac{P_{disc}}{P_{suc}(1 - dp)} \right]^{1/k} \quad (4)$$

where k is the isentropic compression coefficient. The power consumption, in turn, is calculated as follows:

$$W = P_{suc}\dot{m}v_1 \frac{k}{k-1}\left[\left(\frac{P_{disc}}{P_{suc}}\right)^{\frac{k-1}{k}} - 1\right]\left(a_1 + \frac{a_2}{P_{suc}} + \frac{a_3}{P_{disc}}\right) + a_4 \tag{5}$$

where $\dot{m}$ is the mass flow rate and v_1 is the refrigerant specific volume at the compressor suction. Table 3 shows the coefficients values for 35, 50 and 65Hz.

Table 3. Physical-based model coefficients

Coefficient	35Hz	50Hz	65Hz
a_1	-1.493	-1.277	-1.353
a_2	1.109e7	6.949e5	3.371e5
a_3	-1.008e8	-5.37e6	-7.240e6
a_4	-99.21	-108.1	-212.3
b_1	-1.052	-1.085	-0.9444
b_2	-0.1483	-0.1358	-0.05222

4. RESULTS

4.1 Validation of the experimental apparatus

Figure 7 shows the suction line and the test section ambient temperature control at 32ºC, as defined by the ANSI/ASHRAE 23 (2005) standard. It can be seen that a steady state condition was achieved with less than 10 minutes. Figure 7 illustrates the suction and discharge pressures automatically controlled at one of the pre-defined conditions shown in Fig. 6. Again, a short settling time was observed and practically no overshoot was noted.

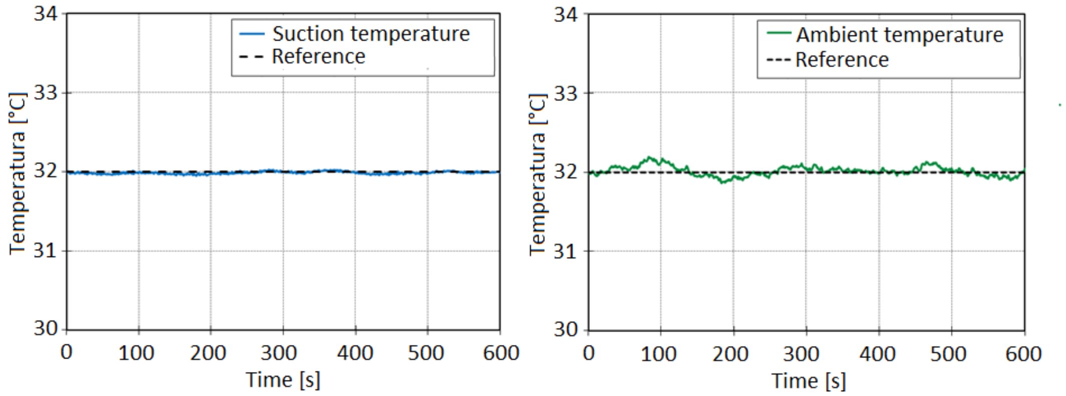

Figure 7. Suction line (left) and ambient temperature (right) control at 32ºC

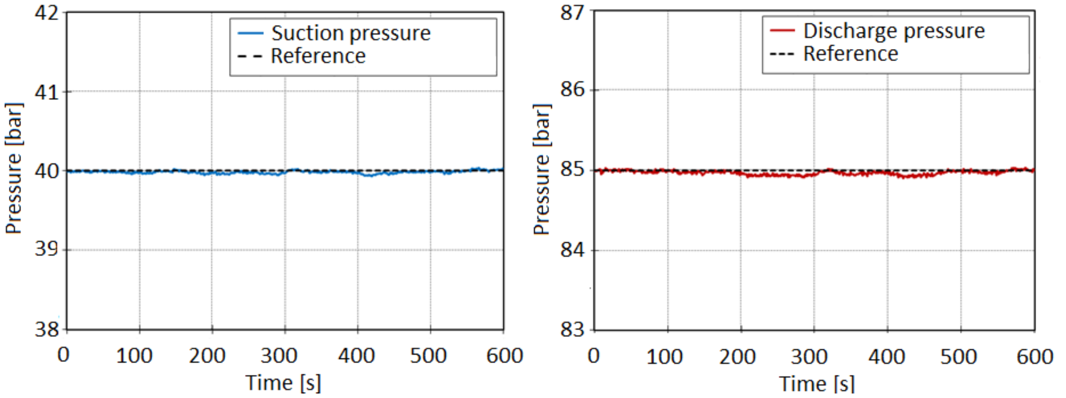

Figure 8. Suction (left) and discharge (right) pressures control

4.2 Compressor curves

Figure 9 shows the power and cooling capacity curve fitting results at 50Hz, considering the empirical model proposed by the standard AHRI 571 (2012). It was observed that this model was unable to capture the experimental trends, especially for the cooling capacity, where huge errors were verified (see Fig. 12). Additionally, it can be seen that both the interpolated and extrapolated cooling capacity values were not physically consistent.

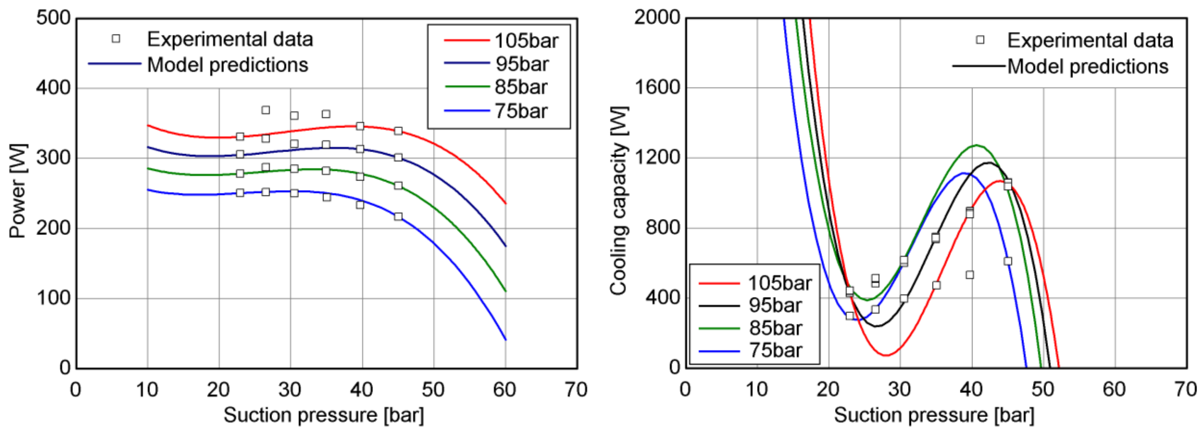

Figure 9. AHRI 571 (2012) empirical model curve fitting

The results obtained with the Li (2012) physical-based model at 50Hz are shown in Fig. 10. It can be seen a reasonable agreement between experimental and calculated values. The predicted values followed closely the experimental trends and coherent behaviors were noted for the extrapolated regions.

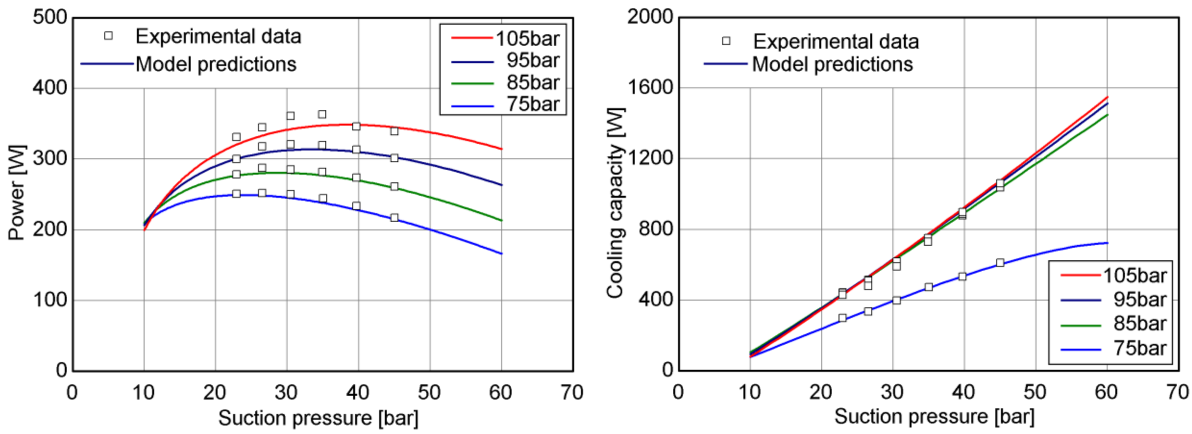

Figure 10. Li (2012) physical-based model curve fitting

Figures 11 and 12 show the relative error between the experimental and calculated values for the compressor power consumption and cooling capacity. As mentioned before, the errors observed using the AHRI 571 (2012) polynomials were low only for the compressor power but not for the cooling capacity. The Li (2012) model, on the other hand, provided much better results, with relative errors between ±5% for both power consumption and cooling capacity.

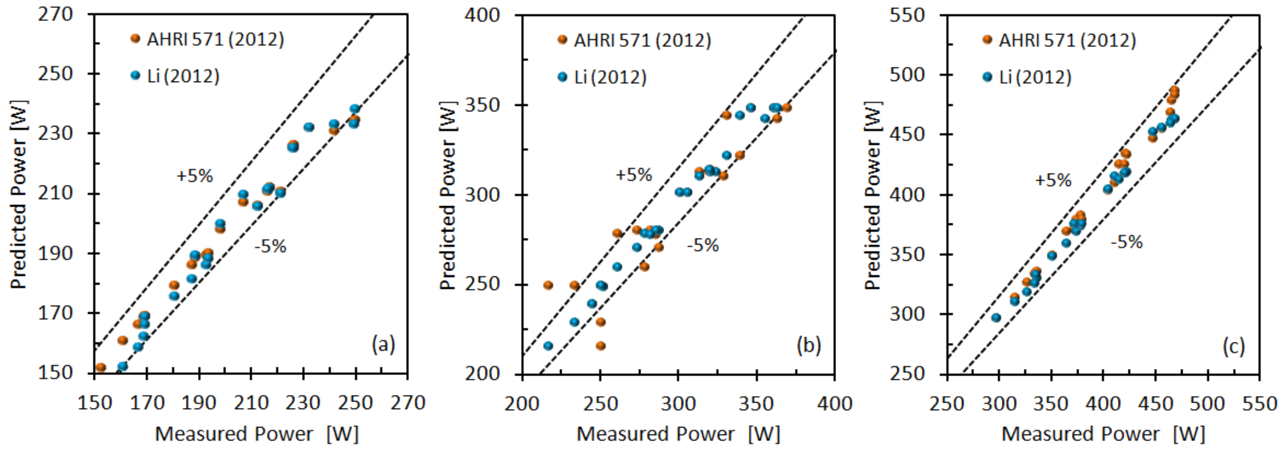

Figure 11. Power relative error: (a) 35Hz, (b) 50Hz and (c) 65Hz

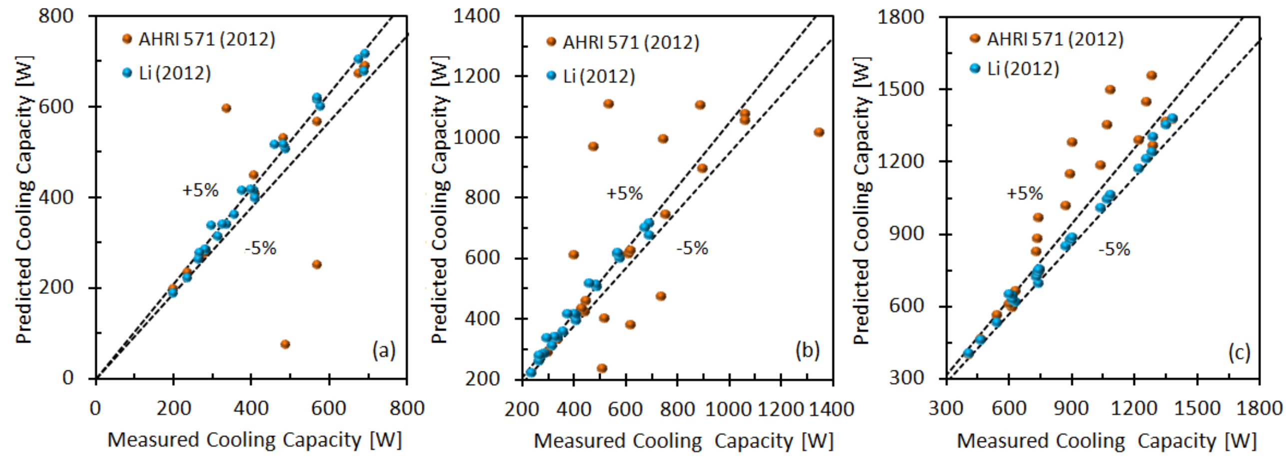

Figure 12. Cooling capacity relative error: (a) 35Hz, (b) 50Hz and (c) 65Hz

5. CONCLUSIONS

An innovative calorimeter for CO_2 compressors was presented herein. The proposed hot-cycle concept with an intermediate reservoir spares the refrigerant charge tuning. The test section temperature and air velocity control strategy showed good results and proved to be indispensable for the conditions where the heat transfer rate through the compressor shell was high. Moreover, the automatic control of the suction and discharge pressures, and the suction and ambient temperatures, made the calorimeter much more agile and robust for a wide range of operating conditions. A full compressor map can be obtained within a very short period of time, compared to the conventional designs.

Additionally, two mathematical models were adopted to present the experimental data: the AHRI 571 (2012) empirical polynomials and the Li (2012) physical-based equations. Huge deviations between the experimental and calculated cooling capacities were observed for the empirical model. Due to the lack of physical consistency, the empirical model was not able to predict coherently the compressor behavior out of the testing range. The Li (2012) model, in turn, which is based on the compressor theory fundamentals, predicted reasonably well the compressor experimental trends within an error band of ±5% for both power consumption and cooling capacity. Furthermore, the physical-based model was able to capture the compressor behavior (even in the extrapolated regions) using only 10 points, which can reduce even more the overall testing time.

6. ACKNOWLEDGEMENTS

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