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Ernane Silva
Federal University of Santa Catarina, Brazil, ernane@polo.ufsc.br

Willian M. Ferreira
Federal University of Santa Catarina, Brazil, willian.mf.floripa@gmail.com

Cesar J. Deschamps
Federal University of Santa Catarina, Brazil, deschamps@polo.ufsc.br

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A Calibration Procedure for Compressor Simulation Models using Evolutionary Algorithm

Ernane SILVA, Willian M. FERREIRA, Cesar J. DESCHAMPS*

POLO Research Laboratories for Emerging Technologies in Cooling and Thermophysics
Federal University of Santa Catarina
88040-900, Florianopolis, SC, Brazil
deschamps@polo.ufsc.br

* Corresponding Author

ABSTRACT

Comprehensive models are widely adopted to predict the performance of compressors due to their low computational cost and acceptable prediction capability. In general, the accuracy of such models depends strongly on the correct adjustment of some parameters that are of difficult to determine both analytically and experimentally. However, due to the nonlinearities of the compressor model, the tuning of such parameters affects many output variables and hence can be very challenging and time consuming. In this paper, we consider this procedure of adjustment of parameters as a multiple objective optimization problem that can be solved by using an elitist non-dominated sorting genetic algorithm (NSGA-II). The parameters of two simulation models are adjusted following this new procedure. In the first model the suction muffler was neglected and a mass-spring-damper system was adopted to predict the suction valve dynamics. The second model solves the suction valve dynamics by the finite element method and the flow in the suction muffler with the finite volume method. In both models the clearance volume and parameters associated with the suction valve were chosen to be adjusted while the deviations between predictions and measurements for the mass flow rate, indicated power, and suction valve dynamics were defined as the objective functions to be minimized.

1. INTRODUCTION

Numerical simulation has been widely adopted in the development of refrigeration compressors, providing detailed results that complement experimental data. Simulation models are less time consuming and less expensive than measurements and, with the current development of computers and numerical methods, can be applied to predict complex physical phenomena in compressors. For instance, valve dynamics and fluid flow through valves can be solved in a coupled manner to analyze the performance of compressor valves (Kim *et al*., 2006).

From the thermodynamic point of view, low-cost simulation models (Link and Deschamps, 2011) are based on the unsteady formulation of the conservation equations for mass and energy in the compression chamber with sub-models for different physical phenomena. The accuracy of such models depends strongly on the adjustment of some parameters that are difficult to determine both analytically and experimentally. A clear example of these parameters is the clearance volume in the compressor chamber, which varies significantly around its nominal value according to the assembling process and the manufacturing tolerances of the compressor components. However, due to the nonlinearities of the simulation model, the tuning of these parameters is very challenging and time consuming since it affects many output variables, such as mass flow rate, indicated power, and valve dynamics.

In this paper, we propose a procedure to adjust parameters of compressor simulation models following a multiple objective optimization problem that is solved via an elitist non-dominated sorting genetic algorithm, NSGA-II (Deb *et al*., 2002). To illustrate this, the procedure is applied to two versions of a simulation model. In the first model the suction muffler is neglected and a mass-spring-damper system is adopted to model the suction valve dynamics. The second model solves the suction valve dynamics via the finite element method and the flow in the suction muffler with the finite volume method. In both models the clearance volume and parameters associated with the suction valve were chosen to be tuned while the deviations between predictions and measurements for the mass flow rate, indicated power, and suction valve dynamics were defined as the objective functions to be minimized.

2. SIMULATION MODEL

The unsteady-state equations for the conservation of mass and energy are applied in conjunction with an equation of state to predict the thermodynamic state of the gas inside the compression chamber. The mass flow rates of gas through the suction and discharge valves are calculated with reference to the isentropic flow in a convergent nozzle and the introduction of the effective flow area coefficient, C_{ep}:

$$\dot{m} = C_{ep} A_o p_{up} \sqrt{\frac{2k}{RT_{up}(k-1)}} \sqrt{\Pi^{2/k} - \Pi^{(k+1)/k}} \tag{1}$$

where A_o is the area of the orifice, p_{up} is the upstream pressure, k is the ratio of specific heats, R is the specific gas constant, T_{up} is the absolute upstream temperature, and Π is the pressure ratio. The leakage of gas through the piston-cylinder gap is obtained by assuming a fully developed Couette-Poiseuille laminar flow.

Two models are available to solve the valve dynamics. The first one considers the valve as a mass-spring-damper system with a single degree of freedom moving perpendicularly to the valve seat. In the second model the valve is discretized in beam elements and solved via the finite element method (Silva *et al.*, 2012). The governing equation of both models can be written as:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F\} \tag{2}$$

where $[M]$, $[C]$, and $[K]$ stand for the mass, damping, and stiffness matrices associated to the valve, $\{x\}$, $\{\dot{x}\}$, and $\{\ddot{x}\}$ represent the vectors of displacement, velocity, and acceleration of the valve nodes, and $\{F\}$ is the vector of forces applied to the valve nodes. In the finite element method, the Rayleigh damping model (Cook *et al.*, 1989) is used:

$$[C] = \alpha[K] + \beta[M] \tag{3}$$

where α and β are the coefficients of the model. If the valve is characterized as a mass-spring-damper system, Equation (2) is reduced to a single scalar equation. The resulting force acting on the valve may result from oil stiction, F_S, preload imposed on the valve assembly, F_P, and gas pressure load, F_v. The magnitude of the latter is evaluated through the concept of effective force area coefficient, C_{ef}:

$$F_v = C_{ef} A_o \Delta p \tag{4}$$

where Δp is the pressure difference across the valve. In the present model, the magnitude of F_S and F_p were assumed constant and prescribed as input parameters of the simulation.

When applicable, the pulsating compressible flow through the suction and discharge mufflers is modeled via the conservation equations for mass, energy, and momentum following a one-dimensional formulation. Such equations are solved using the finite volume method in a coupled manner with the valve dynamics, and taking into account heat transfer and viscous effects. Details of this modeling approach can be obtained in Deschamps *et al.* (2002).

3. EXPERIMENTAL SETUP AND PROCEDURE

Measurements of different parameters were carried out in a compressor placed in a calorimeter, so as to allow tests for operating conditions found in refrigeration systems. As can be seen in the schematic representation of the calorimeter (Figure 1a), the experimental setup is composed of pipelines, control valves (CV1 and CV2), a mass flow meter (FM), heat exchangers (HX1 and HX2), a thermocouple (TC), and pressure transducers (PT1 and PT2). The calorimeter is designed in a way that refrigerant fluid can flow through the high and low pressure lines in the superheated state, as indicated in the pressure–enthalpy diagram of Figure 1b. In this experimental setup, refrigerant from the suction line (1) is admitted in the compressor C and compressed to the condensing pressure, p_C, of the system (2). Pressure transducers PT1 and PT2 monitor pressures in the suction and discharge lines, respectively. The temperature of the fluid is then reduced in the heat exchanger HX1, reaching condition (3), and is expanded

adiabatically through control valve CV1 to an intermediary pressure (4). Following the measurement of the mass flow rate at the flow meter FM, the temperature of the fluid is reduced again in the heat exchanger HX2, reaching condition (5), and expanded adiabatically to the evaporating pressure (6), p_E, through control valve CV2. Finally, with the use of an electrical heater EH and a thermocouple TC, the temperature in the suction line is controlled to reach condition (1) again.
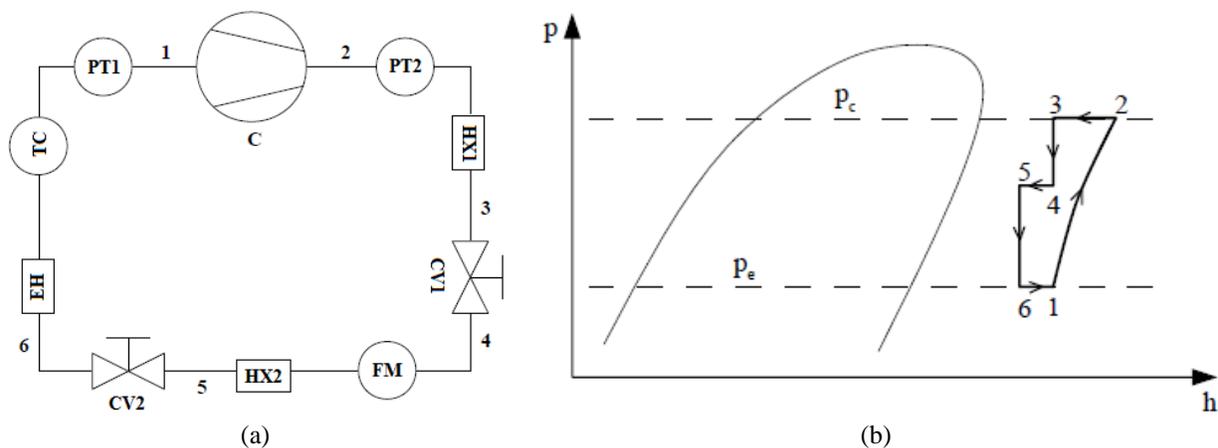

(a)　　　　　　　　　　　　　　　　　　　(b)

Figure 1: Hot gas test bench mechanical scheme (a) and pressure-enthalpy diagram of the thermodynamic cycle (b).

The adjustment of the amount of refrigerant in the system, the opening of the control valves, and the amount of thermal energy exchanged by the fluid in the heat exchangers and with the electrical heater, allow the control of the thermodynamic conditions in the suction (1) and discharge (2) lines and, consequently, the operating conditions to which the compressor is subjected. In addition to mass flow rate, the following parameters were also measured during compressor operation: crankshaft angle, pressures in the suction and compression chambers, temperatures in the suction chamber and cylinder wall, and the suction valve opening. The experimental procedure consisted in mounting the compressor in the system and submitting it to vacuum, allowing the desired amount of refrigerant to enter the system, running the compressor, and adjusting the control valves to reach the desired operating condition.

The measurements were conducted after the compressor had been running for at least 2 hours in order to achieve a fully cyclic operating condition. Then, measurements were carried out during a 30 minute interval. The compressor was evaluated under two operating conditions: A (T_e = - 28 ºC; T_c = 40 ºC) and B (T_e = 10 ºC; T_c = 70 ºC), where T_e and T_c stand for evaporating and condensing temperatures, respectively. The test of each condition was repeated 3 times and global parameters of the compression cycle, such as mass flow rate, indicated power, and local temperatures, were established in terms of a mean value. The parameters described as a function of the crankshaft angle, such as pressures in the suction and compression chamber and valve opening, were obtained from the cycle in which the global parameters were closer to the mean value.

4. CALIBRATION PROCEDURE

The accuracy of models developed to simulate refrigeration compressors depends strongly on the correct adjustment of some parameters that are difficult to determine both analytically and experimentally. Some of these parameters are adjusted in an indirect manner in order to reduce deviations between numerical and experimental results. In view of the complex physical interactions between different phenomena that take place inside the compressor and the consequent nonlinearity present in the models, this is a time-consuming task based on trial-and-error. Considering the importance and the difficulty of this activity, it is more convenient to consider the calibration procedure as an optimization problem. In this sense, the parameters to be adjusted become the design variables and the deviations between numerical and experimental results become the objective functions to be minimized. Therefore, the calibration procedure can be stated as:

$$\text{minimize: } f_i = f_i(x_1, x_2, \dots, x_j, \dots, x_{m-1}, x_m), \quad i = 1 \dots n \tag{5}$$

$$\text{subjected to: } x_j^l \leq x_j \leq x_j^u \tag{6}$$

where f_i represents one of the n objective functions of the problem and x_j represents one of the m design variables, with x_j^l and x_j^u as its lower and upper bounds, respectively.

Two different types of variables can be used as objective functions: global parameters of the compression cycle, which are scalar variables, such as the mass flow rate, and parameters that vary with the crankshaft angle, such as the valve opening. When parameters of the first type are considered, the deviations between the numerical and experimental results are calculated as:

$$f_i = |\varphi_{num} - \varphi_{exp}|$$
(7)

where φ_{num} and φ_{exp} represent the numerical result predicted by the compressor model and the experimental measurement, respectively. On the other hand, the deviations for parameters described as a function of the crankshaft angle, θ, are calculated as

$$f_i = \frac{\int_{0°}^{360°} |\varphi_{num} - \varphi_{exp}| d\theta}{\int_{0°}^{360°} |\varphi_{exp}| d\theta}$$
(8)

The genetic algorithm NSGA-II (Elitist Non-Dominated Sorting Genetic Algorithm) developed by Deb *et al*. (2012) was used to solve this optimization problem, since it is commonly used in problems with multiple objectives and real-valued variables. In this algorithm the ranking of the feasible solutions is based on the fitness and density of solutions in the objective space and the maintenance of the optimum solutions as the analysis evolves is based on the evaluation of the combined population of solutions formed by solutions belonging to the parental population, that is, the group of optimum solutions from the last iteration, and the offspring population, which represents the population of optimum solutions of the current iteration. In the current analysis, the adopted stopping criterion was based on the number of generations of the algorithm. The algorithm was implemented in the programming language C++ and parallelized using the application programing interface OpenMP to reduce the processing time of the calibration procedure.

5. RESULTS

In the present analysis, we used two different simulation models. In the first one, the presence of the suction muffler was neglected and a mass-spring-damper system was adopted to predict the suction valve dynamics. On the other hand, the second model solves the suction valve dynamics by the finite element method and the flow in the suction muffler with the finite volume method. In both models the discharge valve dynamics was solved considering a mass-spring-damper system and the presence of a discharge muffler. The preload force acting on the suction valve, F_p, was considered negligible and the value of the stiction force, F_s, was adjusted together with the clearance volume, V_c, and parameters of the suction valve.

The parameters of the suction valve adjusted in the first model were its mass and stiffness, without considering any damping effect. In the second model, only the parameters α and β associated with the modeling of the valve damping were considered, since the mass and stiffness matrices were obtained directly from the valve geometry. In both cases the absolute error of the mass flow rate, $\Delta\dot{m}$, indicated power, $\Delta\dot{W}$, and suction valve opening, Δd, in relation to experimental data, were defined as the objective functions to be minimized.

Both models were calibrated in the condition A ($T_e = -28$ ℃; $T_c = 40$ ℃) and validated in the condition B ($T_e = 10$ ℃; $T_c = 70$ ℃), which is quite different from the one used in the calibration. The optimization algorithm adopted the number of generations equals to 20, population size 80, mutation probability 0.375, and recombination probability 0.9. These parameters were defined based on a sensitivity analysis for the specific case considered.

The results of the calibration procedure for the first model at condition A are presented in terms of the objective functions in Figure 2. In this figure, the solutions that provide the minimum deviation in relation to the measurements for each of the objective functions considered are highlighted. Since each generation forms a Pareto frontier in the NSGA-II algorithm, the results obtained from the last iteration form a group of non-dominated solutions, that is, all solutions are equally optimal. Therefore, an additional criterion has be adopted in order to select a single solution. In

our study, we selected the solution that produced the smaller deviation for the suction valve dynamics, since its correct prediction affects the compressor efficiency and reliability. The deviations of the predictions in comparison to experimental measurements are presented in Table 1 for both operating conditions. It is possible to observe that the simulation model provided satisfactory results for indicated power and mass flow rate even in the operating condition far from the one used for calibration.

Figure 3 presents the comparison between numerical and experimental results for the suction valve displacement, s, as a function of the crankshaft angle, θ. It is worth mentioning that the adopted measurement technique did not allowed measurements of valve displacements greater than 2 mm. Based on the curves, it is possible to verify that the numerical model reproduced the valve displacement with good accuracy in the calibration condition A, with the opening and closing points matching the measurements. However, predictions of the valve opening and closing points do not match so closely the measurements for condition B, though the opening amplitudes seem to be well predicted. Such discrepancies may be associated with the consideration of a constant stiction force, independent of the operating condition selected. Therefore, the use a model for the stiction force based on physical reasoning could improve the predictions of valve dynamics in operating conditions far from that used in the model calibration.
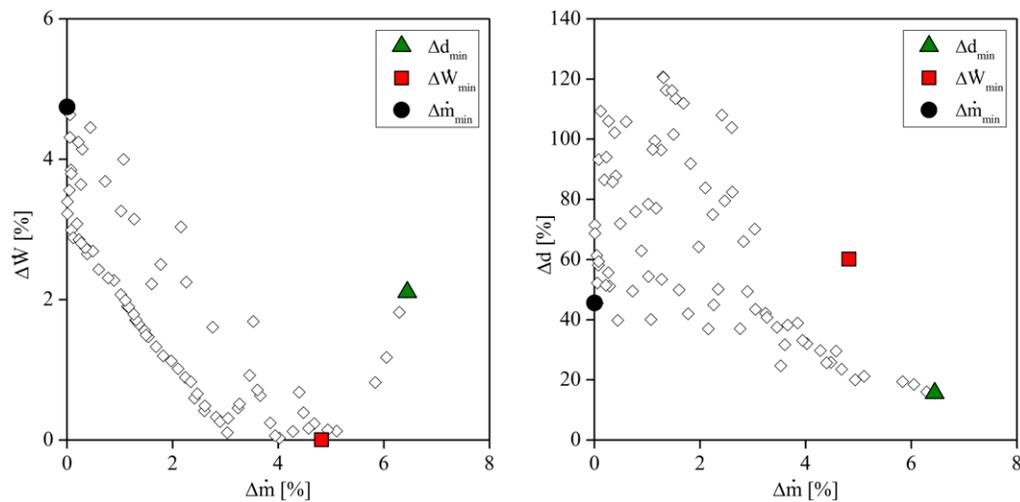

Figure 2: Optimal solutions in the solution domain.

Table 1: Deviations between predictions and measurements of
mass flow rate and indicated power.

Parameter		Operating conditions	
		A ($T_e = -28$ ºC; $T_c = 40$ ºC)	B ($T_e = 10$ ºC; $T_c = 70$ ºC)
1st model	$\Delta\dot{m}$ [%]	6.4	3.3
	$\Delta\dot{W}$ [%]	2.1	2.3
2nd model	$\Delta\dot{m}$ [%]	4.0	2.6
	$\Delta\dot{W}$ [%]	1.0	1.8

The results obtained in the calibration of the second model are presented in the solution domain in Figure 4, while the deviations for mass flow rate and indicated power are presented in Table 1. As can be seen, this model presents smaller deviations when compared to experimental measurements as one would be expected, since it provides more complete descriptions of the valve dynamics and fluid flow in the suction muffler. Again, larger deviations are observed in the predictions for the opening and closing of the valve (Figure 5) at the operating condition B ($T_e = 10$ ºC; $T_c = 70$ ºC). Lack of modeling for the stiction force and variation of the damping matrix $[C]$ with the operating conditions are possible sources of such deviations. In spite of that, the model predicts the valve displacement in reasonable agreement with the measurement.

In terms of computational time, 1600 simulations were run with both models to complete the calibration procedure. The calibration of the first model took about 4 hours and 15 minutes on a computer with 2.93 GHz Intel Core i7 processor, 6 GB DDR3 RAM and 64-bit Windows 7 operating system. The parallel processing allowed us to reduce the computational time by a factor of almost 5 in comparison to serial processing.
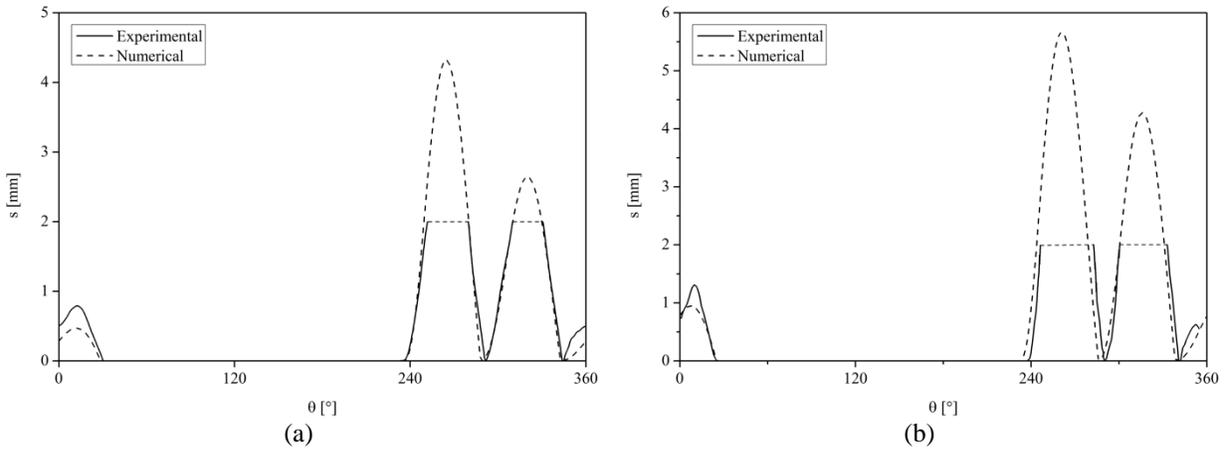
Figure 3: Comparison of numerical predictions and experimental measurements of valve displacement in the conditions: (a) ($T_e = -28$ °C; $T_c = 40$ °C); and (b) ($T_e = 10$ °C; $T_c = 70$ °C).
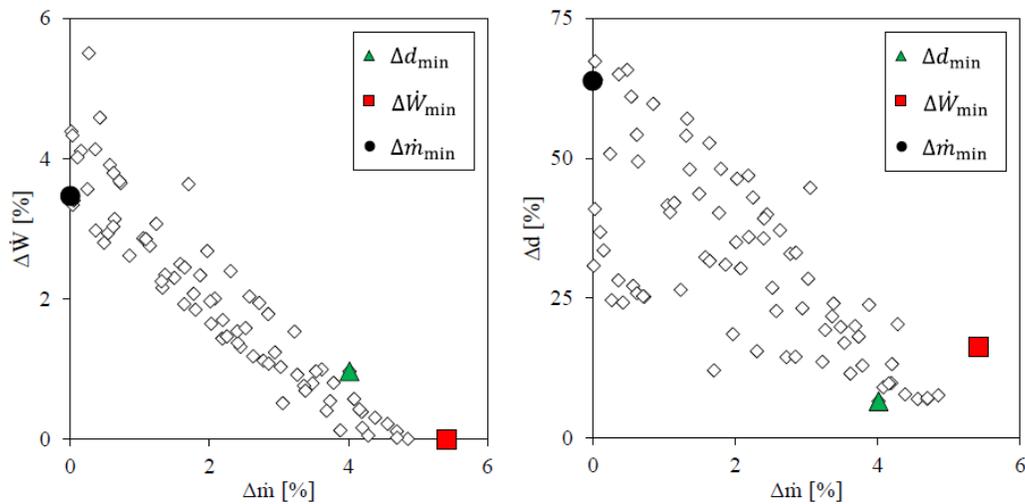
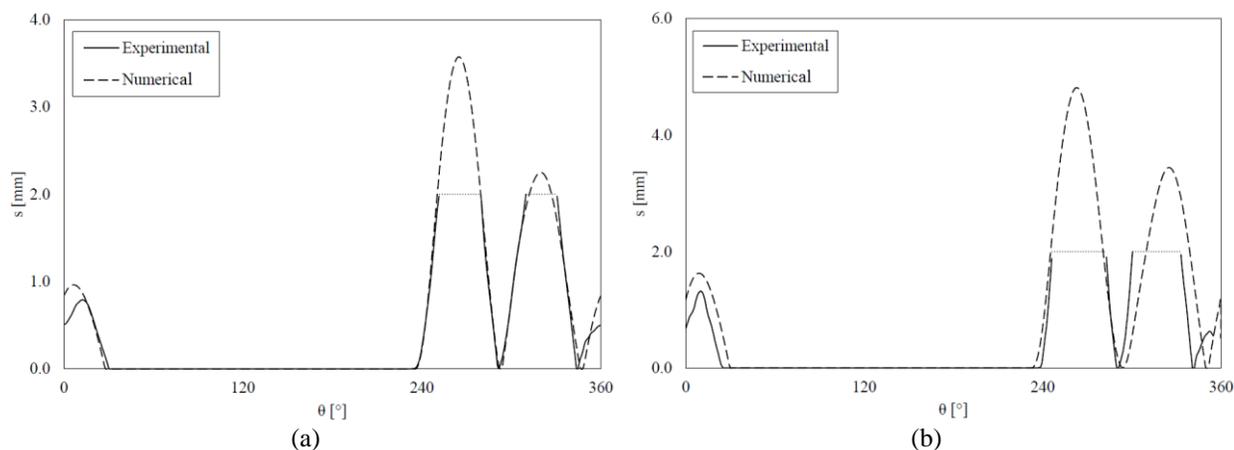**Figure 4:** Optimal solutions in the solution domain.

Figure 5: Comparison of numerical predictions and experimental measurements of valve displacement in the conditions: (a) (T_e = - 28 ºC; T_c = 40 ºC); and (b) (T_e = 10 ºC; T_c = 70 ºC).

6. CONCLUSIONS

A calibration procedure was developed to adjust input parameters of compressor simulation models that are difficult to obtain from measurements, such as clearance volume and stiction force. The procedure consists of a multi-objective optimization problem that is solved with an evolutionary algorithm. The parameters to be adjusted are considered the optimization variables while the deviations between predictions and measurements of output quantities, such as mass flow rate, indicated power, and valve displacement, were considered the objective functions. The procedure was applied to the calibration of two different simulation models. The two calibrated models were seen to predict mass flow rate, input power and valve displacement in close agreement with experimental data, even in operating conditions far from the one used for calibration. Nevertheless, some discrepancies were observed between predictions and measurements for the opening and closing crankshaft angles of the suction valve and attributed in part to lack of modeling of the stiction force. Overall, the calibration procedure showed to be effective in reducing the time required to calibrate simulation models, which give predictions in closer agreement with experimental data than models adjusted by the trial-and-error method.

NOMENCLATURE

A	area	(m^2)
C_{ef}	effective force area coefficient	(–)
C_{ep}	effective flow area coefficient	(–)
f	generic objective function	
F	force	(N)
k	ratio of specific heats	(–)
$\dot{m}$	mass flow rate	(kg/s)
p	pressure	(Pa)
R	specific gas constant	(J/kg K)
s	valve displacement	(mm)
T	temperature	(K or °C)
x	generic variable	
$\{F\}$	force vector	(N)
$\{x\}$	displacement vector	(m)
$\{\dot{x}\}$	velocity vector	(m/s)
$\{\ddot{x}\}$	acceleration vector	(m/s^2)
$[C]$	damping matrix	(N s/m)
$[K]$	stiffness matrix	(N/m)

[*M*]	mass matrix	(kg)
α	1st damping coefficient	(s)
β	2nd damping coefficient	(s^{-1})
Δd	deviation of valve dynamics	(%)
$\Delta \dot{m}$	deviation of mass flow rate	(%)
Δp	pressure difference	(Pa)
$\Delta \dot{W}$	deviation of indicated power	(%)
φ	generic value	
Π	pressure ratio	(–)
θ	cranckshaft angle	(°)

Subscript

c	condensing
e	evaporating
exp	experiment measurement
l	lower bound
num	numerical prediction
o	orifice
p	preload
s	stiction
u	upper bound
up	upstream
v	gas pressure load

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