ON THE CONTROL OF SUCTION VALVE OF A RECIPROCATING COMPRESSOR

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Abstract. *Due to the fact that nowadays people are living in smaller places, the concern related to the noise generated by domestic refrigerators acquired great relevance. Reciprocating compressors have a fundamental role in the overall noise produced by domestic refrigerators. In recent years several studies were intended to reduce the operational vibration of this kind of compressors and, consequently, mitigate the overall noise radiated by the refrigerator. One important vibration source of a reciprocating compressor is the pulsation of the refrigerant fluid, which flows within the suction chamber modulated by the movement of a passive suction valve. The objective of this work is to investigate different aperture profiles that could be prescribed to the suction valve in order to reduce the system's radiated noise and yet, preserve the compressors thermodynamic efficiency. This is achieved by conducting an optimization routine based on the genetic algorithm technique in order to find the best aperture profile for the valve. The aperture profiles suggested by the optimization analysis were evaluated numerically in a reciprocate compressor model, and showed that both considerable noise reduction and (thermodynamic) efficiency can be achieved. Moreover, The implementation of the optimized aperture profile in terms of active control are also discussed in this article.*

Keywords: *Suction valve, hermetic compressor, noise control*

1. Introduction

Noise generated by domestic appliances has become a relevant problem both to consumer and manufacturer. Therefore, efforts have been made to reduce the noise generated by these devices. In the case of domestic refrigerators, one of the major noise sources is the compressor. Usually in these applications reciprocating compressors hermetically closed are used. In compressors the pulsation flow of gas inside the suction muffler is one of the most significant noise sources. The pulsation characteristics is mainly driven by temporal behavior of the valve's aperture along one duty cycle. These valves are called automatic because they open and close depending on the pressure difference between the cylinder and suction/discharge chamber, established by piston motion (Rovaris and Deschamps, 2006).

A number of works have been made to shed some light on these mechanisms. C. Deschamps and Pereira (2002) predicted gas pulsations inside suction muffler according to two different methodologies: i) a Resonant Helmholtz model and ii) a one dimensional fluid dynamics model. After finished the analyses, it was concluded that the fluid dynamics model offers a good route to simulate the gas dynamics of suction valve, considering the muffler. In another work, Junghyoun Kim and La (2006) aimed to investigate the dynamic of suction valve considering forces that originate the motion. This goal has been achieved by conducting experiments and and analising the results from a fluid-structure interaction model of the valve. The results led to a validated model of the valve, from which it was possible to derive an optimized valve prototype. In the work of Klaus Brun (2006) a semi-active compressor valve has been constructed and tested. An electromagnetic actuator was used to provide a force to reduce the valve impact to the valve seat. It was found that this system has a potential capability to improve both valve life and efficiency.

The main objective of the present work is to develop a system to control the suction valve movement aiming to the increase of compressor efficiency and to reduce the noise associated to suction valve movement.

2. Mathematical Method

The numerical model of the compressor has been developed by Ussyk (1984). The code accounts for piston displacement as a function of crankshaft angle, the thermodynamic process inside the cylinder, the mass flow rate through the valves, valves dynamics, gas pulsations inside the mufflers and refrigerant thermodynamical properties. During the compressor cycle several parameters are available, such as instantaneous pressure throughout the compressor, mass flow rate, valve dynamics, energy and mass losses (Rovaris, 2004). The transient equations associated to the compressor simulation code are solved via a fourth order Runge-Kutta method. More details on the compressor simulation program can be obtained in the work of Fagotti (1994).

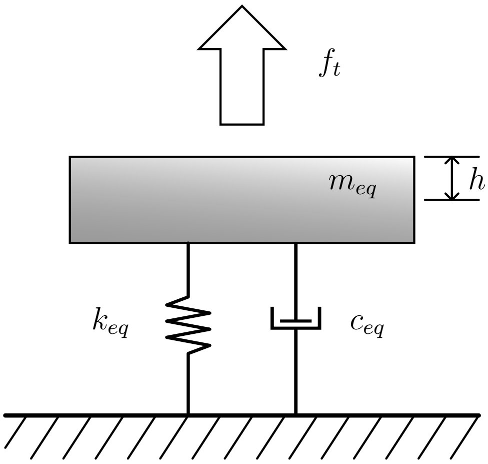


Figure 1: Model of suction valve.

Valves are modeled as a 1 degree-of-freedom element, as shown in the Fig. 1 and their movement are ruled by the expression:

$$M\ddot{h} + C\dot{h} + Kh = f_t \tag{1}$$

The present study assumes the reed moving parallel to the valve seat, with h being the instantaneous distance between the reed and the seat, $\dot{h}$ and $\ddot{h}$ the velocity and acceleration of the reed, respectively. The valve stiffness (K), mass (M) and damping (C) coefficients are determined experimentally. We can rewrite Eq. (1) in terms of equivalent natural frequency ω_n and the damping ratio ζ, which becomes:

$$\ddot{h} + 2\zeta\omega_n\dot{h} + \omega^2 h = \frac{f_t}{M} \tag{2}$$

where the equivalent natural frequency and the damping ratio are given by:

$$\omega_n = \sqrt{\frac{K}{M}} \text{ and } \zeta = \frac{C}{2M\omega_n}\ , \tag{3}$$

respectively. The variable f_t represents the force acting on the reed during the compressor cycle and is defined as:

$$f_t = f_p + f_o \tag{4}$$

where f_p and f_o represent the force generated by the pressure difference across the valve and other forces acting on the valve, respectively. The force f_p is obtained through the use of the concept of effective force area (A_{ef}). According Rovaris, A_{ef} can be understood as a parameter related to how efficiently the pressure difference Δp_v opens the valve, that is, $A_{ef} = f_p/\Delta p_v$. A_{ef} is determined by obtaining a (quasi)-steady flow, i.e., a hovering valve plate, and measuring the values of h and Δp_v (Habing and Peters, 2006).

Since in the present work is desired to impose different movements on the suction valve movement, an additional force is added to Eq. 4. This component, known as the force of actuation (f_{at}), is calculated in such a way an additional component is added to the system resulting in a desired movement on the suction valve. Adding the force of actuation in the Eq. 4 results the following equation:

$$f_t = f_p + f_o + f_{at} \tag{5}$$

To evaluate how the modification on suction valve movement affects the generated noise of compressor, the pulsation of gas inside the suction muffler (PSC) is obtained numerically. The main transmission path of noise associated to suction valve movement is shown in the Fig. 2. The pulsation inside the suction muffler excites the cavity, which, in turn, excites the compressor shell. Finally, the vibration of the shell implies in sound radiation to the compressor's surroundings. Since the pulsation inside suction muffler is the source of this transmission path, it is going to be evaluated for each modified aperture profile.

In order to get an indication of the sound pressure level (SPL) of PSC for each aperture profile, a new parameter is declared as follows:

$$TN = 10\log_{10}\left(\sum \frac{P_n^2}{P_0^2}\right) \tag{6}$$

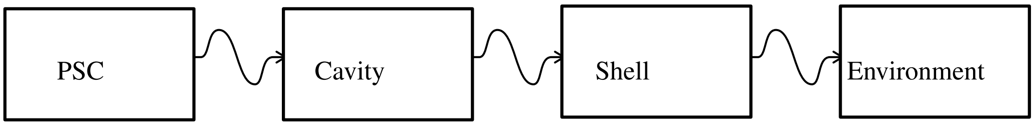


Figure 2: Diagram depicting the main transmission path associated to the suction valve movement.

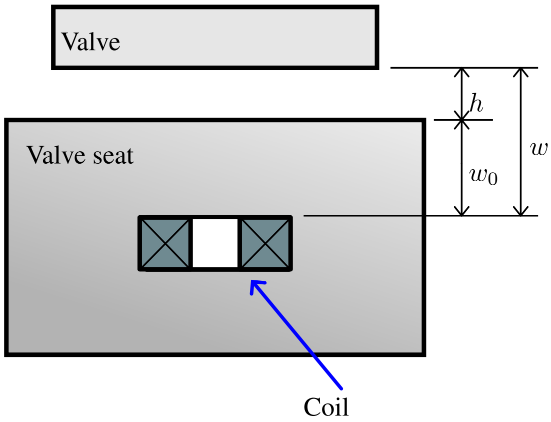


Figure 3: Schematic of the assembly of suction valve and EMA.

where P_n refers to the pressure for frequency n and P_0 is the reference pressure, which is equal to 2 $\times 10^{-5}$. As shown in the Eq. 6, TN is measured using an logarithm scale and express the maximum SPL inside the frequency range adopted.

The efficiency of the compressor for each modified aperture profile of suction valve will be measured by using of the concept of the coefficient of performance (COP). COP is well-known parameter used to characterize the thermal efficiency of compressors and usually is defined as the ratio between the useful refrigerant effect and the net energy supplied by the electrical motor (Pérez-Segarra *et al.*, 2005).

2.1 Electromagnetic Actuator

An electromagnetic actuator (EMA) is modeled to produce the necessary f_{at} in order to modify the suction valve's movement. The basic idea is use a coil to generate an electromagnetic force, as shown in the Fig. 3. The magnetic force is approximately proportional to the square of the applied current i and is inverse proportional to the square of the air gap w, i.e.,

$$f_{mag} \propto \frac{i^2}{w^2} \tag{7}$$

using a proportionality term, Eq. 7 can be written as:

$$f_{mag} = K_l \frac{i^2}{w^2} \tag{8}$$

where K_l is known an the coefficient of electromechanical conversion (Roberto Galvão and Machado, 2003). In order to apply the linear control techniques on Eq. (8), it should be linearized by performing a Taylor expansion, neglecting the terms with order higher than 1, as expressed in Eq. (9)

$$\bar{f}_{mag} = f_{mag_0} + \kappa_1 \Delta i + \kappa_2 h \tag{9}$$

where $\bar{f}_{mag}$ is the linearized magnetic force. The term Δi represents the variation of current with reference to the current at the point of operation ($\Delta i = i - i_0$) and h refers to the variation of air gap ($h = w - h_0$), where w is the total distance between EMA and the valve. This relationship can be viewed in the Fig. 3. On the other hand, κ_1 and κ_2 are constants associated to the variation of current and displacement, respectively. The values of these constant can be obtained through experiments. Herein the results obtained by R. Gomes and Neto (2003) will be used.

Considering the magnetic force generated by the EMA as the force of actuation (i.e., $f_{at} = f_{mag}$) and substituting in Eq. (2), the following expression is obtained:

$$\ddot{h} + 2\zeta\omega_n \dot{h} + \omega^2 h = \frac{1}{M}(f_p + f_{mag_0} + \kappa_1 \Delta i + \kappa_2 h) \tag{10}$$

2.2 Modeling of Suction Valve and the EMA System

From Eq. (10) there is a relationship between the displacement of valve and magnetic force generated by EMA. Applying the state space representation on the Eq. (10) considering as state variables $x_1 = h$, $x_2 = \dot{h}$, $x_3 = i$, as input signal $u = V$ and output signal $y = h$ results the following expression:

$$\begin{Bmatrix} \dot{x}_1(t) \\ \dot{x}_2(t) \\ \dot{x}_3(t) \end{Bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ -\omega_{eq}^2 + \frac{\kappa_2}{M} & 2\zeta\omega_n & \frac{\kappa_1}{M} \\ 0 & 0 & -\frac{R}{L} \end{bmatrix} \begin{Bmatrix} x_1(t) \\ x_2(t) \\ x_3(t) \end{Bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \frac{1}{L} \end{bmatrix} u(t) \tag{11}$$

$$y(t) = \begin{bmatrix} 1 & 0 & 0 \end{bmatrix} \begin{Bmatrix} x_1(t) \\ x_2(t) \\ x_3(t) \end{Bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix} u(t) \tag{12}$$

Table 1: Parameters considered in the present study.

Parameter	Value	Unit
R	1	Ω
L	0.01	H
K_l	0.0001	
κ_1	0.392	N/A
κ_2	1.176	N/m
ω_{eq}	1325.8	rad/s
w_0	0.005	m
ζ	0.01	
k_{eq}	192	N/m
m_{eq}	1.09×10^{-4}	kg

Table 1 shows the parameters and respective values used here. Considering these values in the simulations, Tab. 2 presents some dynamic characteristics of valve and EMA assembly. The Table 1 shows that the rising time is insufficient to meet the requirements of the project, since it takes too long to achieve the desired value. Therefore, a control system is implemented to alter the dynamic of the assembly of the suction valve and the EMA. Linear Quadratic Regulator (LQR) is select to perform this task. In LQR is sought a control that minimizes a cost function, which is shown below:

$$J = \int_0^\infty \left(\mathbf{x}\mathbf{Q}\mathbf{x}^T + \mathbf{u}^T\mathbf{R}\mathbf{u} \right) \mathrm{d}t \tag{13}$$

where **Q** and **R** are matrices associated to the energy of state and input signal, respectively. Furthermore information can be find in literature, for instance in DORF (2001).

Applying a step function at the input of the valve and the EMA system, the resulting dynamic can be viewed at the Tab. 3. It is observed from the result of rising time that the system with LQR control responds much faster than the system without control. Although an increase in maximum overshoot is observed a significant reduction of settling time is obtained as well.

Based on the aforementioned results the system with LQR control is chosen to be used inside the compressor simulation to modify the suction valve aperture.

3. Results and Discussion

In order to evaluate the performance of reciprocating compressor with a different suction valve aperture, different movements are applied on the suction valve. The modification of suction valve is achieved is by the LQR control system constructed in last section. Figure 4 depicts how the reference movements are applied and the respective output (displacement output, voltage, current, f_{at}).

Table 2: Dynamic characteristics of suction valve and EMA assembly.

Characteristic	Value
Rising time	19.9 ms
Maximum overshoot	3.23 %
Settling time	98.7 ms

Table 3: Dynamic characteristics of suction valve and EMA assembly with LQR control.

Characteristic	Value
Rising time	0.15 ms
Maximum overshoot	7.83 %
Settling time	0.43 ms

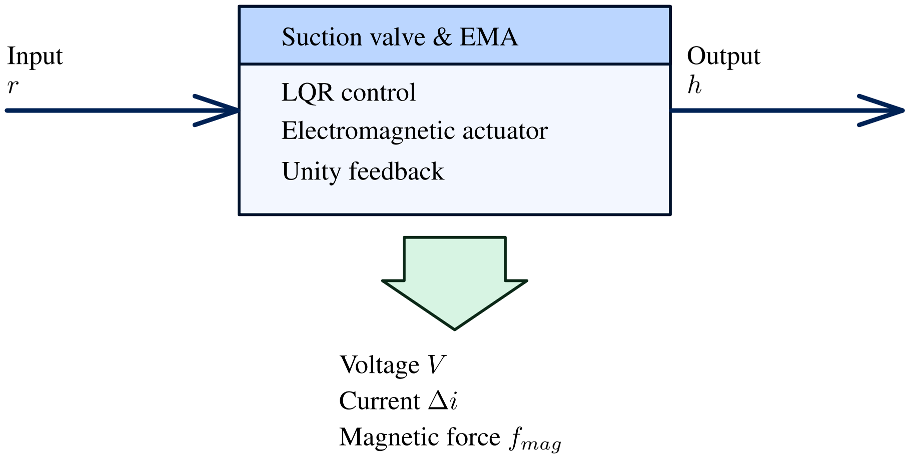


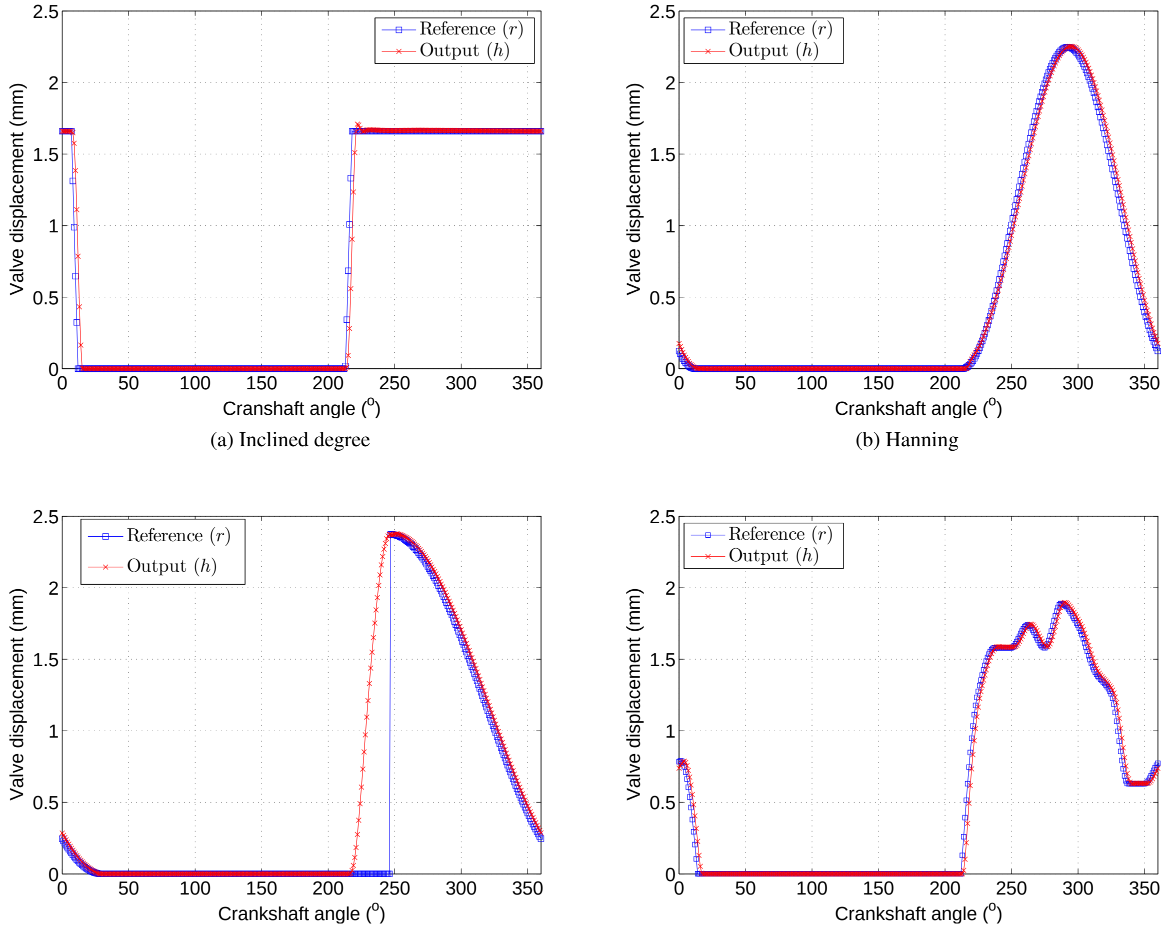
Figure 4: Diagram depicting the control system of the suction valve aperture.

Figure 5 shows the reference movements applied in the input of control system of suction valve movement and their respective outputs. It may be noted that even with the presence of a small delay, the proposed modified aperture profiles were obtained on suction valve. Figure 6 presents the values of the voltage, current and magnetic force along one duty cycle necessary to modify the valve aperture profile. From Fig. 6a can be noted that a high value of voltage is necessary to generate Perfect degree, Hanning and Optimized movements. For the Original&cosine movement is observed and high value at the exact instant that the EMA starts to act on the valve. It can be also noted that the voltage for the Optimized movement has many variations along the period that EMA is acting on the valve. For the results of current, which is shown in Fig. 6b, an out-of-range amplitude is observed in the Perfect degree and also for Optimized movement. A similar behavior is observed for the magnetic force (f_{mag}), shown in Fig. 6c. In a general manner f_{mag} stood within a range between -1 N and 1 N. The f_{mag} for Perfect degree and Optimized movement presents a high amplitude at the moment that the valve is opening since a very high acceleration is applied by these movements. On the other hand, Hanning movement resulted in a negative amplitude in the interval that valve is opening, between 216^o and 254^o. The magnetic force for Original&cosine movement presented only positive values in a smaller period in comparison to others applied movements. This result is useful because it indicates that the EMA can be set to work only forward, which implies in a greater actuator simplicity.

The results of TN of the modified movements in comparison to the normal aperture can be seen in Tab. 4. For the considered transmission path, the modification on suction valve movement resulted in the reduction of the TN. The Optimized movement presented the best result, with a reduction of approximately 3.5 dBA in SPL. Table 4 also presents the results of COP considering the power consumed by the EMA to generated each modified movement on suction valve, which is called herein COP^*. For all modified movements, it is observed a reduction in COP^*, the Inclined degree and Optimized movements presented an accentuated reduction since these movements required an large amount of power to be applied on suction valve. Although in comparison to the Inclined degree and Optimized movement presented better results of COP^*, Hanning and Original&cosine movements presented considerable reductions in comparison to normal aperture, with reductions in COP^* of 10.59 and 6.9 %, respectively. Since this is a preliminary study, a further study in the EMA conception and its construction may improve these results.

4. Conclusions

A control system is developed to modify the suction valve movement with the purpose of improving the compressor's sound radiation without affecting its thermodynamic performance. The control system is implemented by means of an electro-magnetic actuator, which has been evaluated by the use a numerical model of the compressor. The reference movements were applied on the suction valve movement and the results of efficiency COP and total noise (TN) were obtained. From results is observed that through the modification of the aperture profile of the suction valve it is obtained



(a) Inclined degree (b) Hanning

(c) Original&cosine (d) Optimized movement

Figure 5: Movements applied on the suction valve aperture. Blue curves (square markers) are the signal applied at the input of system control and red curves ('x' markers) are the obtained movement.

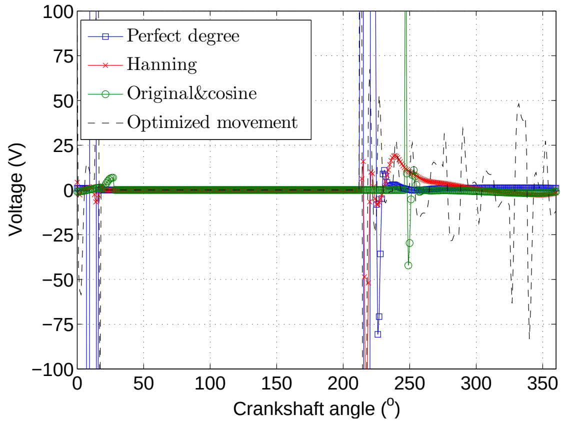
Table 4: Comparison of results obtained by modified movement with normal aperture of suction valve.

Movement	TN (dBA)	COP^*
Normal aperture	31.54	1.472
Inclined degree	29.6	0.158
Hanning	30.49	1.316
Original&cosine	29.01	1.37
Optimized movement	28.07	0.774

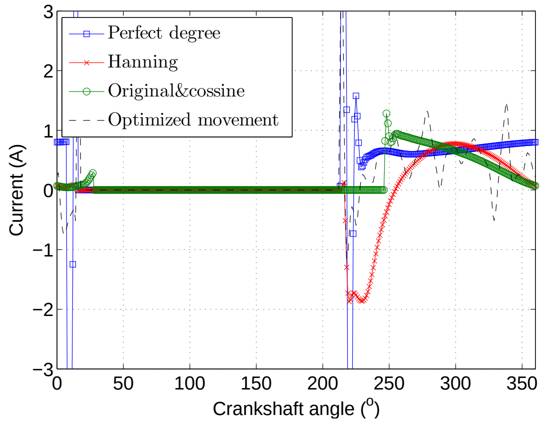
a reduction in TN.The Optimized movement resulted in a reduction of approximately 3 dBA in comparison to normal aperture. The Original&cosine movement presented a reduction of TN as well, of approximately 2.5 dBA. However, it is seen a decrease in the efficiency with the addition of power consumed by the EMA to produce the modified movement. The Inclined degree and the Optimized movement required a large amount of power to provide enough force to produce the modified movement on suction valve.

5. ACKNOWLEDGEMENTS

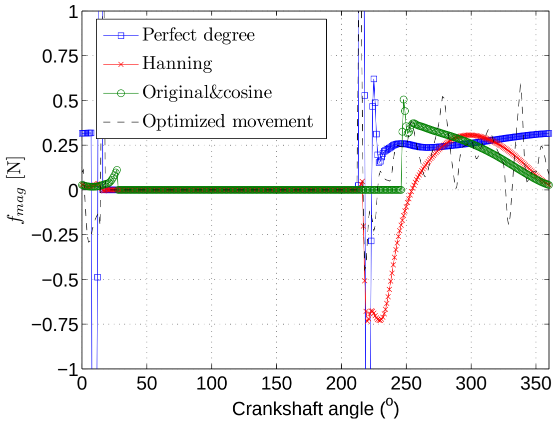
The first author would like to thanks FAPEAM (The State of Amazonas Research Foudation) and EMBRACO for the financial support of this work.



(a) Voltage



(b) Current



(c) Magnetic force (f_{mag})

Figure 6: Results obtained by control system to generate each movement.

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