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STUDY AND DEVELOPMENT OF THE VANE FOR A NEW PNEUMATIC MOTOR-COMPRESSOR.

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Abstract. The rolling piston conversors (RPC) are motor-compressors of simple construction and widely used in industry. In this article structural parts are studied a new model of RPC without the use of valves and it offers a constant torque due a inlet pressure also constant at all the vanes work positions, except in the moment that the it is totally retract. The methodology used was modelling analytically the conversor for a variable blade profile, so in this way it will be able to control the exposing pressure area and also the equivalent force actuation radius applied at the vane. The topological analysis of the material distribution were used at the structural project of pieces. Lastly, the main result of this work was getting new model of motor-compressor with characteristics that make it technically viable.

Keywords: Dynamic model, Rolling piston compressor, Swing vane compressor.

1. INTRODUCTION

The rolling piston conversors (RPC) are widely used at refrigeration, because it has some benefits as, simple construction, few pieces, reduced weight and size, insulation between the chambers of input and output, as low cost, that attract attention of industry for researchers since 1970's according to the work of papers (Yanagisawa *et al.*, 1982), (Ooi, 2005) and (Kussul *et al.*, 2016). A schematic of RPC can be seen at Fig. 1.
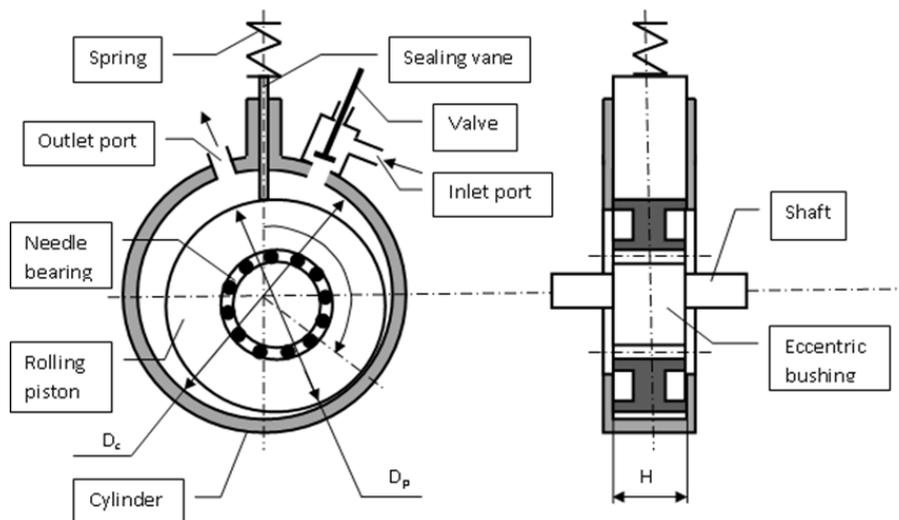

Figure 1. Schematic operation RPC (Source: (Kussul *et al.*, 2016)).

The operation consists at a rolling piston, that when it spins at eccentric shaft to cylinder causes a volume increase in the input chamber, sucking the inlet air, and reducing the output chamber volume, compressing the outlet air. This is only possible because the blade that slides on piston surface, sealing the output of input chamber. The spring keeps the contact between the vane and piston.

This configuration cause big torque oscillations at shaft. In Fig. 2, present in the work of Ishii *et al.* (1984), is showed components of torque load in a test of suddenly stop at the compressor in constant operation. Realized the big variation of torque due the pressure of the cylinder before even reversing direction (at 180 degrees), it starts intense, continues weak and suddenly stops. And in Fig. 3 is showed the torque pattern at constant operation.
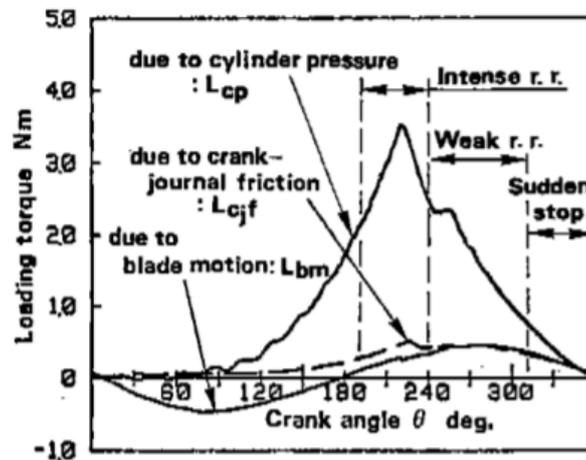
Figure 2. Loading torque of RPC at constant operation (Source: (Ishii *et al.*, 1984)).
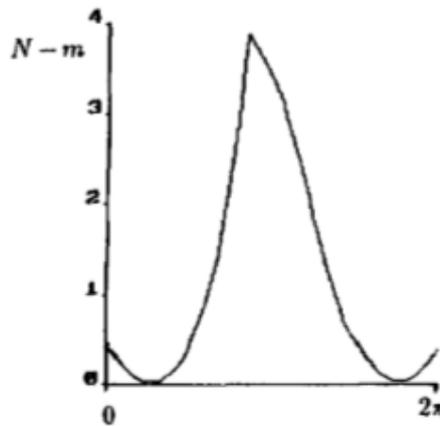

Figure 3. Torque pattern at RPC (Source: (Young *et al.*, 1993)).

This study proposal is to model and scale out a conversor for a variable profile of vane to obtain a constant torque due a input pressure also constant, except at the moment that the blade is totally retract.

2. COMPUTATIONAL PROCEDURE

The development of this project started through the mathematics model of mechanism attending the requirements of constant torque due the constant inlet pressure. For this, the equations of Newton-Euler and statics fluids over motor housing and over the blade searching was implemented computationally to determinate better geometry for it.

2.1 Mathematic Model

Knowing that the torque is given by Newton-Euler relations as the product between the radius in which the force will be apply by the force itself, we have the following relation of Eq. 1.

$$T = R \cdot F \tag{1}$$

Where, T is the torque at the machine shaft, in $N \cdot m$; R the radius where the force will act, in m; F the applied resultant force, in N.

However, the radius R of equivalent force actuation is just the distance from machine shaft to the vane area centroid, given by Eq. 2.

$$R = \oint \frac{Y \, da}{A} = \frac{1}{A} \oint Y \, da \tag{2}$$

Where, A is the total vane area exposed to pressure, in m^2; da is the vane area element; Y is the distance between the shaft and the vane area element da, in m.

Furthermore, the force F is provoked by the pressure difference P between the chambers of inlet and outlet. This pressure acts on the vane area A, we obtain the Eq. 3.

$$F = P \cdot A \tag{3}$$

Where, P is the pressure difference between the inlet chamber and outlet, em N/m^2; Substituting the Equations Eq. 2 e Eq. 3 in Eq. 1 we have the Eq. 4.

$$T = \frac{1}{A} \oint Y da \cdot P \cdot A = \oint Y da \cdot P \tag{4}$$

Both torque T and pressure P must be constants, as consequence the Eq. 5 is reached.

$$\frac{T}{P} = \oint Y da = G \tag{5}$$

Assuming G is the machine torque gain, in m^3.

Therefore, to ensure a constant torque due to a constant pressure, it is enough that $\oint Y da$ defined on the vane surface, normal to pressure P, in others words, the profile blade itself, must be constant for any vertical blade displacement.

2.2 Profile Definition and Optimization

For reasons of balancing, the curve that caracterizes the profile must be simetric with respect to the radial shaft Y, as showed at Fig. 4. Thus, the function must be even with respect to radial shaft. It could be seen at Fig. 4 as a scheme for the development of Eq. 6, that defines a general profile.
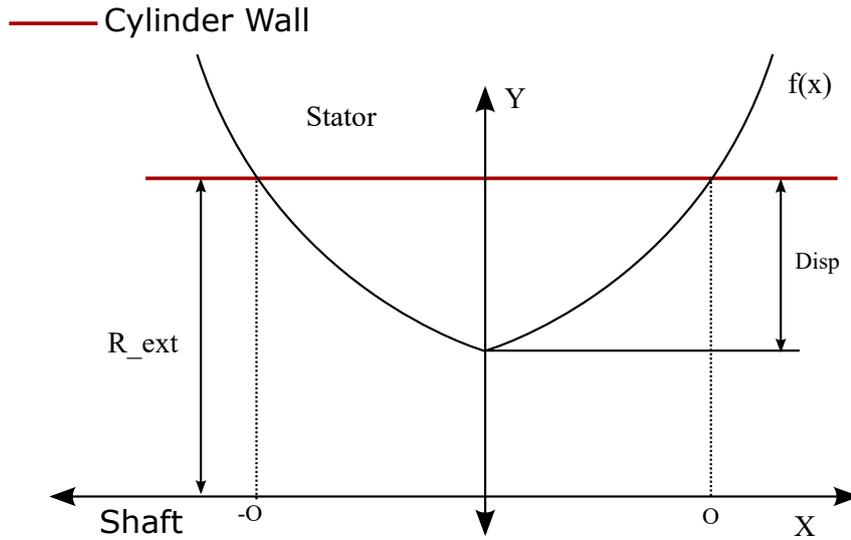

Figure 4. Generic Vane Profile (Source: By Authors).

$$G = \int_{-O}^{O} \int_{f(x)}^{R_{ext}} Y dy dx = 2 \int_{0}^{O} \int_{f(x)}^{R_{ext}} Y dy dx = 2 \int_{0}^{O} \left(\frac{f(x)^2}{2} - \frac{R_{ext}^2}{2} \right) dx = \int_{0}^{O} f(x)^2 dx - R_{ext}^2 O \tag{6}$$

Defining the linear function as $f(x) = ax + R_{ext} - Disp$ to describe the vane profile, and substuting it in the Eq. 6, we achieve the Eq. 7.

$$G = -\frac{Disp^2(Disp - 3R_{ext})}{3a} \tag{7}$$

For contructives reasons, the maximum reachable displacement ($Disp$) occurs when the angular coeficient $a = 1$ (generating an angle of 45 degrees with respect to the shaft). In order to maintain G constant, when the displacement $Disp$ decreases (and it approaches to zero), the angular coeficient a tends to zero more quickly. Thus, the point O tends to infinite, when $Disp$ tends to zero. To mitigate this effect, which would make any construction impossible, it is assumed that a $Disp$ critically smaller than the maximum displacement ($Disp_{max}$) has a value approximately equal to zero.

Using this approximation, the construction becomes feasible without significant losses in the model's accuracy. Thus, for the analysis of the topological optimization of material's distribution, and consequently, the volume occupied, it is

fulfilled the minimization of the distance between the point o (referring to the maximum displacement ($Disp_{max}$)) and the point o' (referring to the minimum displacement ($Disp_{min}$) more smaller than maximum displacement($Disp_{max}$)). This minimization entails the reduction of the hidden area of the vane, without prejudice the exposed area, which is essential.

A point O generic is defined by Eq. 8.

$$O = \frac{Disp}{a} \tag{8}$$

The point o, for a $Disp_{max}$ and $a = 1$ is defined by Eq. 9.

$$o = Disp_{max} \tag{9}$$

The displacement $Disp_{min}$ is given by $Disp_{max}$ reduced by a factor f and it is defined by Eq. 10.

$$Disp_{min} = Disp_{max} \cdot f \tag{10}$$

Therefore, the point o' is given by Eq. 11.

$$o' = \frac{Disp_{max} \cdot f}{\frac{-(Disp_{max} \cdot f)^2 (Disp_{max} \cdot f - 3 \cdot R_{ext})}{3 \cdot \frac{(Disp_{max}^2 (Disp_{max} - 3 \cdot R_{ext}))}{3}}} \tag{11}$$

Then the target function D of distance $D(Disp_{max}, R_{ext})$ to be minimized is given by Eq. 12.

$$D(Disp_{max}, R_{ext}) = \sqrt{(o - o')^2} \tag{12}$$

The optimization is fulfilled by Steepest Descent method, where in each iteration the independents variables of target function are directed by the gradient to decrease the value of the function D according to Colaço *et al.* (2006). The quantity of iterations until reach a local or global minimum depends of the size of the step defined. The stop criterion should be when the minimum value is reached, or the gradient is equal zero, or in this case a satisfactory value. The Eq. 13 describes a vector h of independents variables of the target function in each iteration.

$$h^{K+1} = h^K - \lambda \cdot \nabla D(h^K) \tag{13}$$

Where K is the iteration index; h is the vector of independent variables of target function; λ is the step; and D is the distance function to be minimized.

The occupied volume sinalizes what is the material expenditure and is given by the portion occupied from the center shaft, until the blade boundary. The Equation in which represents this volume, using Pappus Guldin's theorem, is the Eq. 14.

$$v = \left(\frac{4 \cdot \sqrt{o'^2 + (o \cdot f)^2}}{3 \cdot \pi} + R_{ext} \right) \cdot \left(\frac{\pi \left(o'^2 + (o \cdot f)^2 \right)}{2} \right) + \left(\frac{R_{ext}}{2} \right) \cdot \left(2 \cdot \sqrt{o'^2 + (o \cdot f)^2} \cdot R_{ext} \right) \tag{14}$$

Where v is the volume occupied by the set, in m^3.

3. RESULTS AND DISCUSSION

The results for optimization, calculus and graphics production were obtained by using a computer with configuration: CPU, Intel(R) Core(TM) i7-6700 3.40 GHZ, Ram Memory 64,0GB. Operational System Windows 10 pro 64 bits. Software Matlab 2016a.

The results for optimization of the target function acquire from Eq. 12 by Steepest Descent method used in Eq. 13, the start values were designated by the independents variables of target function of Eq. 12. The stop point was accomplish iterations until the distance D was smaller or equal to $0.10(m)$, with the intention to satisfy the project specification. In Tab. 1 is possible to verify the results after the program execution.

Table 1. Numericals results for optimization

Variable	Initial value	Optimized value
R_{ext}(m)	0.070000	0.067843
$Disp_{max}$ (m)	0.050000	0.011282
D (m)	0.340244	0.095875
K (-)	1	12
$v(m^3)$	0.058290	0.002538
Execution Time (s)	0	0.271417

At the Fig. 5 is possible to visualize the evolution of the optimization with the passage of the iterations until the stop.
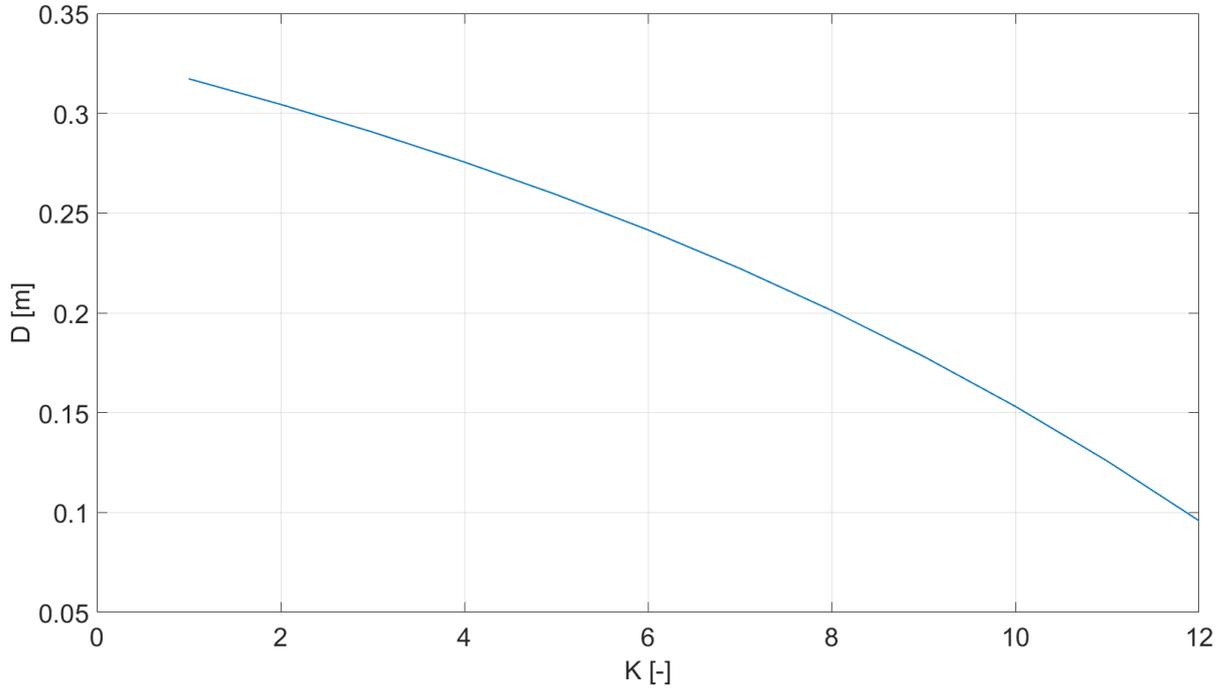

Figure 5. Optmization Evolution

With the results obtained through the topological optimization and the definition of the profile, in accordance with the requirements of the constant torque due to the constant pressure of the input, then, we have the variable profile as can be seen in the Fig. 6.
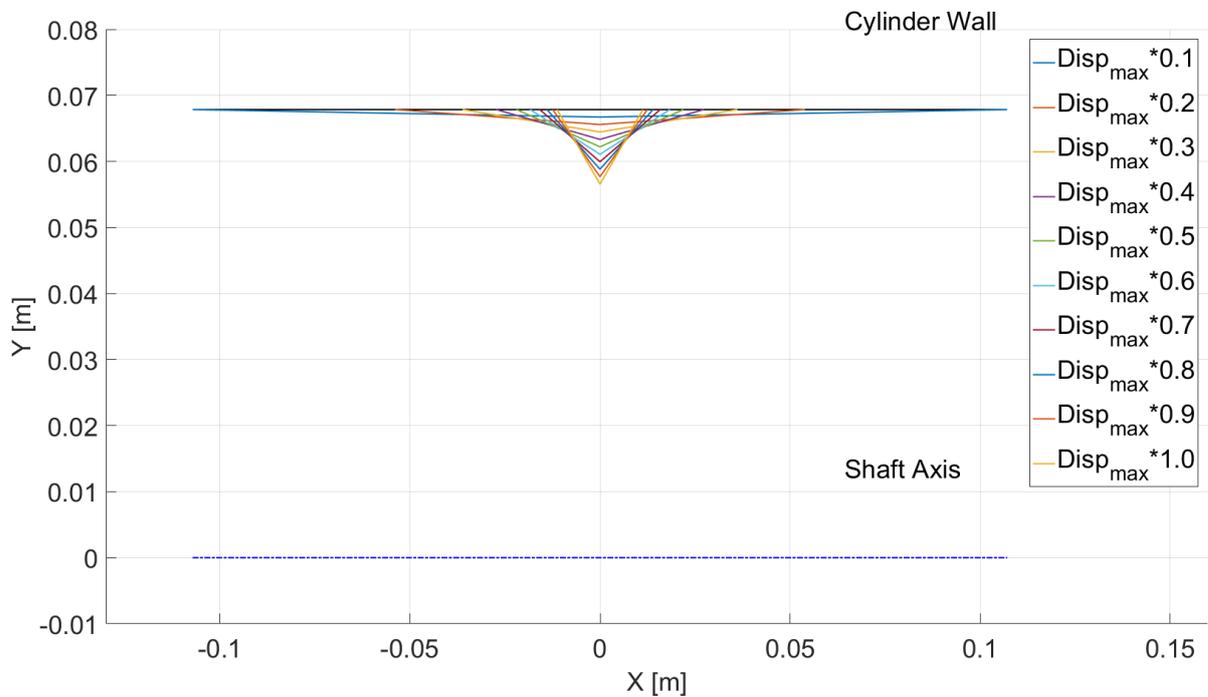

Figure 6. Variable Profile (Source: By Authors).

To establish a mechanism that complies this variation of profile format. According to the purpose of the project, the vane was divided into two pieces. Each piece has the shape of a 90 degrees circular sector, in which the surfaces are in mutual contact sliding one over the other, providing the necessary insulation between them. The vertices of the two pieces, they are joined allowing the centered rotation in this union. With these features the vane is allowed to have a vertical (upward and downward) movement towards the radius of the machine axis, and at the same time, a pivoting

movement which adapts the blade assembly to the variable cam shape.
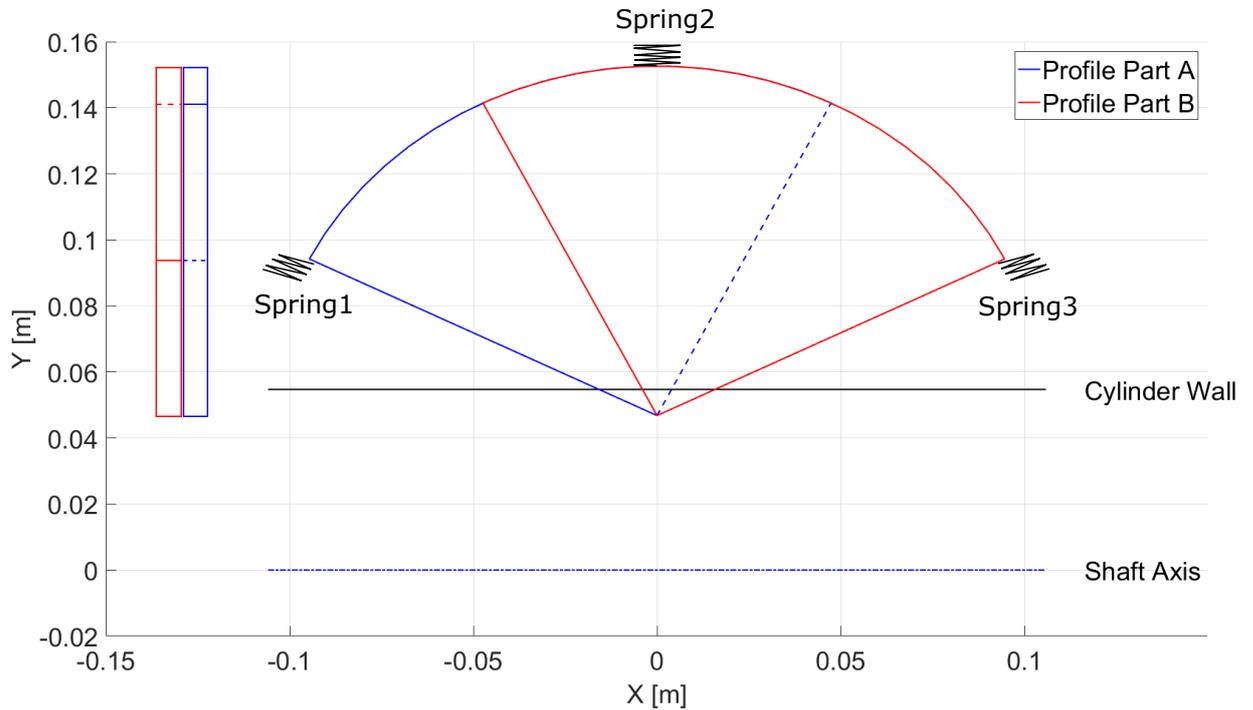

Figure 7. Mechanism Vane (Source: By Authors)

At the Fig. 7, the general scheme of the set is shown to play the variable cam form. The area exposed to pressurized air is below the cylinder wall, while the remainer distributes and transmits the charge to the stator. To force the contact of the blade with the cam, guided springs are used. They tend to keep the blades away from each other by pressing them sideways against the sides of the cam. Moreover, springs are used to execute vertical force and keep the peak of the assembly in contact with the vertex of the valley generated by the cam.

The Figures from Fig. 8 until Fig. 11 show the result of the pallet's movement for four displacements. Note by the tendency of the movement, that if the linear function's angular coefficient, which describes the profile, was greater than 1, $a > 1$, the pallet would increase its area and would try to invade the cam, and thus would not be able to keep up with the profile variation.
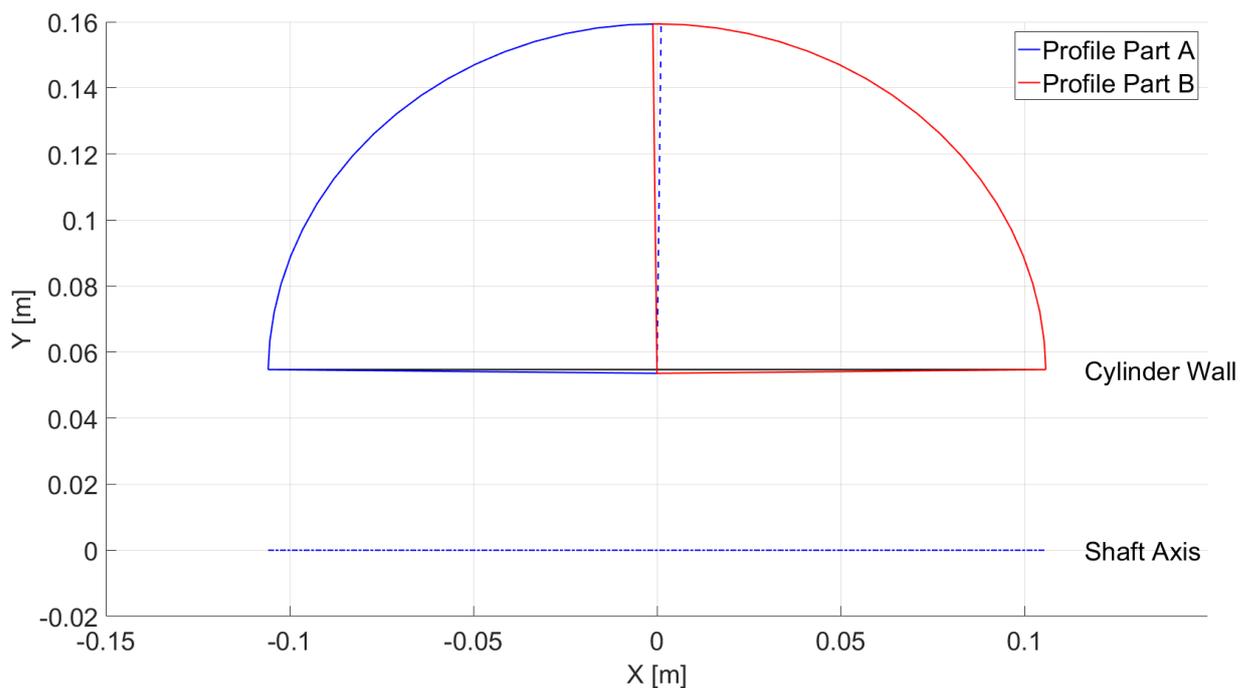

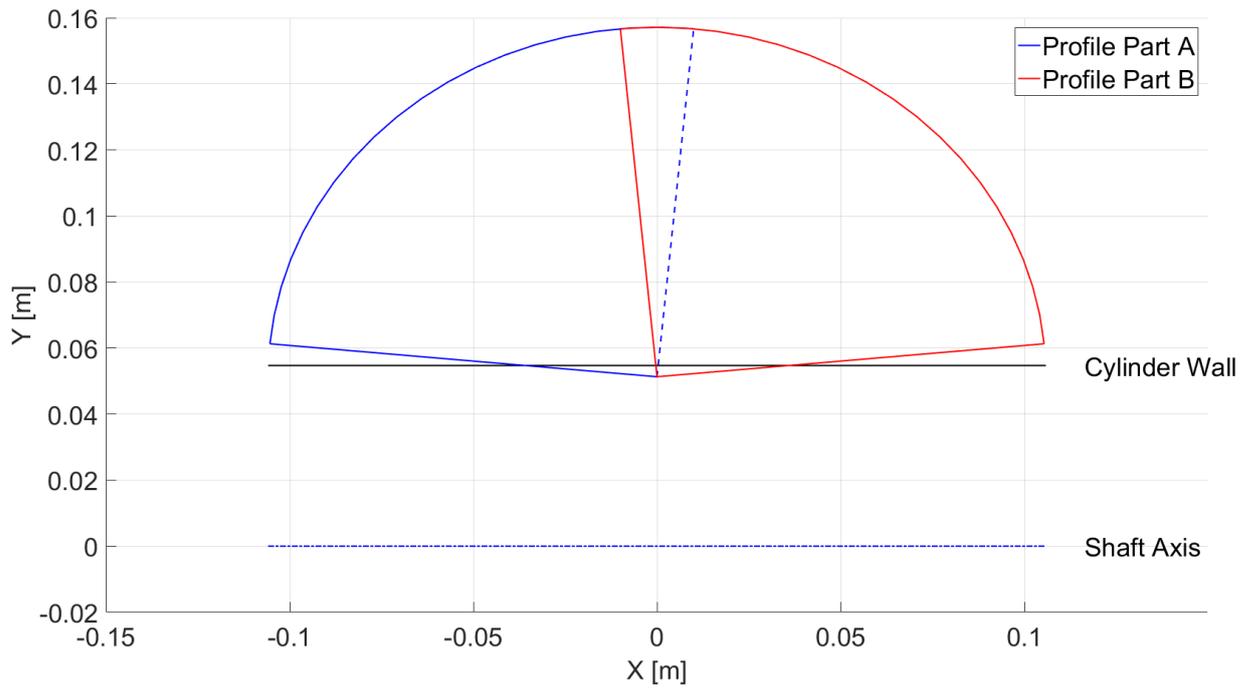Figure 8. Vane Position at 10% of Displacement (Source: By Authors)

Figure 9. Vane Position at 30% of Displacement (Source: By Authors)
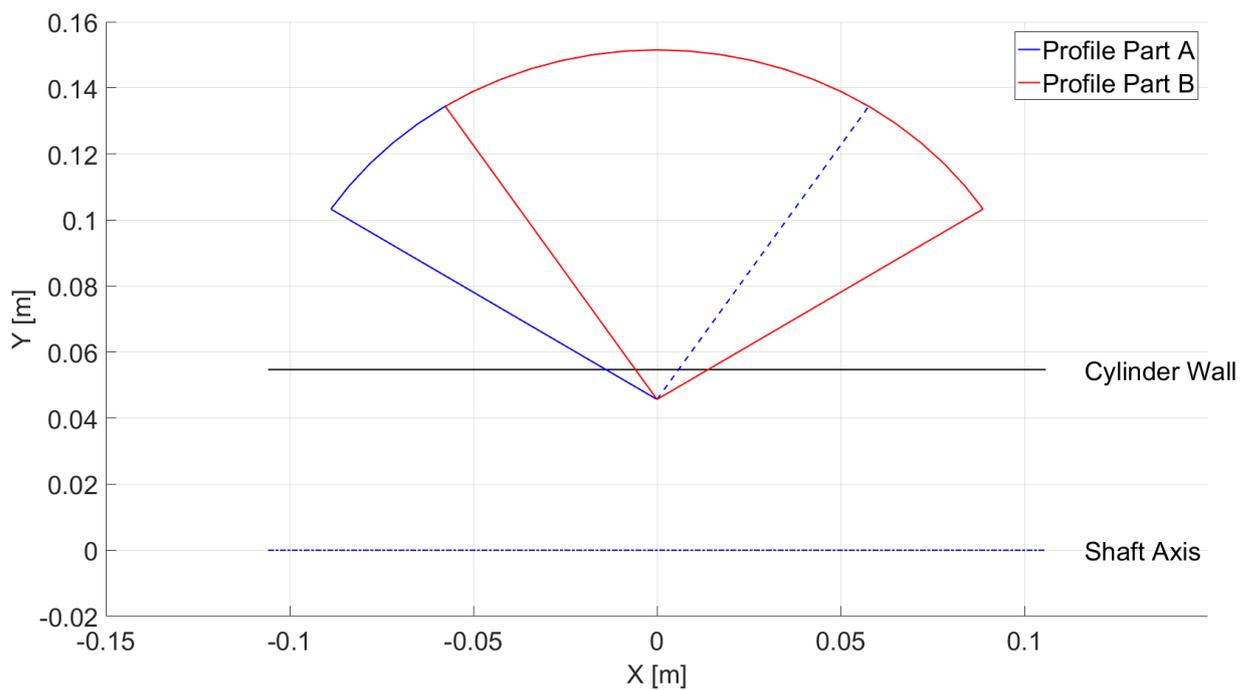

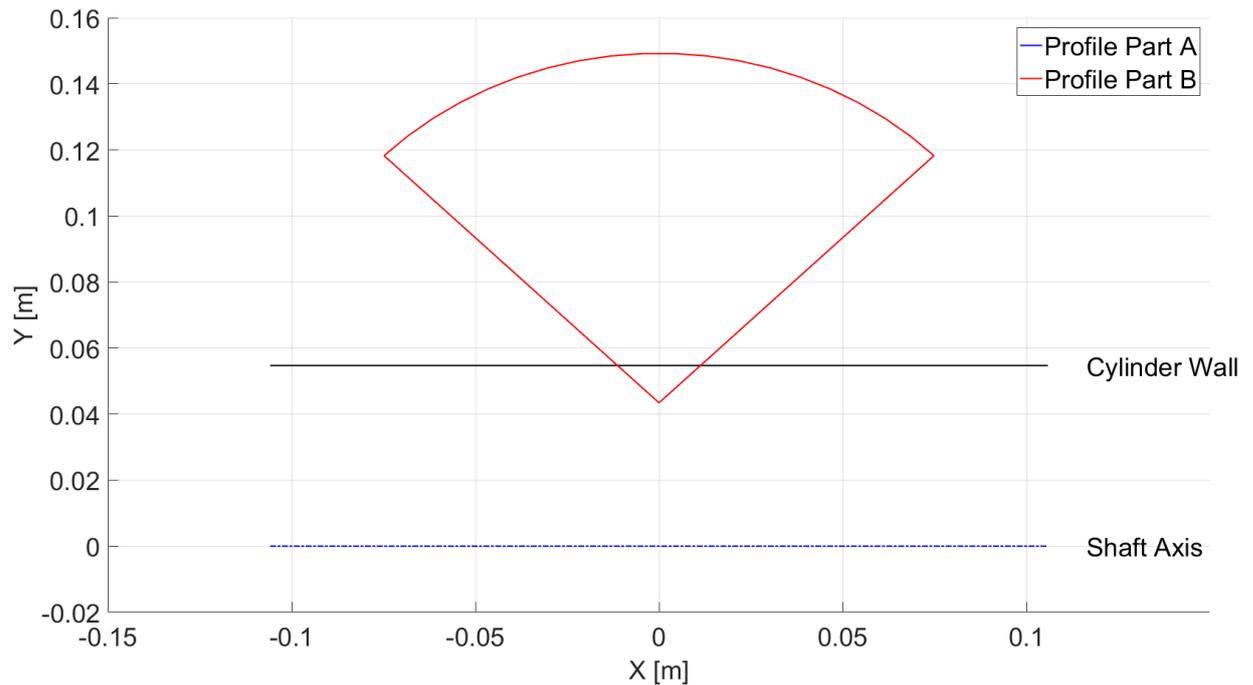Figure 10. Vane Position at 80% of Displacement (Source: By Authors)

Figure 11. Vane Position at 100% of Displacement (Source: By Authors)

4. CONCLUSIONS

The study, of the model profiles of blade, allowed to advance with the development of new model of motor-compressor with constant torque. The optimization performed reduced the unnecessary expenses of material without the damage of the exposed area. The structural analysis allows the future dimensioning of the pieces that will vary according with the degree of application's demand. Therefore, a mechanism was found that would agree with the variable profile proposed for the constant torque due to the constant pressure of the input, thus reducing vibrations and noises of the device itself and also its propagation to the peripherals.

This study can conclude through the analysis and simulations that a profile of triangular shape is the most viable model, considering the kinematics aspects, the topological and the flexibility to develop others projects.

5. ACKNOWLEDGEMENTS

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