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NUMERICAL ANALYSIS OF NEW SUPERELASTIC SMA DAMPERS FOR MECHANICAL VIBRATION CONTROL IN ROTOR SYSTEM

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Abstract. *Mechanical vibrations in rotor systems can become a serious problem that affects performance. Vibration control in these systems has been well studied in the last years, mainly applying Shape Memory Alloys (SMA) to reduce vibration problems and currently the superelastic behaviour of SMA is being extensively studied and applied. Although designs of SMA dampers devices are not being planned for better distribution of the vibration force. This paper aims to simulate two different types of passive dampers made of superelastic SMA to mitigate mechanical vibrations in a rotor system. Numerical analysis was carried out with two SMA dampers of different designs. Experimental data of an unbalanced rotor system were employed to simulate the strain behavior in these new types of passive dampers. The damper designed with a larger contact area with the rotor shaft showed better behavior for the phase transformation of the SMA, making it possible to reduce the mechanical vibration of the system.*

Keywords: *Rotor Systems, Shape Memory Alloys, Dampers, Mechanical Vibrations, Superelasticity.*

1. INTRODUCTION

Vibration control in rotor system is a subject where many researchers are working in these last years, applying smart materials as Shape Memory Alloy (SMA) to avoid the high amplitudes caused by resonance. The capacity to return to the original shape of the SMA, because the martensite and austenite phases of the material, has been extensively studied and applied using the shape memory effect (SME) or superelasticity (SE) behaviors (Liang e Rogers, 1993).

Since the SE behavior of the SMA is associated to a phase change with applied stress, using the vibration of the rotor system it can provide strain and changes the crystalline structure, creating a hysteresis zone and increasing the damping of the system, decreasing the vibration. However, this direct application of superelastic SMA to control vibrations in rotor systems using the damping of the material (mechanical hysteresis), has not been well studied. Some researchers applied this behavior as in (Liu *et al.*, 1994) or with another type of support bearing, for example magnetic (Enemark e Santos, 2016a). The design normally applied to the dampers are helical coil springs (Enemark e Santos, 2016b) or wires (Yogaraju *et al.*, 2016).

The design of superelastic SMA damper is important principally to obtain a better distribution of the vibration forces and to change the mechanical response of the SMA and then attenuating the vibration.

This paper aims to perform a numerical analysis of a new conception of spring passive dampers made of a superelastic SMA, to introduce it in the support bearings of the rotor system to reach a better control of mechanical vibrations.

First was made an experimental test with a real rotor system, to achieve the maximum displacement in unbalanced system. All analysis were performed using the software *Ansys Workbench 17*, where was applied a displacement on two types of damper, convex sheet spring and concave irregular sheet spring, in a superior part where the bearing makes contact and applied the superelastic SMA theoretical data. All analysis were taken the force-displacement curve with that was possible to achieve the effective stiffness and the specific damping capacity, this last one shows the capacity of mitigate the vibrations of the rotor system. The concave irregular sheet spring due the biggest contact area of contact with the bearing can distribute in a better way the applied force and with that the specific damping capacity of this damper is bigger than other, in this case being the best damper design to apply in the rotor system.

2. NUMERICAL AND EXPERIMENTAL PROCEDURE

First, was obtained the maximum displacements in a rotor system applying unbalanced masses (1 to 5 grams) in the disk positioned in the middle of the shaft and running up the system (0 to 60 Hz). The rotor system is composed of shaft, disk and two support bearing, being one flexible, as in (Senko *et al.*, 2013). This flexible support bearing is composed by four coil springs of SMA with the memory effect, the same used in Borges (2016). The displacement was measure next to flexible bearing in the shaft with a proximity sensor (SKF CMSS 665) in both axes, Figure 1.

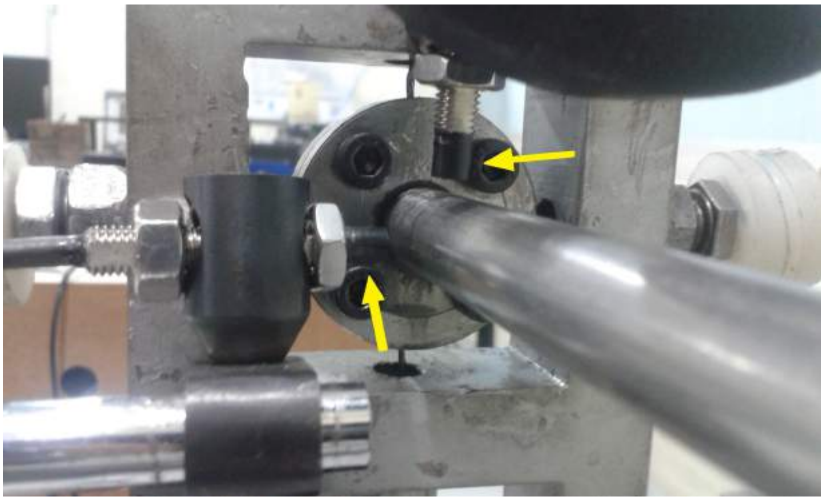

Figure 1. Proximity sensor SKF CMSS 665.

These displacements were used as reference for numerical analysis of the mechanical behavior of the new SMA passive dampers. Two designs of SMA spring passive dampers were evaluated: convex sheet spring and concave irregular sheet spring (Figure 2). This last features a larger contact area with bearing from rotor system.

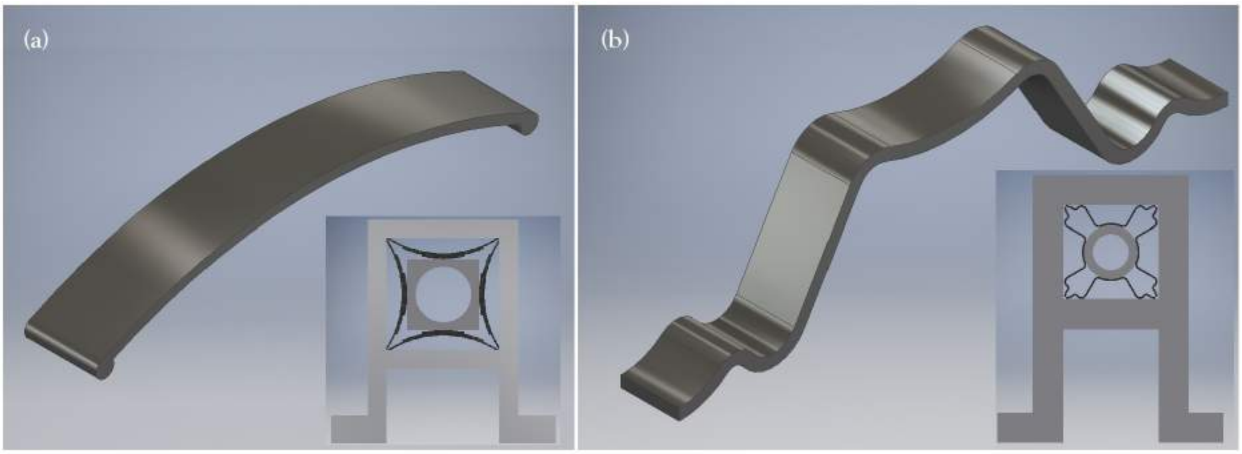

Figure 2. Conceptions of SMA spring dampers: (a) Convex sheet spring and (b) Concave irregular sheet spring.

Both types of the springs have the same width and length, in the Table 1 is possible to see the principal dimensions of each passive damper.

Table 1. Damper dimensions

Damper	Length (mm)	Width (mm)	Thickness (mm)	Height (mm)
Convex sheet spring	48	5	1	7,528
Concave Irregular sheet spring	48	5	1	21,6

Numerical analysis were performed using the FEM in *Ansys Workbench 17*, applying on the passive dampers (Figure 2) the maximum displacement obtained in the unbalancing experiments and added more 5% of that result to provide more strain on that, all in the same direction. The constraints (boundary conditions) applied were to work in a situation closer to a real system. The mesh applied has 2015 elements and 14948 nodes, where hexahedral element with 20 nodes was used in the static analysis with SMA material, Figure **3**.

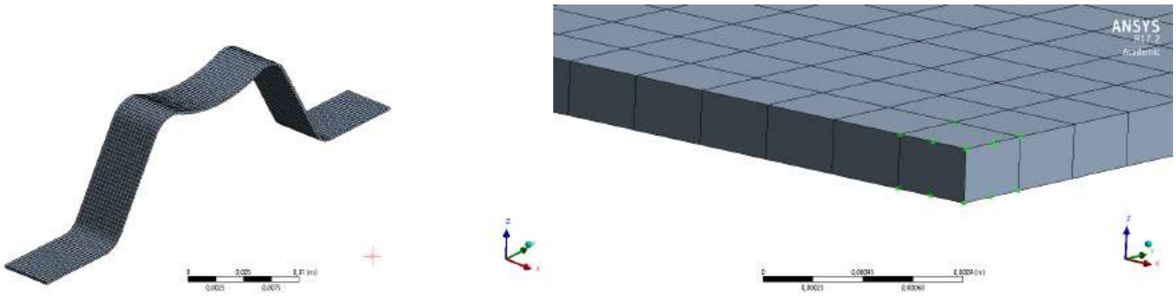

Figure 3. Mesh and hexahedral element with 20 nodes.

From the experimental tests the essentials boundary conditions used was fixed support and frictionless support and natural boundary conditions was a displacement, Figure **4**.

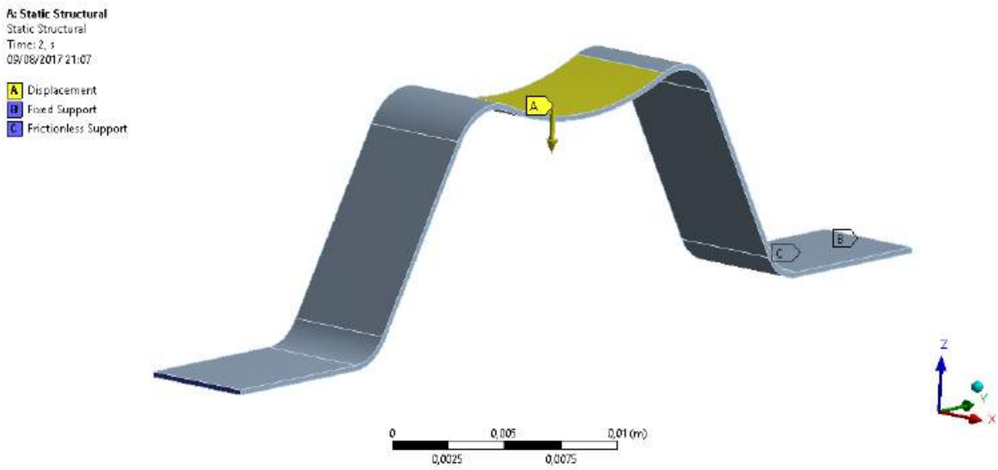

Figure 4. Boundary conditions

Also were obtained in numerical analysis the effective stiffness and the specific damping capacity. According Doggenweiler (2010) the effective stiffness correspond to the slope of the line connecting the origin to the point of the maximum deformation of the curve, Figure 5.

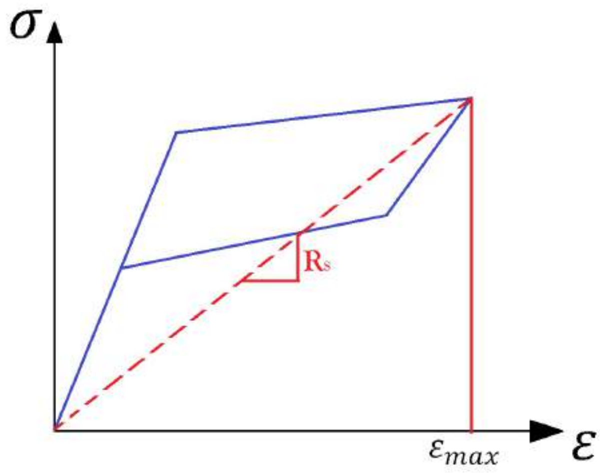

Figure 5. Effective stiffness.

The specific damping capacity is calculated using the following Equation (1),

$$\xi_{eq} = \frac{E_d}{4\pi E_s} \tag{1}$$

where E_d is the area inside the curve A (hysteresis zone) and E_S is the area of the hatched triangle in the curve B, as showed in Figure 6.

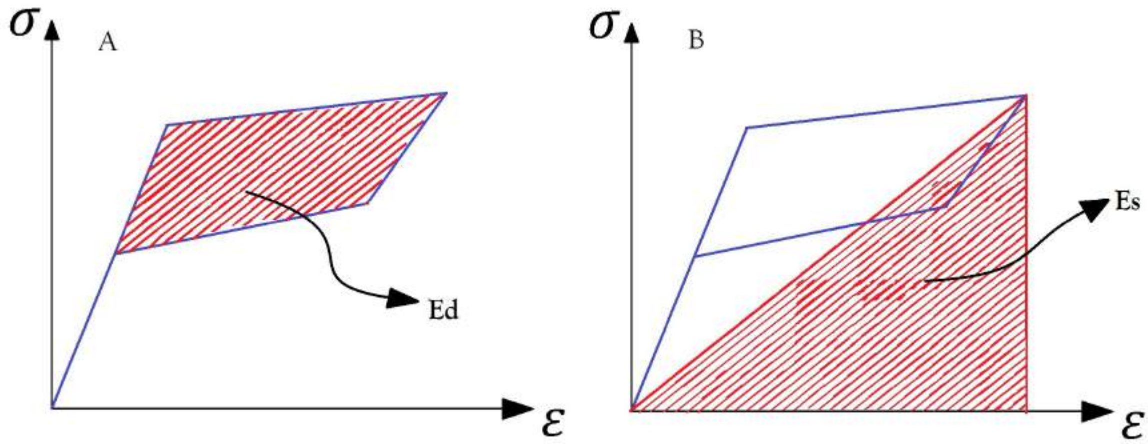

Figure 6. Curve areas for calculation of specific damping capacity.

The Equation (1) is calculated using the Stress-Strain curve, in the case of this paper were used the force-displacement curves, however to obtain the equivalent damping factor using this curve is necessary to divide the areas Ed, area of the curve, and Es, the area of the crossed triangle by the volume of the damper.

The superelastic SMA data used was for an alloy of NiTi as show in Otsuka e Wayman (1999), where was created a new material in *Ansys Workbench 17* and applied all the theoretical data, Table 2.

Table 2 - Superelastic SMA theoretical data.

Young's Modulus (Pa)	Poisson's Ratio	Bulk Modulus (Pa)	Shear Modulus (Pa)	Σ_{SAS} (Pa)	Σ_{FAS} (Pa)	Σ_{SSA} (Pa)	Σ_{FSA} (Pa)	ε (mm/mm)
6E10	0,33	5,88E10	2,26E10	4,5E8	5E8	3E8	2,5E8	7E-2

3. RESULTS AND DISCUSSSION

First was made some experimental tests with an unbalanced rotor, where was applied unbalanced masses (1 to 5 grams) in the disk of the system, Figure 7, and was obtained the maximum displacements in some frequencies, as showed in a Table 3.

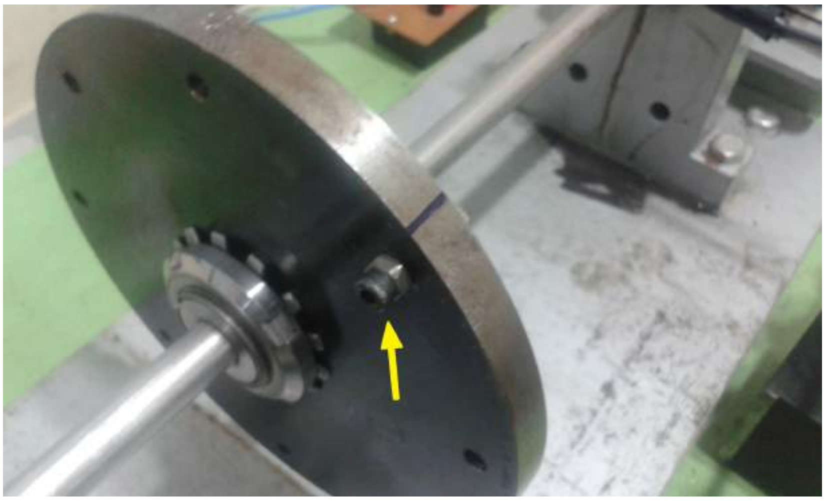

Figure 7. Disk with unbalanced mass

Table 3. Maximum displacements with unbalanced mass.

Axis	X (mm)						Y (mm)				
Freq. \ Mass	1g	2g	3g	4g	5g		1g	2g	3g	4g	5g
25 (Hz)	0.23	0.28	0.37	0.46	0.55		0.15	0.17	0.13	0.21	0.26
28 (Hz)	0.58	0.94	1.25	1.56	2.00		0.22	0.31	0.18	0.32	0.45
29 (Hz)	1.24	1.47	1.71	1.98	2.35		0.19	0.27	0.19	0.33	0.41
30 (Hz)	1.20	1.5	1.7	1.79	1.96		0.19	0.25	0.16	0.39	0.51
31 (Hz)	1.08	1.08	1.24	0.96	1.29		0.29	0.38	0.37	0.52	0.76
32 (Hz)	0.77	0.96	0.96	1.00	1.04		0.42	0.6	0.55	0.74	1.01
33 (Hz)	0.68	0.78	0.73	0.8	0.88		0.72	0.77	0.63	0.86	1.26
34 (Hz)	0.61	0.7	0.68	0.76	0.79		0.89	1.00	0.89	1.24	1.54
35 (Hz)	0.58	0.61	0.62	0.76	0.94		1.13	1.19	1.2	1.53	1.83

With the maximum displacement of 2.35 mm in 29 Hz supported by the system, for simulation was applied a maximum displacement of 2.5 mm on the SMA damper springs, it was verified the martensite transformation fraction, the principal maximum stress and the force – displacement behavior. Figure 8 shows how much of the material has reached the martensitic transformation for both springs. It was noted that for the convex sheet spring only 32% of the transformation was reached, while concave irregular sheet spring reached 100% of the transformation in the selected areas.

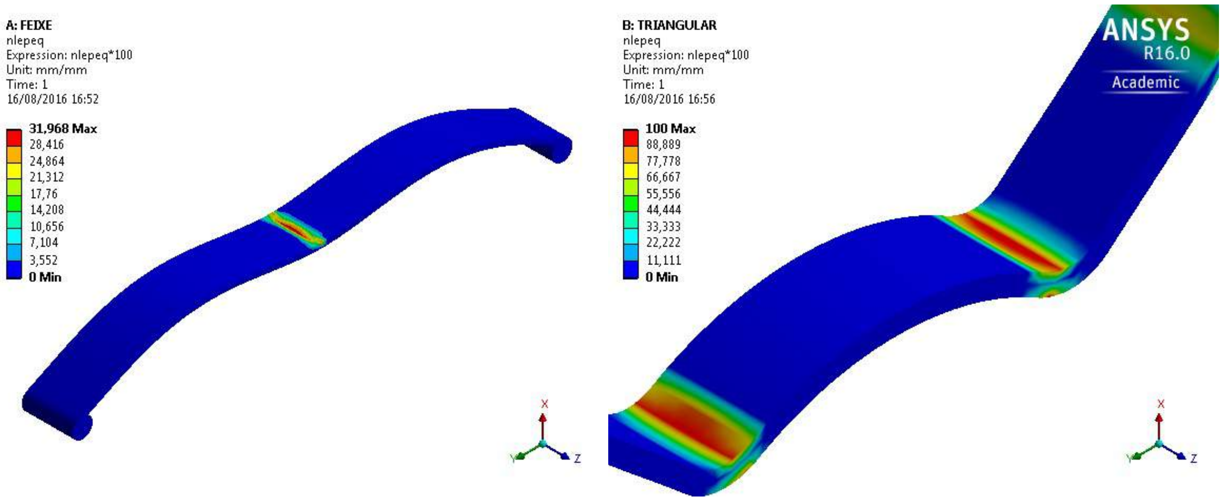

Figure 8. Martensite transformation: (A) Convex sheet spring and (B) Concave irregular sheet spring.

The maximum principal stress in each spring can be verified in Figure 9.

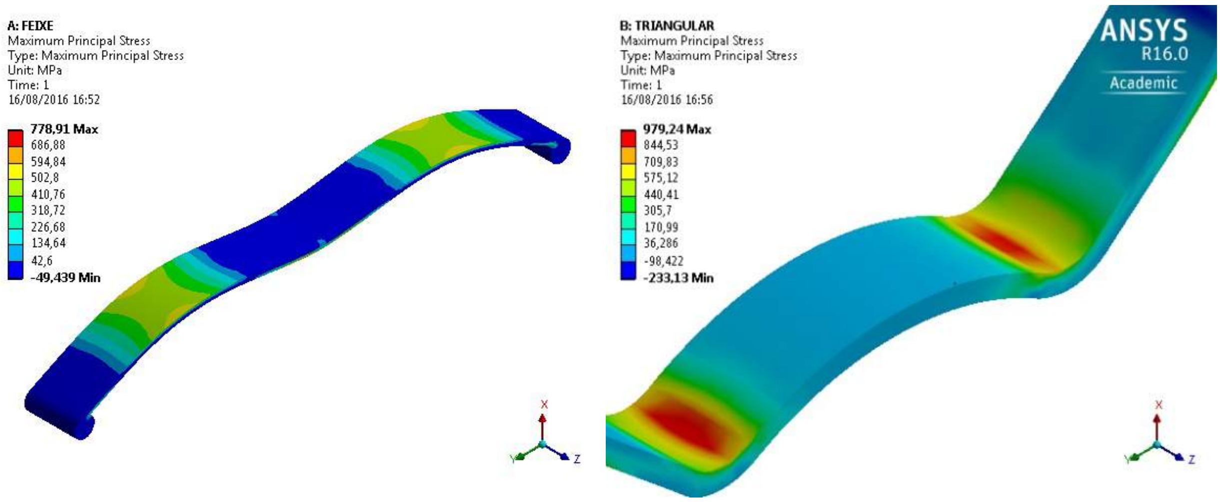

Figure 9. Maximum stress: (A) Convex sheet spring and (B) Concave irregular sheet spring.

How the SMA material can supports 1200 MPa in the maximum principal stress before suffer the plastic strain, both dampers should be under this limit. In this case both devices supports this applied displacement without undergoes to plastic strain, but the convex sheet spring can support more displacement than concave irregular sheet spring.

Figure 10 shows the force – displacement behavior in the middle point of each SMA spring damper, where is noted the difference in the hysteresis zone between both dampers, being the concave irregular sheet spring larger than the convex sheet spring.

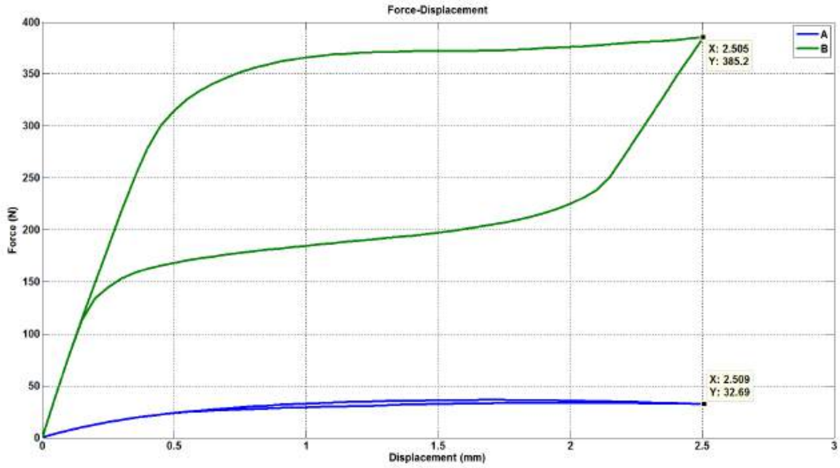

Figure 10. Force – displacement: (A) Convex sheet spring and (B) Concave irregular sheet spring.

For a complete analysis was obtained the effective stiffness and the specific damping capacity as showed in Table 4.

Table 4. Effective stiffness and Specific damping capacity

Damper	Volume (mm³)	Effective Stiffness	Specific Damping Capacity (%)
Convex Sheet Spring	152.172	13.09	0.94
Concave irregular sheet spring	152.372	153.77	5.40

Analyzing the Figure 10 and Table 4 is notorious the difference between dampers, where the concave irregular sheet spring has a bigger specific damping capacity than the convex sheet spring, more than five times, that means this type of design can mitigate the vibration with hysteresis for this displacement applied. The effective stiffness of the concave irregular sheet spring is also bigger than the convex sheet spring, where it means to disturb this type is needed more force applied, as noted in Figure 10.

4. CONCLUSION

The superelastic SMA concave irregular sheet spring due the larger contact area with the bearing of the rotor system, leading to a better distribution of the vibration force, it makes the material reaches the hysteresis zone necessary to dissipate energy and control the vibration, as showed in the numerical analysis and comparisons with the convex sheet spring. The hysteresis of the concave irregular sheet spring is larger than convex sheet spring, making possible attenuate more vibration of the rotor system with less disturb. Although this type also has a bigger effective stiffness, in this case to disturb this damper needs more force than the convex sheet spring.

This happens because the thickness of the material, in this case with less thickness will decrease the force to disturb the device for the same displacement applied. Therefore the concave irregular sheet spring, even being stiffer than the other type it has a biggest area of hysteresis, is a better choice of design to dissipate the vibration provided by rotor systems.

The numerical analysis made are important due to made a better chose on the device which will be applied in rotor system to mitigate the mechanical vibration with the damping and to decide the better dimensions of this damper before to manufacture it and made experimental tests.

5. ACKNOWLEDGEMENTS

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