VARIABILITY OF THE TRANSMITTED AXIAL FORCE BY THE HELICAL SPRINGS IN THE SUSPENSION OF HERMETIC COMPRESSORS

Ricardo Mikio Doi, rmdoi@lva.ufsc.br
Arcanjo Lenzi, arcanjo@lva.ufsc.br
Federal University of Santa Catarina
Department of Mechanical Engineering
Acoustic and Vibration Laboratory
C.P. 476
88040 420 – Florianópolis-SC, Brazil

Edmar Baars, Edmar_baars@embraco.br
Empresa Brasileira de Compressores – Embraco S/A
Acoustic Laboratory
Rua Rui Barbosa 1020
88219 901 – Joinville-SC, Brazil

Abstract. *This paper aims at investigating the variables that influence the axial dynamic force transmitted by the suspension springs to the hermetic compressor. The objects of the present study are cylindrical helical spring whose helix pitch angle varies at both ends and are constant at the intermediate region. The geometry variation and the assembly of the springs in the spring stop were evaluated within a frequency 0 to 10000 Hz. The first step consisted of evaluating the geometry variation of the springs. The spring geometries were obtained with sequences of images of the springs rotating around a central axis. Some spring parameters were obtained from those pixels images. In the second step, considering the input acceleration and the transmitted force, an apparatus was built for obtaining the axial dynamic behavior of the spring. The results showed that while the variation of the geometry is small, the assembly of the spring on the spring stop has a relevant influence on the variability of the transmitted axial dynamic force.*

Keywords*: cylindrical helical spring, suspension, variability, transmitted force.*

1. INTRODUCTION

The helical springs are components that have a wide application and great relevance in engineering. In the hermetic compressors, the helical springs support the compressor and provide isolation of vibration from compressor to the hermetic shell. A good spring performance depends on dynamic characteristics which are not easy to determine. In fact, springs are nonlinear mechanical components and effects of their extension deformations and torsional are coupled (Jiang et al, 1991).

The helical springs have been a active objective of the research for a long time, the contributions of the springs about investigation consisted of improvements in the formulations of motion equations, and on the analytical and numeric solutions in the search of those equations. Witrick (1966) obtained a group of differential equations to describe the spring motion, considering a spring element as a Timoshenko beam with six degrees of freedom. Mottershead (1980) used the Witrick differential equations in order to determine the natural frequencies and mode shapes, by using Finite Elements Method, and compared the results of this study with obtained experimental data.

The understanding of the dynamic behavior of spring addressed some investigation about the influence of the geometric parameter on the dynamic behavior of the helical springs. Pearson (1986) discussed the influence of the ends of the helical springs and the number of active turns on the natural frequencies. Yildirim (1999) obtained the natural frequencies for free vibrations by varying the helical turn numbers, the helix pitch angle, the shape cross-section and the ratio of the diameter of the cylinder to the wire.

Despite the great amount of work on helical springs found in the literature, few studies involving experiments to determine the dynamic behavior of springs has been published. Della Pietra (1976); Guido et al., (1978), Della Pietra & Della Valle (1982) have used an experimental setup to visualize the first mode shapes and compared the findings with analytical previous model results. Pearson (1986) compared analytical results with experimental data. Two methods were used in the experiment, one of the transient vibration and other of the natural frequencies excitation, both methods use transducers for observing the natural frequencies.

As regards helical springs used as suspension in the hermetic compressors, the studies have been carried out in medium frequency range, because in this frequency high transmissibility of the dynamic force can excite the compressor shell and consequently irradiate high noise level. Kjeldsen and Madsen (1978) studied the optimality of the place of the springs in the hermetic compressors, in order to minimize the transmitted forces by motion of rigid body. Carmo (2001) validated a finite elements model of helical spring, constituted of beam elements using experimental data obtained in frequency until 5000 Hz.

This paper will describe the results of studies in the axial dynamic forces transmitted by helical springs used in a prototype of hermetic compressor. In the first step, the geometric differences of the springs were evaluated by using a method of sequential images. In the second step, the axial dynamic behavior of the springs was evaluated through the axial forces transmitted in the frequency range 0 to 10000 Hz and differences on the average dynamic behavior were associated with differences on the spring geometries. In order to represent the real conditions of the springs in the spring stop, an experimental setup was designed, which will be next presented.

2. GEOMETRY OF THE HELICAL SPRINGS

The length of the helical springs are about 29.1 mm, the diameters of the cylinder is 13.6 mm and the wire diameter measures 1.4 mm. The springs have 5 ¾ of active turns and 2 inactive turns in both ends. Fig. (1-a) shows the assembly springs in the spring stops of a hermetic compressor prototype and Fig. (1-b) displays the drawing of the springs.

Figure 1-a. Cylindrical helical suspension springs in the compressor.

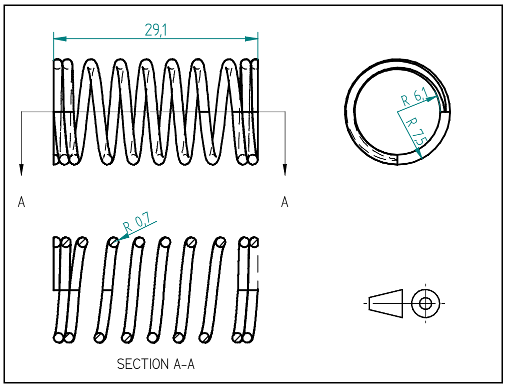
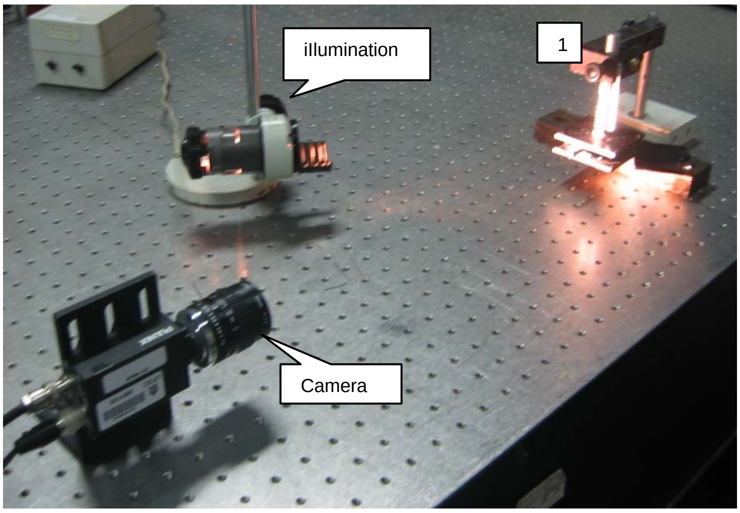
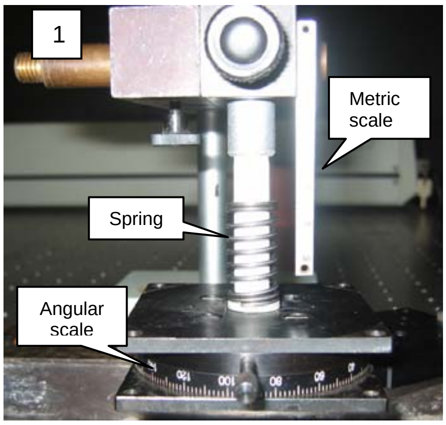


Figure 1-b. Dimensions [mm] of the spring.

The geometry of the helical spring can show some variability according to the fabrication process. The experimental obtention of the spring geometry, i. e., by optic method, is not an easy task. Sometimes the obtention of the geometry is not possible due to geometry complexity and equipment limitations.

In order to verify the geometry variation, the three springs in the prototype of the compressor were analyzed. The geometries were obtained by sequential images. A base with angular scale was used to hold the spring through a cylindrical axis, the latter passing inside the coils of the spring. A metric scale was adapted in the same front plan of the spring. A camera (Pulnix Model TM-75) was positioned 2 m from the spring in order to obtain the images. Fig. (2) presents the setup used to obtain the images of the springs.



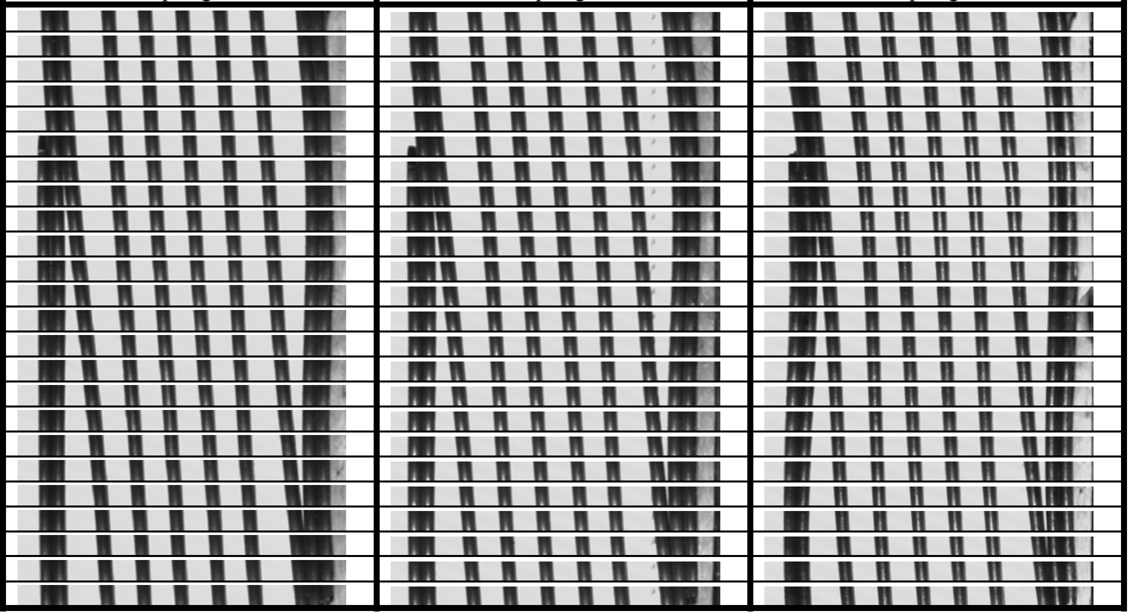


Spring holdin base.

Figure 2 -Setup for obtaining of the springs image.

An image of the spring was obtained at every 15 degrees. The front area equivalent to 15 degrees of each images was cut out and put on sequential order to form a plan image of the spring. The distortion of the images was small because the slice of the spring was small. Tab. (1) displays the sequential images of the three springs.

Table 1. Plan images of the springs.



Angle	Spring 1	Spring 2	Spring 3
345°			
330°			
315°			
300°			
285°			
270°			
255°			
240°			
225°			
210°			
195°			
180°			
165°			
150°			
135°			
120°			
105°			
90°			
75°			
60°			
45°			
30°			
15°			
0°			

The images show the map of the wire along the coils of the springs. Furthermore, a relation between scale and pixels was obtained from the images with metric scale used in setup. This relationship, Eq. (1), can be used to quantify discrepancies of the geometries and other required information based on springs images

$$x = 6.7 \times 10^{-2} \times n \tag{1}$$

Where: x - length [mm];
n - number of the pixels in figure.

Irregularities are more observed in third spring, particularly notable in the inactive coils. In the intermediate area of the springs the helix pitch angles are constant for 4 coils, as can seen in Tab. (1). In order to calculate the helix angle pitch a straight line was interpolated by Minimum Square for each coil.

For each spring, the values of the helix pitch angles of the 4 central coils were used to calculate the arithmetic average. In spring 1, 2 and 3, these average were α_1= 4.49 degrees, α_2= 4.56 degrees and α_3= 4,50 respectively, and were displayed in Tab. (2).

Table 2. Helix pitch angles.

Coils	Helix pitch angles - spring 1	Helix pitch angles - spring 2	Helix pitch angles - spring 3
1	4.68 °	4.74 °	4.55 °
2	4.38 °	4.51 °	4.66 °
3	4.43 °	4.40 °	4.76 °
4	4.47 °	4.38 °	4.26 °
Average values	4.49 °	4.51 °	4.56 °

As displayed in Tab. (2), the 3 springs have the same number of the active coils. This can be verified by observing the initial point of the inactive coils. The diameter of the coils and the diameter of the wire can be obtained through direct measurements.

3. AXIAL DYNAMIC BEHAVIOR OF THE SPRINGS

3.1. Experimental Methodology

The spring dynamic behavior above 5000 Hz is not an easy task to obtain. Previous studies on these springs concluded that the contact effect between the areas of the inactive coils and the spring stops influences the dynamic behavior of the spring. An apparatus was designed in order to identify the axial dynamic characteristics of the assembly springs and of the spring stops in the real conditions, and within the frequency of 0 to 10000 Hz.

Initially with the use of the commercial software Ansys, a shape for a steel block was defined with an equivalent mass of 1,6 kg. Which was defined based on the mass of the compressor prototype. The format of the block was designed to have natural frequencies above the frequency range of the analysis. The same spring stops used in the compressor were adapted in the assembly of the apparatus to reproduce more realistic conditions of the spring ends in the spring stops. The assembly consisted of coupling the springs between the inertial mass and the equivalent mass, Fig. (3).

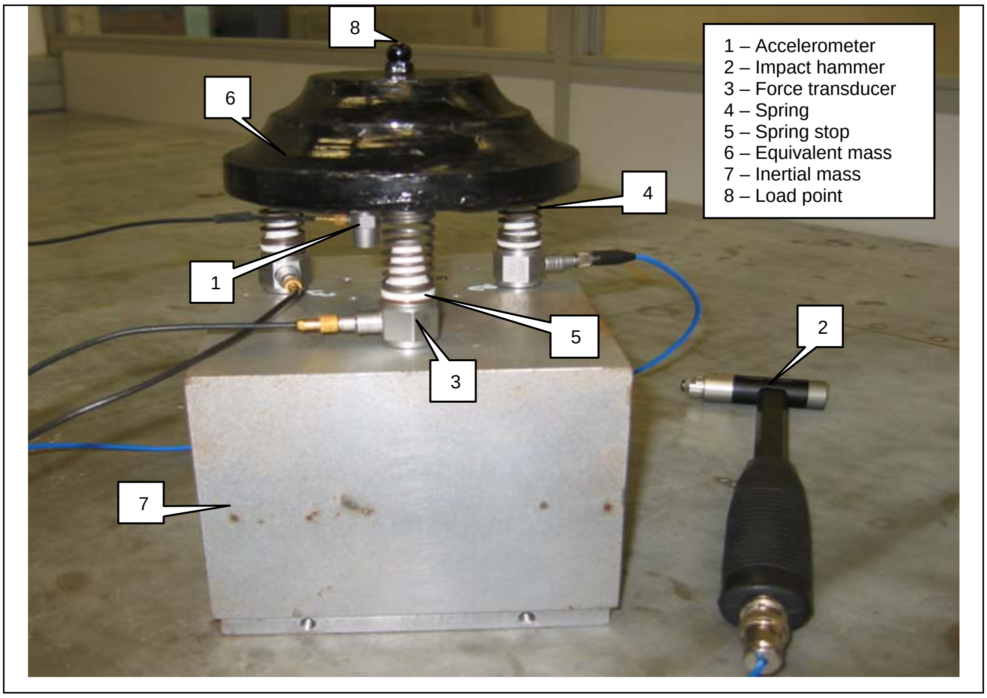


Figure 3. Aparatus to obtain the springs transmissibility.

Three uniaxial force transducers (PCB Piezotronics 208C022) were localized between the inertial mass and the spring stops to measure the force transmitted by the springs. The load was applied with an impact hammer (PCB Piezotronics 086C01) in the central load point of the equivalent mass. Here it is convenient to introduce the concept of transmissibility (τ) of coupled systems (Douville et al., 2005), as defined in:

$$\tau = \frac{F_T}{F_{Exti}} \tag{2}$$

where, F_T is the transmited force and F_{Ext} are excitation force.

The linearity of the spring was tested through two ways: the analysis of the reciprocity and the variation of the amplitude of the input force. The tests did not show severity of nonlinearity in the system. This setup can be considered similar to a vibrometer mechanism. The first natural frequency of vertical translation of the system is 10 Hz. It was assumed that the force weight of the equivalent mass and the impact force were equally distributed on the 3 springs. Then the transmissibility of one spring in this system can be obtained by the Eq. (3). A signal analyzer (LMS Scadas III) was used for the acquisition and the signals processing. The signals were measured in the frequency range from 0 to 10000 Hz using 12800 sampling frequencies.

$$\tau_M = 3 \times \frac{F_{TR}}{F_{Ham}} \tag{3}$$

where, F_{TR} are the transmited force by spring and F_{Ham} are hammer force, both measured.

3.2. Results

In the system described in the previous session, the measurements were obtained individually for the three springs. For each spring, 30 transmissibilities were obtained, 1 for each assembly. The assembly consisted of removing the springs out of the setup and put them on the spring stops again for each transmissibility measured. In order to measure the transmissibility always in the same position of the equivalent mass, the three springs exchanged locations every 30 measurements (Box et al., 1978). The quality of the force transmited spectrum was also monitored for each measurement.

In order to measure the axial dynamic behavior of the spring the input force was applied in the vertical direction. The dispersion of the natural frequencies increased in high frequencies. Fig. (4), (5) and (6) present the transmissibilities obtained for the 3 springs and the average transmissibilities, which were obtained by arithmetic average from the 30 values of transmissibilites of every frequency line.

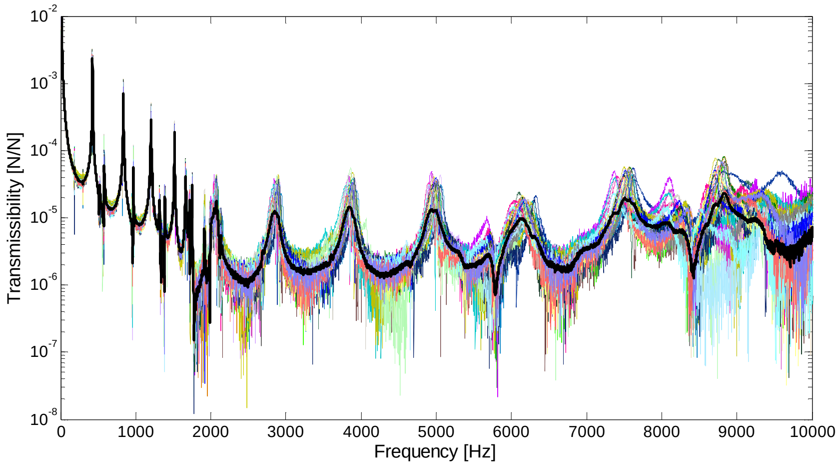


Figure 4. Transmissibilities obtained for spring 1 and the average of transmissibilities (black line).

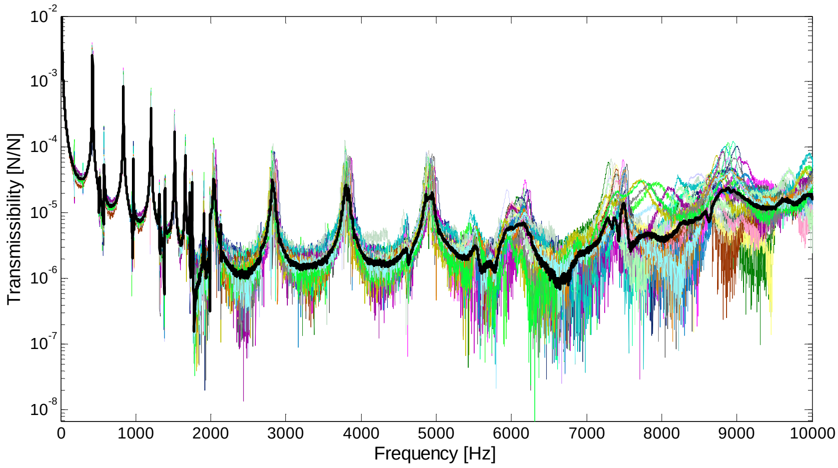


Figure 5. Transmissibilities obtained for spring 2 and the average of transmissibilities (black line).

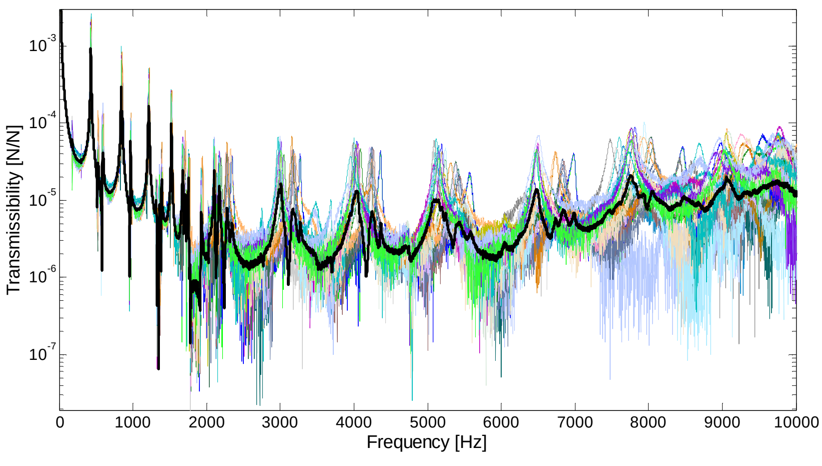


Figure 6. Transmissibilities obtained for spring 3 and the average of transmissibilities (black line).

Natural frequencies were identified for each transmissibility measured. The natural frequency average and the standard deviation were obtained for each resonance, Tab(3). The average was calculated from the 30 values identified for each resonance. Small dispersion was verified in the first 14 natural frequencies of the 3 springs. However, more dispersions are observed in the third spring.

Table 3. Average and standard deviation of natural frequencies for each spring.

Mode shape	Spring 1		Spring 2		Spring 3	
	Natural frequency [Hz]	Standard deviation[Hz]	Natural frequency [Hz]	Standard deviation[Hz]	Natural frequency [Hz]	Standard deviation[Hz]
1	424.45	0.43	424.48	0.96	429.09	2.47
2	517.20	0.69	517.00	0.65	522.40	2.68
3	579.41	1.47	577.79	1.87	591.12	4.15
4	834.98	0.82	835.41	1.03	844.79	4.92
5	968.42	0.70	967.87	0.86	969.13	2.74
6	1202.90	1.25	1202.94	1.59	1214.78	4.22
7	1320.05	1.19	1316.86	1.38	1319.27	3.29
8	1387.95	0.79	1388.49	1.31	1388.96	2.69
9	1516.61	1.05	1518.71	1.34	1524.10	10.09
10	1664.70	1.60	1660.57	1.76	1685.84	13.09
11	1716.12	0.89	1719.01	9.97	1721.28	5.12
12	1729.69	0.92	1732.16	0.96	1754.90	4.86
13	1751.60	0.75	1752.42	0.79	1933.20	5.90
14	1915.55	2.87	1908.03	2.78	2107.05	8.13
15	2067.10	13.28	2039.05	8.41	2212.23	64.35
16	2876.07	28.35	2830.36	25.26	3069.27	98.30
17	3868.43	36.05	3813.28	33.67	4104.27	123.29
18	4988.23	44.14	4911.62	56.74	5258.82	153.68
19	6173.79	83.53	6108.32	95.47	6551.40	317.30
20	7527.03	83.88	7440.01	92.31	7967.82	317.22
21	8757.30	107.09	8774.19	163.59	9257.96	395.06

4. CONCLUSION

In this article, a methodology used to evaluate the geometric discrepancies of helical cylindrical springs through sequential images was presented. An experimental apparatus was designed in order to obtain the axial transmissibility of the helical springs, considering the influence of the assembly of the springs on the spring stops.

The plan images of the spring allowed the identification of irregular regions and some geometric parameters, such as helix angle pitch and the number of the active coils in degrees. The intermediate region of the three springs presented small variation in the helix angles pitch and the spring 3 presented larger irregularity of the geometry in the inactive coils regions.

The axial transmissibility of the helical springs was measured within the frequency range of 0 to 10000 Hz with satisfactory quality. The variability of the dynamic characteristics of the assembly springs in the spring stops was determined and the natural frequencies presented considerable dispersion in the assembly of the third spring. The natural frequencies of the first spring were close to the frequencies of the second spring. However, the natural frequencies of the third spring presented a considerable difference when compared with the frequencies of the first and second springs. Variability was more observed in the transmissibility of the third spring, probably due to the larger irregularity in its geometry.

5. ACKNOWLEDGEMENT

This work was supported by CNPq (Conselho Nacional de Desenvolvimento Científico e Tecnológico) and Embraco (Empresa Brasileira de Compressores). The authors wish to thank them for the support and Prof. Armando Albertazzi Gonçalves for the valuable suggestions.

6. REFERENCES

Box, G. E., Hunter, W. G., Hunter, J. S., 1978, Statistics for Experiments, Ed. John Wiley & Sons, New York, 653p.

Carmo, M. G. V., 2001, "Fluxo de Energia Vibratória do Conjunto Moto-Compressor para a Carcaça de um Compressor Hermético Através das Molas de Suspensão", Dissertação de Mestrado, Universidade Federal de Santa Catarina, Departamento de Engenharia Mecânica, Florianópolis-SC, Brasil.

Della Pietra, L., 1976, "The Dynamic Coupling of Torsional and Flexural Strain in Cylindrical Helical Springs", Meccanica, Vol. 11, pp. 102-119.

Della Pietra, L., Della Valle, S., 1982, "On the Dynamic Behaviour of Axially Excited Helical Spring", Meccanica, Vol. 13, pp.31-34.

Douville, H., Masson, P. Berry, A., 2006, "On-ressonance Transmissibility Methodology for Quantifying the Structure-borne Road Noise of an Automotive Suspension Assembly", Applied Acoustics, Vol. 67, pp. 358-382.

Guido, A.R, Della Pietra, L., Della Valle, S., 1978, "Transverse Vibration of Cylindrical Helical Springs", Meccanica, Vol. 13, pp. 90-108.

Kjeldsen, K. Madsen, P., 1978, "Proceeding of International Compressor Conference at Purdue", pp. 55-59.

Mottershead, J. E., 1980, "Finite Elements for Dynamical Analysis of Helical Rods", International Journal of Mechanical Sciences, Vol. 22, pp. 167-283.

Pearson, D., 1986, "Modelling the Ends of Compression Helical Springs for Vibration Calculations", Proceedings of the Institution of Mechanical Engineers, Vol. 200, No. C1.

Wittrick, W. H., 1966, "On Elastic Wave Propagation in Helical Springs", International Journal of Mechanical Sciences, Vol. 8, pp. 25-17.

Yildirim, V., 1999, "An Efficient Numerical Method for Predicting the Natural Frequencies of Cylindrical Helical Spring", International Journal of Mechanical Sciences, Vol., 41, pp. 919-939.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.