FLEXIBILITY INFLUENCE ON THE TRANSMITTED LOAD IN A SHAFT-ROTOR SYSTEM

Érico Romera Fulco
Federal University of Santa Catarina - UFSC
Department of Mechanical Engineering
Acoustics and Vibration Laboratory
C.P. 476
88040-900 - Florianópolis-SC, Brazil
e-mail: ericorf@lva.ufsc.br

Arcanjo Lenzi
Federal University of Santa Catarina - UFSC
Department of Mechanical Engineering
Acoustics and Vibration Laboratory
C.P. 476
88040-900 - Florianópolis-SC, Brazil
e-mail: arcanjo@lva.ufsc.br

Edmar Baars
Empresa Brasileira de Compressores – Embraco S/A
Acoustics Laboratory
Rua Rui Barbosa 1020
88219-901 – Joinvile-SC, Brazil
e-mail: edmar_baar@embraco.com.br

Abstract. *Due to practical issues dynamic mechanisms, like rotors, are usually represented by rigid structures, with no consideration of the flexibility properties. These systems are submitted to transient loads, making necessary a high discretization in the time domain to obtain a refined representation. Therefore, one of the challenges is to develop an algorithm capable of solving this problem in acceptable time scale. The solution of this issue can be found by using the finite element software ANSYS with a transient analysis by modal superposition method. The compressor shaft-rotor system is a mechanism with three acting loads, one comes from the electricity induction motor, which produces the spinning of the shaft, another one comes from the compression piston, which is applied to the upper shaft. The third load comes from the block, holding the shaft inside the structure. All the mechanisms connected to the shaft are considered rigid, except the block contact bearings, where contact due to the oil film is modeled as a spring. More than one type of springs are used for a rigid and flexible shaft-rotor. The results light on the flexibility importance of the transmitted load to the block, hence show that flexibility can't be neglected on this type of system.*

Keywords: Flexibility, Transmitted Load, Shaft-Rotor, FEM

1. INTRODUCTION

In modern times, the industry has been worrying more and more on reliability of its products. For refrigerators factories, where the products play a big competition for the consumers, the functionalities of compressors must be fully optimized. It is also important to predict the bad functioning of the product.

The compressor piece studied here is the shaft-rotor system shown in Fig. 1. It has three parts, one motor rotor, generating the system movement, one shaft connected to the rotor, and one upper shaft. There are two bearings in the shaft doing the contact with the compressor block due to oil film lubrication. The upper shaft applies a load to the piston making the air compression in the compression chamber. This system is under heavy loads and these loads are transmitted to the block by the contact oil film.

The pressure in the compression chamber is transmitted by the piston mechanism to the upper shaft. This pressure has the main influence on the load transmitted to the block, showing us the curve characteristics. The path of this pressure to the upper shaft is modeled analytically with some important assumptions discussed in section 2.

The shaft-rotor structure is modeled by Finite Elements and validated with simple experiments of modal analysis. The program used to do this task and build the procedure of dynamics analysis is ANSYS. Some important assumptions are made at this point, being further discussed at the section 3.

The results of this procedure are discussed in section 4 and the conclusions in section 5.

2. DYNAMIC EQUATIONS

The pressure load in the compression chamber is transmitted to the piston, than to the connecting rod that makes the contact with the upper shaft. The piston and the connecting rod are considered rigid bodies and their contacts are without gaps. The block is also considered a rigid body and it is impossible to move it.

The pressure was acquired in a controlled experiment with some special conditions not relevant to the article.

One of the most important assumptions is that the angular velocity of the shaft is considered constant, giving us a simple way to build the dynamic equations of piston-connecting rod movement as well as to the numerical analysis.

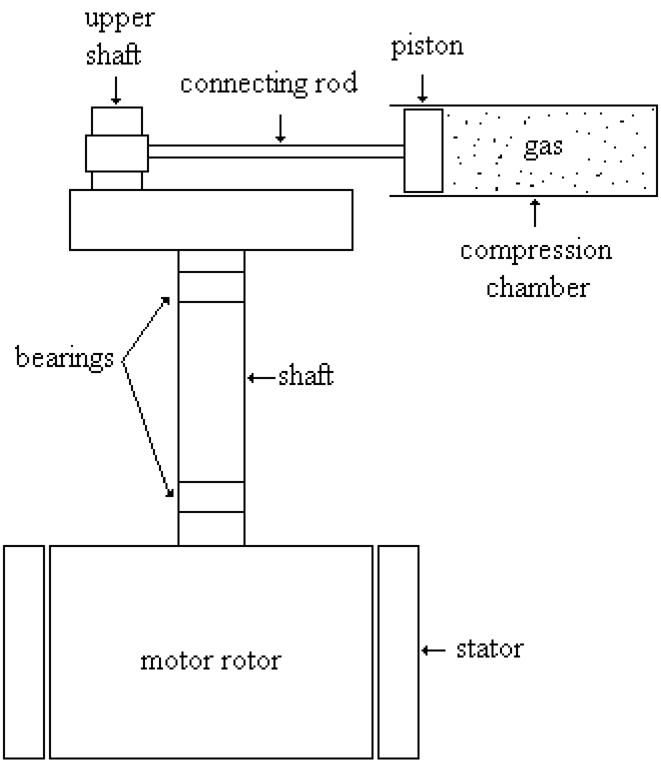

Figure 1. Parts of the system

The equation of the transmitted load by the gas compression, movement of the piston and connecting rod to the upper shaft is Eq. 1 and Eq. 2. as developed by Rodrigues, R.S. Where Fb_i and Fb_j are the transversal and axial loads transmitted to the upper shaft respectively, *P* is the pressure in the compression chamber, *A* is the piston area in contact with the gas, *mb* and *mp* are the connecting rod and piston mass respectively, $\ddot{y}$ is the acceleration of the piston, $\dot{W}$ and $\ddot{W}$ are respectively the angular velocity and angular acceleration of the connecting rod and Lb_1 is the distance between the piston and the mass center of the connecting rod.

$$Fb_i = P \cdot A + mb \cdot \dot{W}^2 \cdot Lb_1 - (mp + mb)\ddot{y} \tag{1}$$

$$Fb_j = mb \cdot Lb_1^2 \ddot{W} \tag{2}$$

The product $P \cdot A$ it is the resultant load over the piston due to the pressure in the compression chamber. This pressure is not constant in space in the chamber. Therefore, this pressure is considered constant in space.

3. THE FINIT ELEMENT MODEL

The shaft rotor structure is modeled in finite element and validated with a modal analysis experiment, with the results shown in Tab. 1. Structural damping alfa and beta were also obtained by this procedure. The kind of element used to modeling the structure is as simple as possible, making the program faster. The chosen element was a linear tetrahedral element with 4 nodes. These subjects are well explained in Cook reference.

ANSYS software was used to do all FEM work. The FEM mesh was constructed with the fewer possible number of elements making all finite element analysis faster. Another important issue was establishing the kind of transient method used on ANSYS to solve the dynamics. It was offered three methods: the full method, solving the movement equations completely; the Guyan reduction method, reducing the problem to few degrees of freedom; and the modal superposition analysis, combining the main structural modes to solve the problem.

It is needed the faster solver because this problem require too much time and space discretization. Table 2 shows a comparison of the three methods. The faster one was the modal superposition analysis being chosen to solve this procedure. The time needed to obtain the result is more than one day using the superposition method with a 3.2 GHz, 2 Gb of RAM memory and well suited computer. It is estimated about one year to do the same job with the full method.

The experimental modal analysis was done with the signal analyzer SCADAS, using one of its software called modal impact. This experiment consists on hit the shaft with a hammer on many places and analyse the structure vibrantion. The SCADAS software releases the vibration modes and the structural damping. The frequency range used on this experiment was from 0 Hz to 8000 Hz because there is no significant influence of shaft vibration after this. The importance of vibration modes was achieved testing the numeric procedure.

The structural damping used on ANSYS model was an alpha and beta damping and the structure material was considered composed by only one isotropic material. Those assumptions were made to simplify and increase the procedure speed. Harder assumptions can be made but those are of enough accuracy to this problem.

The numeric modal analysis chosen was the block of Lanczos method and the result is shown on Tab. 1.

Table 1 - Experimental Modal Analysis versus Numerical Modal Analysis.

Modes	Experimental (Hz)	Numeric (Hz)	Error (%)
1	1224	1215	0.7
2	1268	1223	3.5
3	3320	3139	5.5
4	5110	5188	1.5
5	6086	6108	0.4

Table 2 - Comparison of full, Guyan reduction and modal superposition methods.

	Full	Guyan Reduction	Modal Superposition
Time[(1)]	1	5 - 150	150 - 500
Error[(2)]	0	0.6 - 6.5	0.3 - 1.5

[(1)] : times faster than full method
[(2)] : relative error to full method

The contact oil film is modeled as a spring in Eq. 3, where *F* is the load due to the spring load, *k* and *c* are arbitrary constants, *x* and *v* are, respectively, the relative displacement and relative velocity of the bearing, *n* and *m* are arbitrary positive numbers. There is no transverse response of the spring, only the one on radial direction.

$$F = -\left(kx^n + cv^m\right) \tag{3}$$

The number of time steps used is defined by the number of experimental points in the pressure curve acquired in compression chamber. Due to the few points in pressure acquisition, it is needed to interpolate the curve to create more points. It is used a linear interpolation. The pressure experiment was not a subject of this work so it is not commented here.

One important subject is how to know the time step needed for discretization. This is linked with the linear and angular accelerations of the shaft, because with a poor time discretization the shaft can became so fast that it will not pass by a good number of locations to well represent the load variations on the bearings. Than the system would become instable, leading the program to give bad results or make the shaft to transpose the block. This result was achieved by testing the system stability. Figure 2 shows how the instability affects the load transmitted by the bearings.

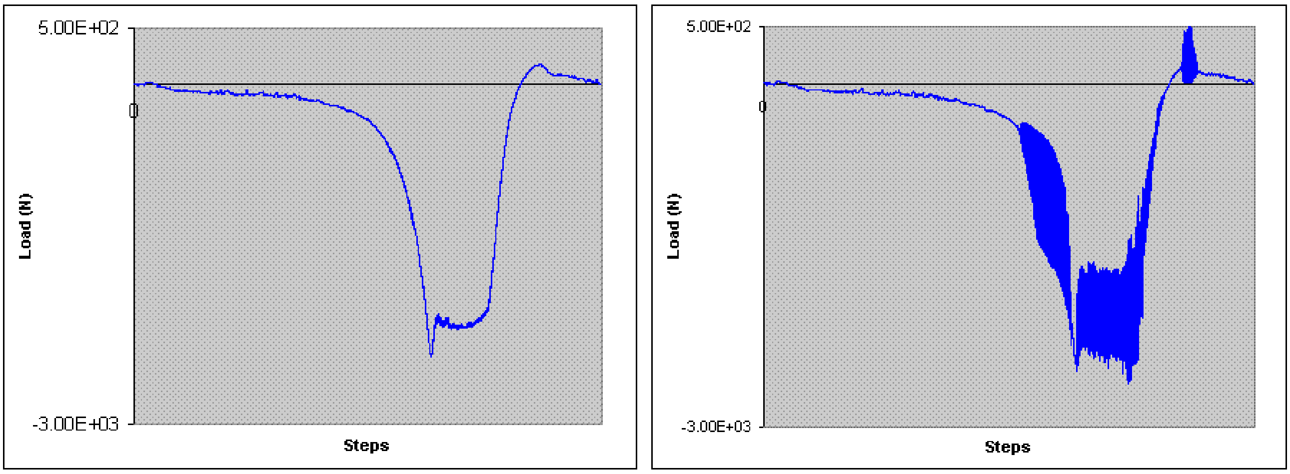

Figure 2. Two loads graphics of the upper bearing. The left system is stable and the right one instable.

An interesting matter is that the instability shown in the load graphic do not appears in a significant scale on the displacement graphic. Figure 3 presents the displacement graphic of the unstable system (right on Fig. 2). This illustrates the importance of load analysis to detect instabilities.

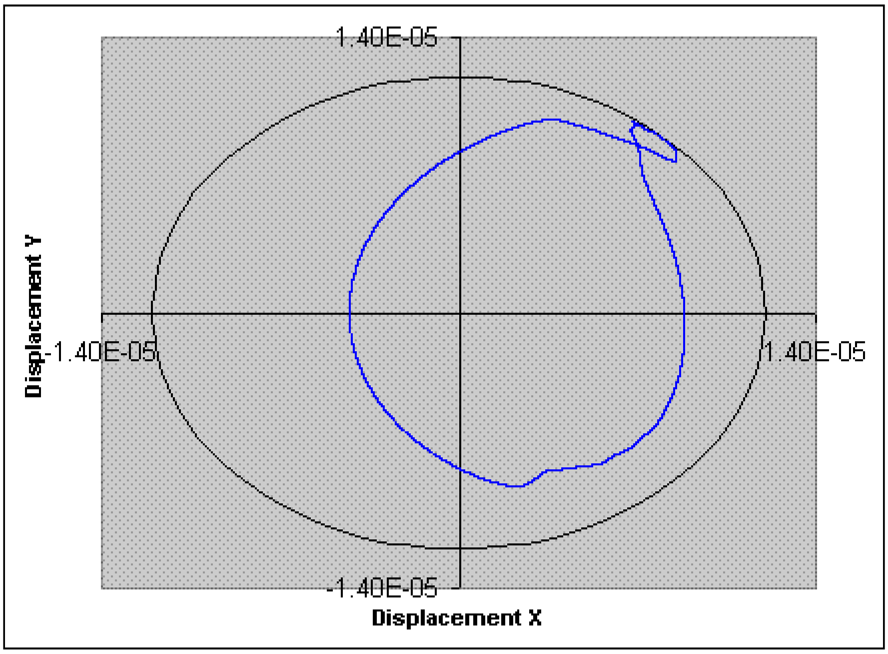

Figure 3. Displacement on both Cartesian directions.
This orbit is related to the same case of Fig. 2 on the right side.

The load over the rotor that impulses the system recalculated in every time step doing the angular velocity of the shaft to be always constant.

Velocity of bearings v are calculated in every time step dt, by the present displacement x_1 and three lasts steps, x_2, x_3 and x_4 as shown in Eq. 4.

$$v = (11 \cdot x_1 - 18 \cdot x_2 + 9 \cdot x_3 - 2 \cdot x_4) \cdot (6 \cdot dt)^{-1} \tag{4}$$

There is a radial distribution of the loads over the bearings as well as a height distribution of these loads to better simulate the contact reality.

As said before, the rotational velocity is constant in time, being used all inertial effects, except the one that deforms the structure due to this angular velocity.

4. RESULTS

The results are split in two sets. A first set about the rigid body analysis and a second one about the flexible body. All the oil film loads mentioned in the sequence of this section are about the shaft upper bearing.

4.1. Transmitted load with a rigid shaft

The first sets of results are about a rigid shaft-rotor. In Eq. 3, the values for *k*, *c*, was set as medium values obtained in Rodrigues reference and *n* and *m* was set in an arbitrary way.

Figure 4 and Fig. 5 show that the transmitted load at bearings has the same aspect than the incoming load over the upper shaft. Can be seen on Fig. 5 that the bearing response has a higher intensity but the same.

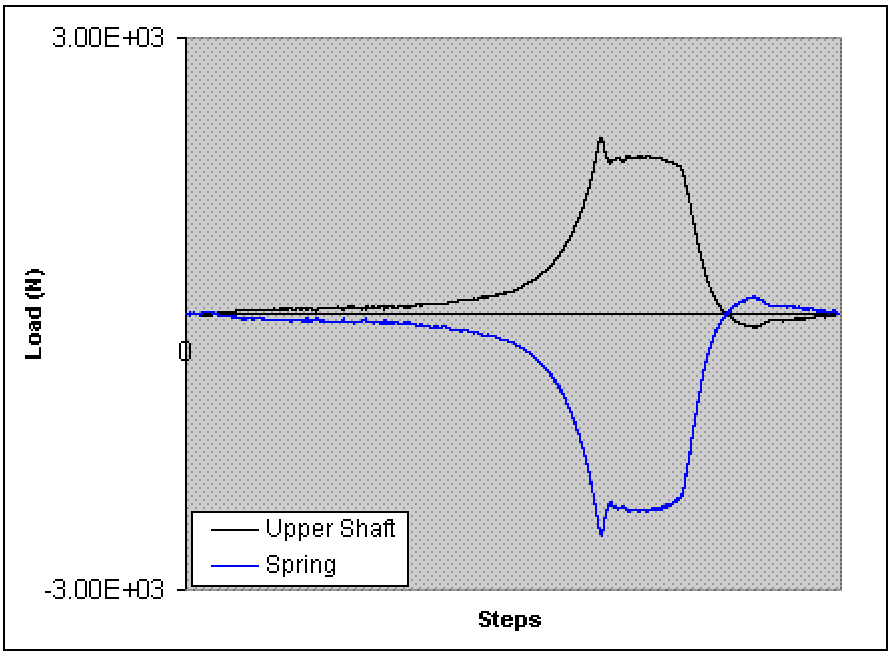

Figure 4. Comparison between upper shaft load and the oil film load.

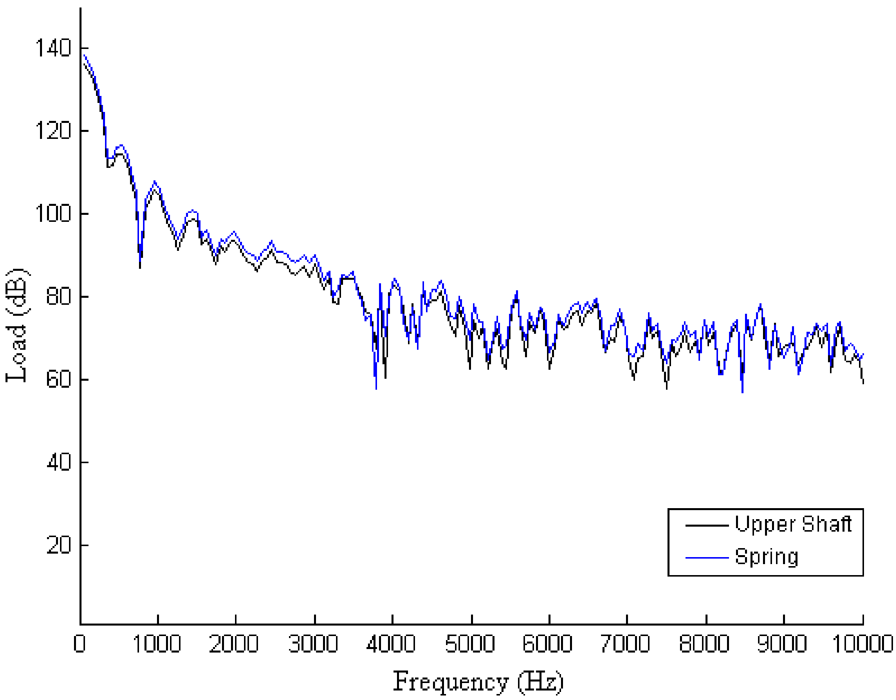

Figure 5. Spectrum of an usual spring bearing.

It is show in Fig. 5 that the bearing transmitted load has the same behavior of the upper shaft load just with a little increase in its magnitude.

4.2 Transmitted load with a flexible shaft.

Those numerical experiments below are related to other shaft and pressure conditions.

In Fig. 6 and Fig. 7 it is seen the high influences of the flexibility on the transmitted load. Some mode influences appear in the spectrum.

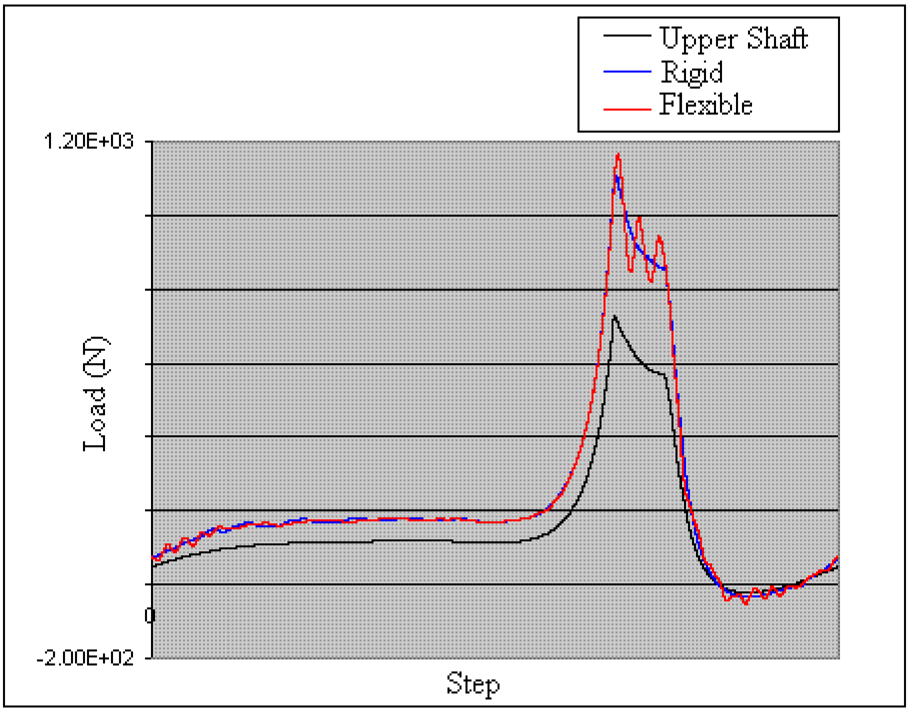

Figure 6. Rigid and Flexible Shaft considering $k = 1 \cdot 10^{8}$; $c = 1 \cdot 10^{5}$; $n = 5$; $m = 1$.

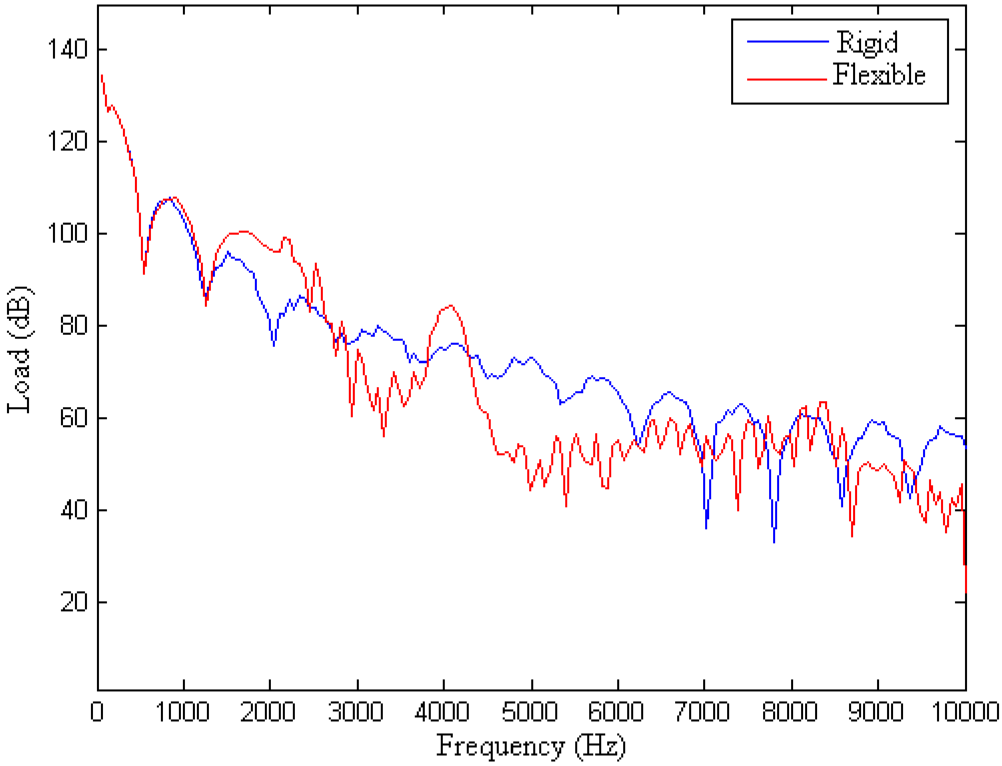

Figure 7. Load spectrum of Rigid and Flexible Shaft considering

Figure 7 shows the great influence of flexibility on the transmitted load. The spectrum has changed from the rigid to flexible body as well the transient load. Increases of more than 10 dB appear on spectrum of flexible body in some bandwidths. Those increases are related to the first 5 structural modes of this shaft.

This kind of effect can lead to a bad and not reliable system if the block has some resonance at those increments. So it is good to have some tool predicting the transmitted load to the block, giving a better chance to find or avoid problems in rotor systems.

5. CONCLUSIONS

A time increment numerical analysis has been presented, in order to investigate the influences of flexibility on the transmitted load by a shaft-rotor system to a block. This project have showed too the importance in using faster analysis methods like modal superposition, showing problems on using poor discretizations.

Given the much discretization needed, furthers reaserchs should build optimized algoritms, building the mesh structure simpler as possible, using powerful machines, making all possible to build a light procedure.

Other important issue is to verify carefully the procedure numerical convergence. It is of extreme importance to verify load behavior instead of the orbit behavior as show in section 3.

The main subject of this article was to study and to create a procedure to evaluate the flexibility influence on the transmitted load by the bearings. In our case, the flexibility shows up great influence at the load, growing it more than 10 dB in some bandwidth, and showing us the importance of flexibility in rotor mechanisms.

Those icreases appears on some of the vibration modes of the shaft. There was no need to evaluate experimentaly more than five modes of the shaft because only the firsts have some significantly influence on the transmitted load.

This kind of analysis can help in structure optimization, dynamic and flexibility analysis, load analysis and anything related to this. This work can help in further studies of rotor dynamics or of structures under variable loads.

6. ACKNOWLEDGEMENTS

A special thank to EMBRACO S/A partnership for the support, making this work research possible.

7. REFERENCES

Cook, R.D., Malkus, D.S. and Plesha, M.E., 1989, “Concepts and applications of finite element analysis”, Third Edition, Ed. John Wiley & Sons. Inc., United States of America, 623 p.
Rodrigues, R.S., 2003, “Análise dos esforços dinâmicos transmitidos pelo eixo e pistão ao bloco de compressores herméticos”, Florianópolis, Brazil,85 p.
Ferreira, E.M.S., 2005, “Modelo de rotores dinâmicos com mancais flexíveis utilizando material viscoelástico”, Curitiba, Brazil, 119p.
Silva, E.L., 2004, “Dinâmica de rotores: modelo matemático de mancais hidrodinâmicos”, Curitiba, Brazil, 110p.
Gerardin, R.C., 2005, “Modelo dinâmico do sistema pistão-biela-manivela com mancais hidrodinâmicos”, São Paulo, Brazil, 73p.