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CAMPBELL DIAGRAM CALCULATION DIDATIC SOFTWARE USING FINITE ELEMENTS METHOD

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Abstract. *The gyroscopic effect changes the natural frequencies of rotator machines, which induces the appearance of a pair of frequencies, called whirling frequencies. The highest whirling frequencies value of a pair presents a rotation in the same direction of the rotor and the other, the lowest one, rotates on the other direction. The occurrence of the highest or lowest frequencies depends on the rotation and on the initial condition. When the velocity of rotation has a whole multiple of the value of the whirling frequencies, there is an excessive vibration that may damages the machine. In the design of rotator machine, the operation closer to these points must be avoided. In this work, a software has been developed to determine the whirling frequencies using finite elements technique, considering the effects of the rotatory inertia and shaft-bending shear.*

Keywords: *Whirling Frequencies, Finite Elements, Didatic Software*

1. INTRODUCTION

The time of high velocity was opened in final of the XIX century, when Carl Gustaf Laval developed a butter separator powered by steam turbine that reached high velocities, up 30,000 rpm, (Dimaragonas, 1976). Nowadays the modern turbo machines reach high rotation velocities, about 50,000 rpm, for example the propulsion jet turbines used in airplanes, (Vance, 1988). This is the reason, why it is very important to study the movements of rotate machines covering concepts such as: critic velocities, whirling frequency and gyroscopic effects. The critic velocities appear when the rotor is excited in same rotation, going in resonant state and having catastrophic vibration, therefore it is indispensable that these critic velocities have been taken in consideration on the design of these machines, (Childs, 1993).

Gyroscopic effect is associated with the variation of direction of angular rotation momentum, which makes the machine show a precession. This phenomenon can be seen in an automobile driving on a curve, the engine turns around its crankshaft, which this movement is damping by dashpot, (Mabie, 1980). This precession effect makes each natural frequency be double, this one highest, is called of forward, and the smallest, is called backward. This couple of frequencies is called as whirling and the direction depends of the initial conditions.

2. FINITE ELEMENTS MODEL

The numbers of degrees of freedom of the Laval rotor, chosen of model, has 4 nodes with four degrees of freedom each node, therefore 16 degrees of freedom. This way all equationing of the rotors was modeled through of 4n matrix dimension. So the dimensions of matrix [M], [C+G] and [K] are 16x16, therefore the dimension of matrix [A] and [B], eigenvalue problem, are 32x32 This way, was obtained 32 eigenvalues double in pairs, that is, 16 natural frequencies.

2.1 TRANSFER MATRIX FROM LOCALS COORDINATES TO GLOBAL

The transfer matrixes transforms the local in global coordinates, are given by Eq. (1), (2) and (3), (Oliveira, 2003):

$$[M] = \sum_{e=1}^{n} [a]_e^T [m]_e [a]_e \tag{1}$$

$$[K] = \sum_{e=1}^{n} [a]_e^T [k]_e [a]_e \tag{2}$$

$$[C+G] = \sum_{e=1}^{n} [a]_e^T [c+g]_e [a]_e \tag{3}$$

Where [M], [K] and [C+G] are: the mass, stiffness, damping and gyroscopic global matrix respectively. The [m], [k] and [c+g] are the mass, stiffness, damping and gyroscopic local matrix respectively. The [a] is the transfer matrix, compound of “1” and “0”.The transfer matrix dimension is 8x8n for shaft element and 4x8n for bearing and disk elements.

2.2 CAMPBELL DIAGRAM

The rotor simulated is showed in the Fig. (1).

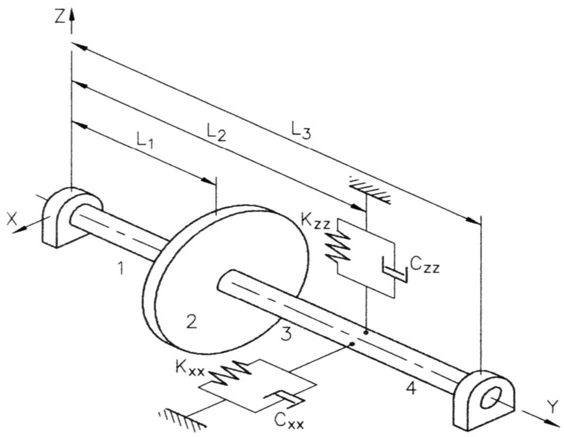

Figure 1. Laval rotor, (Lalanne, 1990).

The constants used to simulate the Laval rotor are in the Tab (1):

Table 1. Inputs of the Laval rotor.

Symbol	Value	Unit
E	2.10^{11}	N/m^2
υ	0.3	-
ρ	7800	kg/m^3
S_{shaft}	$3.142.10^{-4}$	m^2
I_{shaft}	$7.854.10^{-9}$	m^4
L	0.4	m
L_1	0.13333	m
L_2	0.2666	m
I_{DY}	0.1861	$kgxm^2$
I_{Dx}	$9.457.10^{-2}$	$kgxm^2$
I_{Dz}	$9.457.10^{-2}$	$kgxm^2$
R_{in} (disk)	0.01	m
R_{out} (disk)	0.15	m
k_{xx}	2.10^5	N/m
k_{xz}	0	N/m
k_{zx}	0	N/m
k_{zz}	5.10^5	N/m
c_{xx}	40	Nxs/m
c_{xz}	0	Nxs/m
c_{zx}	0	Nxs/m
c_{zz}	100	Nxs/m

Through this developed software, was yield the Campbell diagram of Laval rotor such as showed in the Fig. (2). It was have discretized the Laval rotor in four nodes. So, it was obtained sixteen degrees of freedom.

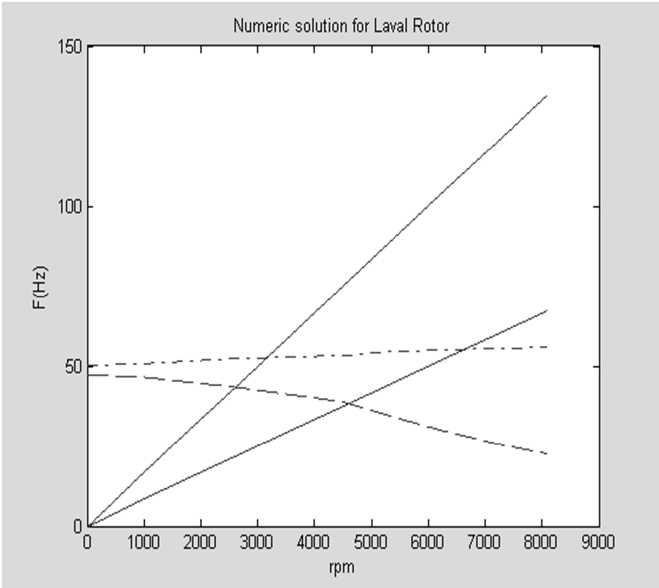

Figure 2. Laval rotor simulated.

It was have compared the solution found in numeric and analytic method. The results are showed in the Tab (2) and the rotation frequency is equal 955 rpm.

Table 2 Natural frequency comparisons to 955 rpm.

Frequency	**Analytic solution value (Hz)**	**Numeric solution value (Hz)**	**Relative error (%)**
F_1	45.89	46.6	1.54
F_2	52.90	50.8	3.98

The flow chart of the didactic program developed is showed in the Fig (3) and graphic unit interface in the Fig (4).

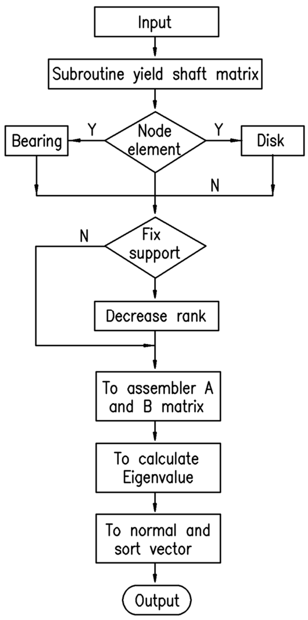

Figure 3. Flow chart of didactic software developed.

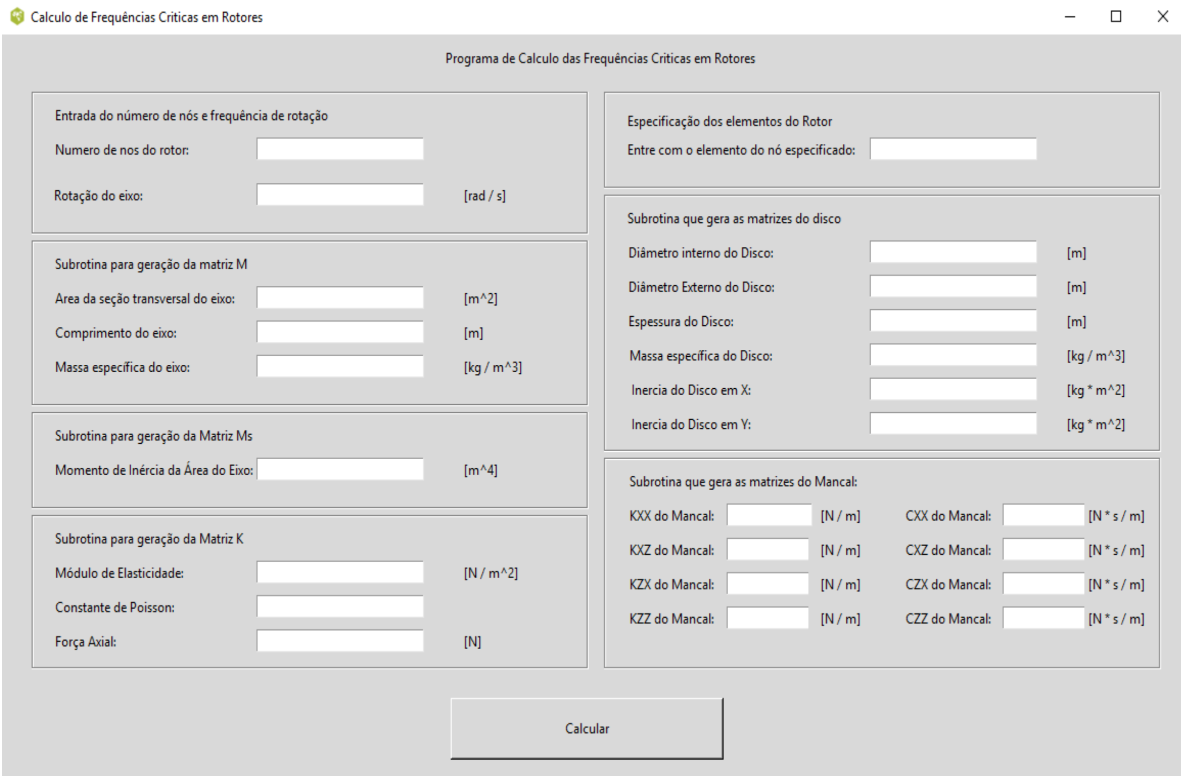

Figure 4. Graphic Unit Interface (GUI).

2.3 Multi disk rotor simulation

The rotor simulated is the multi disk rotor, (Lalanne, 1990), showed in the Fig. (5):

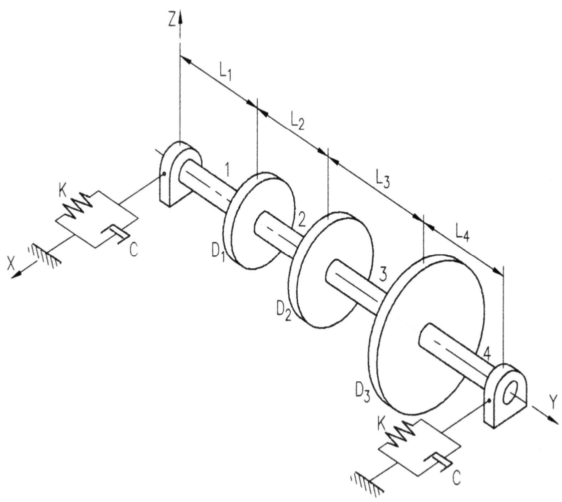

Figure 5 Multi disk rotor simulated

The constants used to simulate the multi disk rotor are in the Tab (3):

Table 3 inputs of multi disk rotor

Symbol	Value	Unit
E	2.10^{11}	N/m^2
υ	0.3	-
ρ	7800	kg/m^3
S_{shaft}	0.0079	m^2
I_{shaft}	$4.9.10^{-6}$	m^4
L_1	0.2	m
L_2	0.3	m
L_3	0.5	m
L_4	0.3	m
I_{DY1}	0.1232	$kgxm^2$
I_{DY2}	0.9763	$kgxm^2$
I_{DY3}	1.1716	$kgxm^2$
$I_{Dx1} = I_{Dz1}$	0.0646	$kgxm^2$
$I_{Dx2} = I_{Dz2}$	0.4977	$kgxm^2$
$I_{Dx3} = I_{Dz3}$	0.6023	$kgxm^2$
k_{xx}	$5.7.10^7$	N/m
k_{xz}	0	N/m
k_{zx}	0	N/m
k_{zz}	7.10^7	N/m
c_{xx}	5.10^2	Nxs/m
c_{xz}	0	Nxs/m
c_{zx}	0	Nxs/m
c_{zz}	7.10^2	Nxs/m

Through this developed software, it was yield the Campbell diagram of multi disk rotor such as showed in the Fig. (6). It was have discretized the multi disk rotor in five nodes. So, it was obtained twenty degrees of freedom.

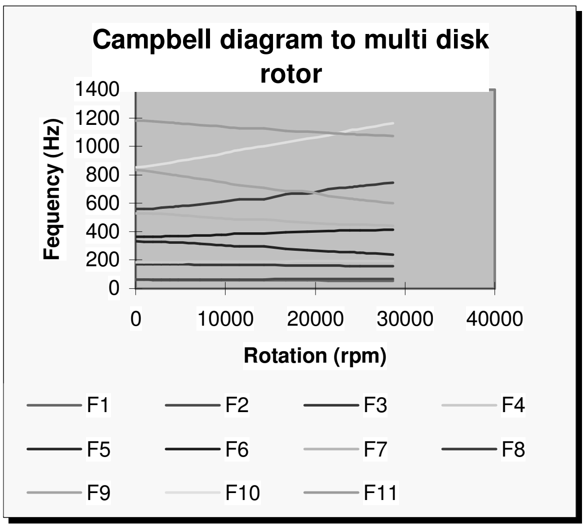

Figure 6 Campbell diagram to multi disk rotor

3. CONCLUSION

The developed software has showed a well performance in comparison with analytic results, because the proposed model has a good agreement with the rotor dynamic theory. The advantage of this software is that it takes in consideration the gyroscopic effect on disk element. The shaft element was modeled according to Timoshenko beam formulation, including shear effect and rotatory inertia in four nodes. Moreover this software can be used in rotors where the stiffness of the bearings are very high, in this case they are modeled such as fix support and it is much cheaper than other specialized softwares. Besides that, this software is a didactic tool for the teaching of rotor dynamics phenomena, so it is very useful for mechanical engineering students.

4. ACKNOWLEDGEMENTS

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6. RESPONSIBILITY NOTICE

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