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FINITE ELEMENT ANALYSIS OF PETROBRAS MOTOR / COMPRESSOR 105-J AND SUPPORT PEDESTAL WITH PROPOSED PEDESTAL MODIFICATIONS

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Abstract. The Petrobras motor/compressor 105-J/FAFEN-SE is installed on reinforced concrete columns. The columns all have a square cross-section of 0.45 meters per , spaced between two and three meters apart. The slendercolumns and the large weight of moto/gearbox/compressors lead to significant support flexibility of the machine. This flexibility would reduce the effectiveness of the bearings and contribute to the large synchronous and twice running *speed vibrations experienced by the motor. This flexibility makes the machine susceptible to electrical unbalance in the motor and rotor misalignment. In the last part of this report, several pedestal modifications are investigated with the goal being to move all support modes away from 60 Hz , however, it became clear that the only way to significantly alter the pedestal dynamics around 60 Hz was by adding additional concrete columns under the motor, thus increasing the vertical stiffness in that area*
Keywords: Pedestal, Natural frequence , turbomachinery, Finite Elements, vibration

1. Introduction

The motor for Petrobras motor/compressor 105-J at FAFEN-SE is installed approximately three meters above ground level on reinforced concrete columns. The motor is coupled to a gearbox and drives two compressors. The columns all have a square cross-section of 0.45 meters per side and are spaced between two and three meters apart. It has been conjectured that the combination of slender columns and the large weight of the motor, gearbox, and compressors lead to significant support structure flexibility in the operating range of the machine. This flexibility would reduce the effectiveness of the bearings and contribute to the large synchronous and twice running speed vibrations experienced by the motor.

A detailed finite element analysis of the pedestal and machinery components was performed in order to better understand the dynamic nature of the structure. The analysis was done in several stages. A preliminary model was built encompassing the four concrete columns directly beneath the motor, the steel truss beams between the columns, the steel support beams that the motor sits upon, and the motor itself. This model was discussed in a previous report and presented at FAFEN-SE in June, 2003. The results from this model showed a vertical pedestal mode with a natural frequency very close to the twice running speed frequency of 60 Hz (3600 rpm). Further, the frequency response indicated that mode can be easily excited by harmonic forces acting at the bearings. These findings suggested that structural resonance is a concern with respect to the 60 Hz vibration.

The preliminary model had several limitations as noted in its description. Most significantly, the boundary condition at the gearbox side of the motor was not well characterized due to omission of the gearbox, compressors, and the large steel box beams that connect them to the motor support beams. In order to address this shortcoming, a more comprehensive model was built to include the additional machinery and beams. This model was also described previously and presented at FAFEN-SE in June, 2003. The results from this model showed two modes close to 60 Hz. One was similar to the vertical mode seen in the first model while the other involved rotations of the motor and radial compressor. As before, the frequency response indicated that the pedestal can be easily excited by harmonic forces acting at the bearings. Additionally, mass and stiffness matrices for the expanded model were produced and incorporated into a separate rotor model. Results from that analysis (described separately) showed the second rotor critical speed to be much closer to twice the running speed than it would be if the motor were on an inflexible support.

The expanded model showed the modes with natural frequencies close to twice the running speed frequency to all have significant bending in the beams between the motor and the gearbox. It was therefore important to make this region of the model as accurate as possible. Toward that end, the modeled gearbox support columns and beams had to be modified to reflect their true geometry. Also, a plant visit to FAFEN-SE in June of 2003 afforded the opportunity to improve the model due to measurements taken on site as well as additional drawings provided by Petrobras at that time. The updated full model is described in the following sections. The results are similar to those found with the previous model but with some differences. Several of the lower natural frequencies have changed due to inclusion of additional stiffening I-beams in the updated model. Also, the updated model shows a number of additional modes with frequencies close to 60 Hz. Several of these additional modes are not significant as they are entirely due to bending of the stiffening I-beams and have no motion at the bearing locations. However, it is important to note that the frequency response at 60 Hz is almost entirely vertical in the updated model whereas the previous model showed some horizontal response as well.

This finite element analysis shows that support structure flexibility plays an important role in the vibrations experienced by Petrobras motor/compressor 105-J, particularly at the twice running speed frequency of 60 Hz. This flexibility makes the machine susceptible to electrical unbalance in the motor and rotor misalignment, either of which can produce excitations at twice the running speed frequency. In the last part of this report, several pedestal modifications are investigated with the goal being to move all support modes away from 60 Hz. It was hoped that this could be done by augmenting the existing steel beams between the gearbox and motor and several variations were tried. However, it became clear that the only way to significantly alter the pedestal dynamics around 60 Hz was by adding additional concrete columns under the motor, thus increasing the vertical stiffness in that area.

2. Existing Pedestal Model

2.1. Description

The model encompasses the ten concrete columns supporting the motor and compressors, the steel beams installed between the columns for lateral stiffening, the large steel box beams that the machinery sits upon, as well as the motor, gearbox, and compressors. The model is comprised of 18,044 elements and 27,378 nodes with a total of 82,134 degrees of freedom (3 per node). It was developed and solved using the ANSYS finite element package.

The concrete columns are made of steel reinforced concrete. They are modeled using ANSYS element number 65. This is an 8-node, reinforced concrete, solid element and includes options for specifying rebar properties. It was found that the natural frequencies determined for a column without including the rebar (essentially a homogenous concrete column) varied by as much as 15% from the frequencies for a reinforced column. This variation is the primary reason that the model was constructed from solid elements rather than beam elements.

For simplicity, the motor, gearbox, and compressors are modeled as homogenous solid objects

2.2. Limitations

It is important to note a couple of limitations in the model. First, no attempt has been made to model the internal geometry of the machines. They are treated as simple lumped masses. Also, the columns, beams, and machinery are assumed to be rigidly connected. Any relative motion between the machinery and the columns or the columns and the truss beams is neglected. The columns are also assumed to be rigidly fixed at the floor level meaning any column dynamics between the floor and the foundation block are neglected.

2.3. Natural Frequencies and Mode Shapes

Two sets of natural frequencies are shown in table 2.1. The first set was determined using the full finite element model. The second set was determined using a reduced order model. This was done to limit the size of the mass and stiffness matrices. Using the full model would have produced huge matrices that could not be practically incorporated into the rotor model. However, the full model can be approximated by a limited set of master degrees of freedom.

In this case we are concerned with motions at the two bearing locations and frequencies up to about 60 Hz which is twice the machine's operating speed. A set of 155 master degrees of freedom was chosen such that the reduced model frequencies and mode shapes were in close agreement with those of the full model up to approximately 75 Hz. The frequency range of interest is then well characterized by the reduced model.

Note that some full model modes do not have a corresponding reduced model mode. Several full model modes only involve bending of the I-beams that were installed for lateral stiffening (modes 9-12 and 16-19). Matching these modes would have required keeping several additional master degrees of freedom and led to a corresponding increase in the size of the mass and stiffness matrices. Unfortunately, larger matrices cause difficulties with the software used in solving the rotor model. All of the full model modes involving motion of the bearings, machinery components, concrete columns, and large steel beams were matched by the reduced order model. The mode shapes shown here were found using the full model but the corresponding reduced model mode shapes are essentially identical. The dotted lines in the mode plots represent the edges of the undisplaced model.

It is clear that the support structure contributes undesirable dynamics to the system. Mode 4 has a natural frequency that is very close to the running speed of the machine while modes 8 and 13 have frequencies near the twice running speed frequency. The mode shapes for these three modes all show movement of the motor and bearings. Fortunately, it will be seen later that the forced response of the structure at 30 Hz is well damped so mode 4 is not as much of a concern as modes 8 and 13. Also it should be noted that the motor motion in modes 8 and 13 is primarily vertical with smaller amounts of lateral rotation (mode 8) and translation (mode 13).

Table 1. Support Structure Natural Frequencies (Hz)

Mode	Full Model	Reduced Model	Mode	Full Model	Reduced Model
1	10.44	10.45	11	59.04	
2	19.67	19.71	12	59.39	61.88
3	21.80	21.85	13	60.28	61.22
4	31.93	32.08	14	62.74	63.95
5	39.06	39.29	15	66.58	67.88
6	50.04	50.53	16	68.86	
7	51.70	52.27	17	68.89	
8	57.28	58.21	18	68.91	
9	58.45	60.79	19	68.98	
10	58.75	62.59	20	71.64	72.52

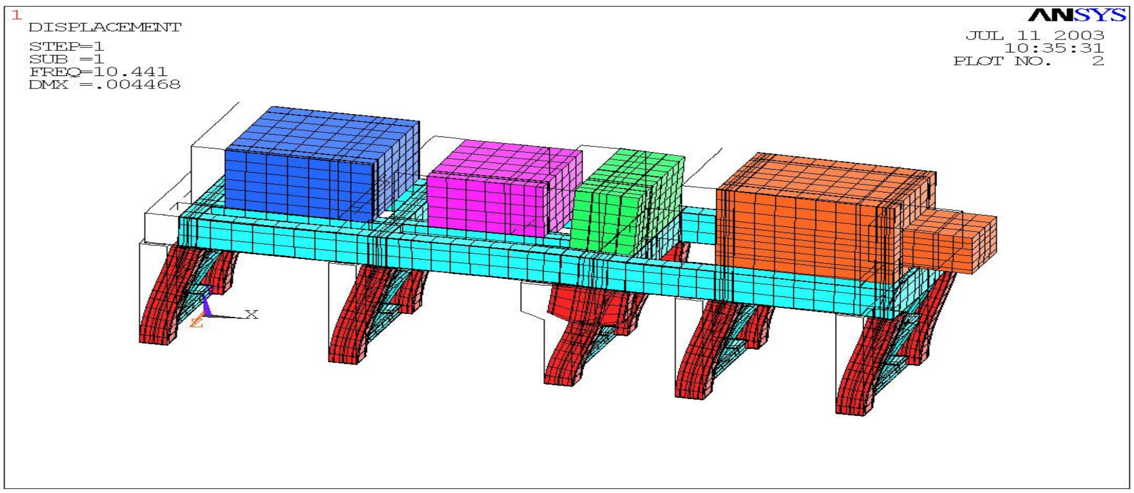


Figure 1. Support Structure Natural Frequencies (Hz)

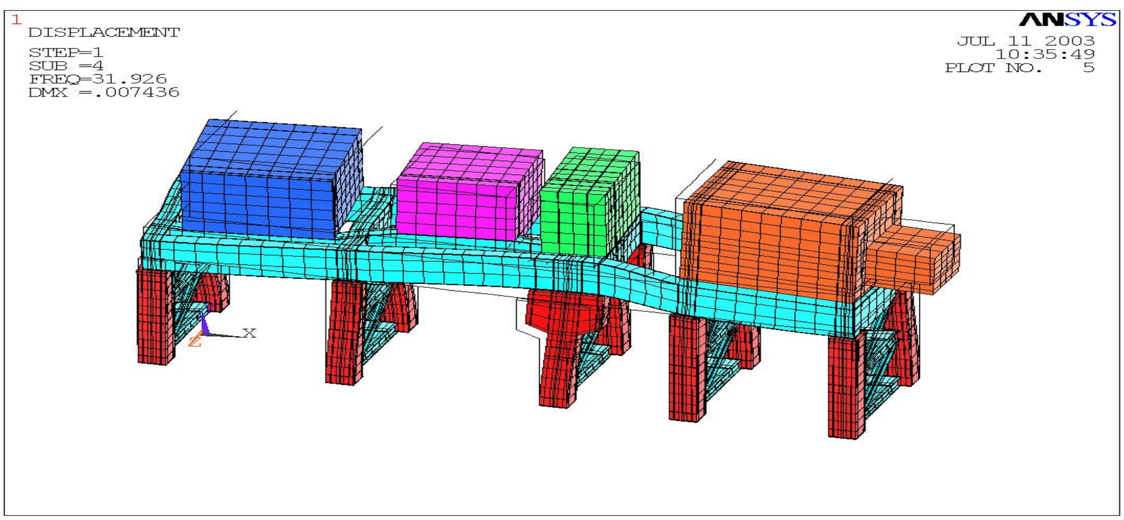


Figure 2. Support Structure Natural Frequencies (Hz)

2.4. Frequency Response Functions

In addition to the natural frequencies and mode shapes it was desired to compute the mass and stiffness properties of the support structure with respect to the two bearing locations. Frequency response functions were calculated for four excitation conditions. Figure 2.4.1 shows the radial vertical and horizontal response functions as well as the axial response function at the gearbox side bearing due to a sinusoidal excitation at the same location with a magnitude of 1 newton in the vertical directionSeveral important observations can be made from the frequency response functions. First, the response at the bearings for forcing frequencies near 30 Hz is much smaller than for forcing frequencies near 60 Hz. This indicates that the support structure is contributing to the machine vibration problems at 60 Hz (twice the running speed frequency) but is relatively less important to the 30 Hz vibrations. Also, with a 60 Hz forcing frequency, the structure is much softer in the vertical direction than the horizontal direction. This means that altering the support structure dynamics will require vertical components such as new concrete columns.

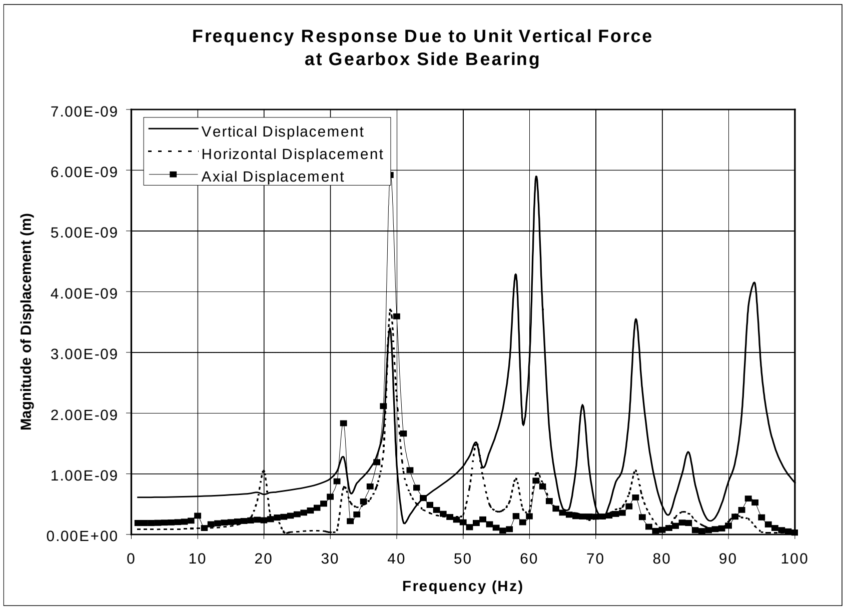


Figure 3. Frequency Response Due to 1 N Vertical Excitation at Gearbox Side Bearing

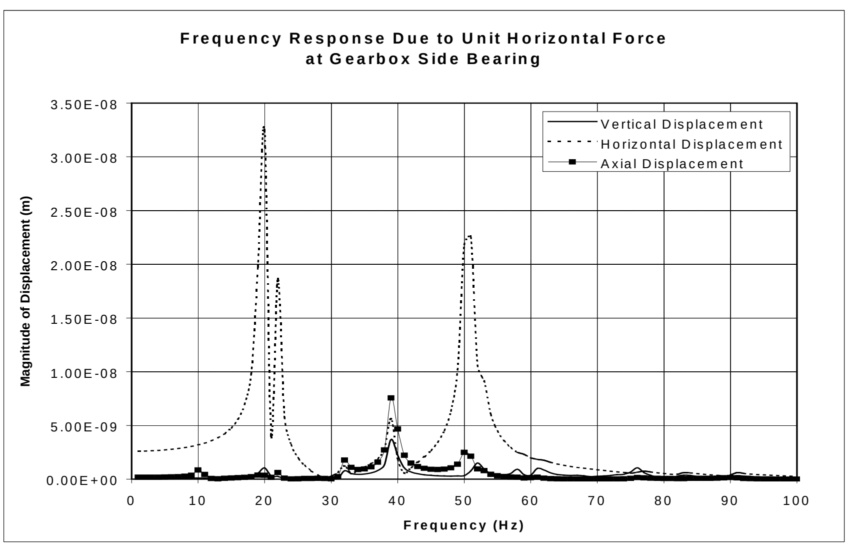
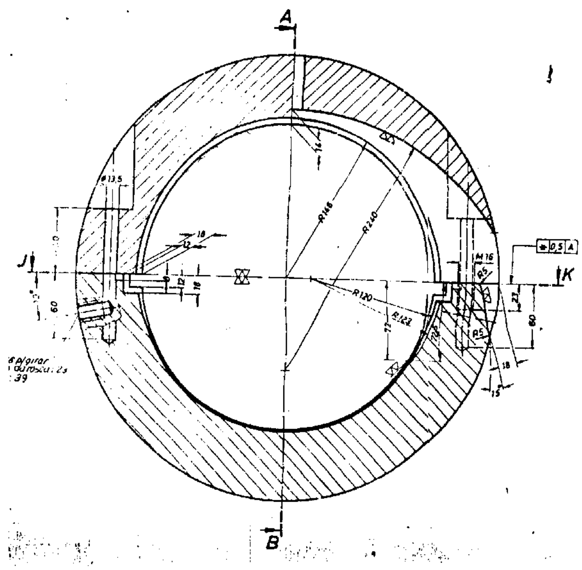


Figure 4. Frequency Response Due to 1 N Horizontal Excitation at Gearbox Side Bearing

3. Bearing Analysis

3.1. Existing Bearing

The rotor-bearing is an integrated system. The bearing dynamic coefficients must be known in order to analysis the rotor dynamics. The existing inboard and outboard bearings are identical partial arc fixed geometry bearings. But their loads are different due to the asymmetrical rotor. The gravity load is 9477 lbf and 11222 lbf on the inboard and outboard bearings respectively. The shaft runs at a constant speed of 1800 rpm. The bearing geometry is shown in Figure 3.1 and its key parameters are list in Table 3.1. The oil has density of 8.26e-5 lbf-s^2/in^4. The oil viscosity is a strong function of temperature with 3.83e-6 Reyn at 104 ^{o}F and 6.40e-7 Reyn at 212 ^{o}F (ISO VG32).



	mm	in
Journal Diameter	280	11
Diametrical Clearance	0.42-0.56	0.0166-0.022
Axial Length	195	7.68
Pad Arc Length	130^o	
Preload	0.0	
Pad Thickness	65	2.5

Figure 5. Gearbox Side Bearing Properties

Since the bearings are ring-lubricated, starvation is inevitable and the oil flow rate cannot be accurately determined. The oil delivery flow rate is a very complex function of the ring design, oil viscosity, shaft speed and the depth of submerge. Thus, a range of starvation levels was investigated in the current analysis. At a certain shaft speed, the oil supply flow rate was decreased from the flooded condition till the maximum pad temperature exceeded 200 ^{o}F or higher. This range is expected to bracket the actual oil feed condition.

Figures 3.2 to show the principal dynamic coefficients of the inboard bearing as a function of oil supply flow rate. The shaft speed was 1800 rpm and the clearance manufacturing tolerance was taken into account. For all three clearances, the oil flow reduction mostly affected the vertical stiffness and damping. Both K_{yy} and C_{yy} increased as the bearing became more and more starved. However, since the stiffness increased more than the damping, the bearing had less effective damping as starvation aggravated. The outboard bearing has similar trends.

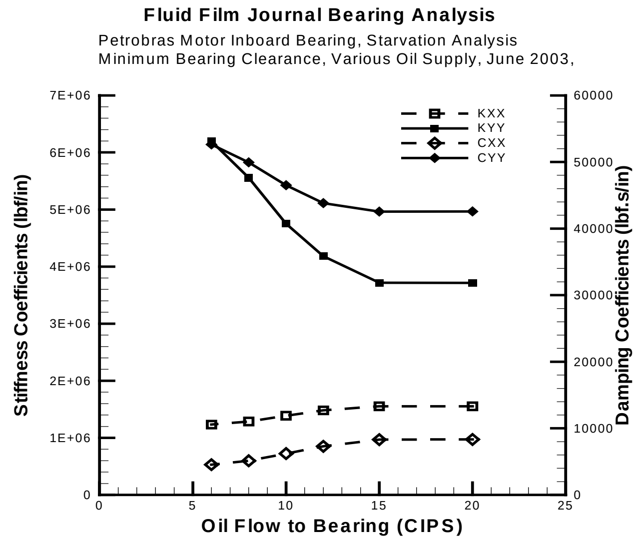


Figure 6. Principal Coefficients vs. Flow Rate, 1800 rpm Inboard Bearing, Minimum Clearance

Tables 3.2 list the principal dynamic coefficients as functions of both shaft speed and oil flow rate. The calculation was based on the average clearance.

Table 2. Levels of Starvation for Various Shaft Speeds and Oil Supply Flow Rates,

	Flooded	20 (CIPS)	15 (CIPS)	12 (CIPS)	10 (CIPS)	8 (CIPS)	6 (CIPS)
1000 (rpm)	8.95	100%	100%	100%	100%	89%	67%
1400 (rpm)	12.66	100%	100%	95%	79%	63%	47%
1800 (rpm)	***16.29***	***100%***	***92%***	***74%***	***61%***	***49%***	***37%***
2200 (rpm)	19.84	100%	76%	60%	50%	40%	30%
2800 (rpm)	24.88	80%	60%	48%	40%	32%	24%
3600 (rpm)	32.04	62%	47%	37%	31%	25%*	19%*
4200 (rpm)	36.91	54%	41%	33%*	27%*	22%*	16%*

*Not calculated due to severe degree of starvation (pad temperature over 200 ^{o}F)

3.2. New Bearing

To solve the vibration problem, the new bearing needs to have more damping to reduce the excitation response. Usually, changing the bearing stiffness could shift the critical speeds farther away from the running speed. But this would not effectively improve the current system. First, the bearing stiffness can hardly be reduced below 2×10^6 lbf/in due to the large rotor weight. Second, the pedestal's dynamic stiffness is about 5×10^6 lbf/in at 30Hz excitation

Frequency. The pedestal is even softer (< 3×10^6 lbf/in) under 60Hz excitation. Thus, even if the bearing is very stiff, the overall stiffness will not be significantly increased when coupled with the flexible substructure. Therefore, the benefit from the new bearing is primarily increased damping. To achieve maximum effective damping, the new bearing needs to be soft and long.

According to Petrobras, the journal section could be extended by 1.97 in (50 mm) to accommodate a longer bearing. Based on the "soft and long" design philosophy, two new bearings were proposed. One has the existing axial length, denoted as "Soft Short", the other is 1.97 inch longer and named as "Soft Long". In addition, the motor manufacture also recommended two designs that are referred as "Zollern Sleeve" and "Zollern Elliptical". All those designs and their performances are summarized in Tables 3.12 and 3.13. Figure 3.5 shows the drawing of the "Soft Short" bearing.

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Table 3. Key Geometrical Parameters of the Existing and Proposed Bearings

	Existing	Soft Short	Soft Long	Zollern Sleeve	Zollern Elliptical
Diameter (in)	11	11	11	11	11
Length (in)	7.68	7.68	9.64	6.9	6.9
Dia. C_b (in)	0.019	0.02	0.02	0.015	0.012
Pad Arc (deg)	130	150	150	161	161
Preload	0	0	0	0	0.66

Table 4. Key Performances of the Existing and Proposed Bearings

	Existing*	Soft Short	Soft Long	Zollern Sleeve	Zollern Elliptical
K_{xx}	1.18	1.39	1.38	2.10	0.75
K_{yy}	4.68	3.21	2.31	3.42	5.17
C_{xx}	4.77	7.14	8.92	13.82	5.24
C_{yy}	39.83	33.14	34.92	44.26	29.37
h_{min} (in)		0.0037	0,0049	0.0036	0.0033
T_{max} (^{o}F)		155	150	164	157

The stiffness $K \times 10^6$ lbf/in, damping $C \times 10^3$ lbf-s/in.

The ring lubricated existing bearing coefficients are based on 60% flow starvation condition.

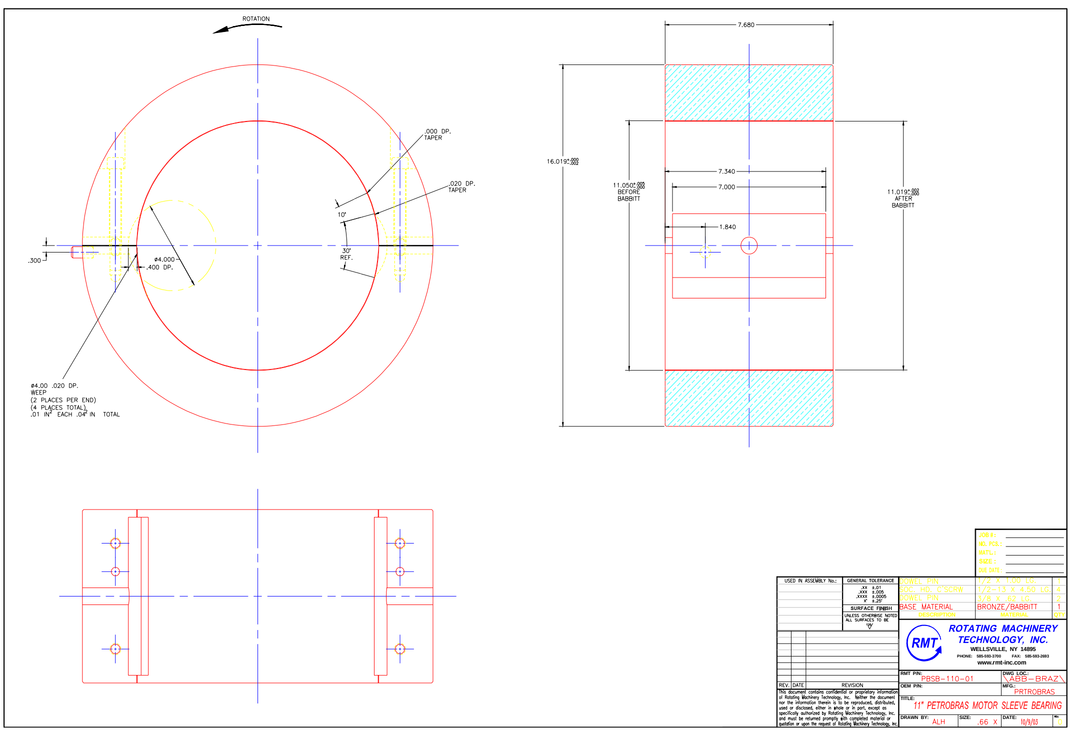

Figure 7. Autocad Drawing of the Soft Short Bearing

4. Proposed Pedestal Modifications

4.1. Design Philosophy

Analysis of the existing support structure indicates that two lightly damped pedestal modes with natural frequencies near 60 Hz are contributing to the two times running speed frequency vibrations experienced by the motor. These are modes 8 and 13 from section 2.2. Modes 9, 10, 11, and 12 involve only the lateral stiffening beams and are not a concern with respect to motion at the bearings. From the frequency response plots it is clear that the primary bearing motion due to the pedestal at 60 Hz is in the vertical direction. This explains why the addition of lateral stiffening beams in 2001 was not effective in reducing the vibration in this machine. There is also a mode close to the operating speed of 30 Hz (mode 4). However, the frequency response indicates that this mode is well damped at the bearings.

Although the primary concern in modifying the pedestal is to reduce the response at 60 Hz, there are the additional considerations of time and cost. Installing new concrete columns involves considerably more effort than simply welding additional steel beams into the existing structure. Therefore, several designs were considered that only involved adding new steel beams. Several of these are shown in appendix A along with the corresponding frequency response plots. Unfortunately, none of these designs were successful at altering the pedestal response near 60 Hz.

That leaves the option of building additional concrete columns. Two alternatives are considered. The first involves adding two new columns at either end of the motor. The second adds one new column at the gearbox end of the motor. Adding two columns was more effective at reducing the 60 Hz vertical response at both bearing locations but caused a large increase in the horizontal response at low frequencies. Adding just one new column was somewhat less effective at reducing the 60 Hz vertical response, particularly at the outboard bearing, but also showed a much smaller low frequency horizontal response. Both designs will be incorporated into the rotor model to determine which is the best alternative., likely involving several more columns.

4.2. Proposal B: One Additional Concrete Column (Implemented Solution)

The second proposed pedestal modification is to add one concrete column under the gearbox end of the motor. The column is placed directly beneath the corner of the motor. Figure 4.1 shows the modified geometry while figures 4.2 and 4.3 show the frequency response plots.

This modification has a similar effect to that seen in the previous section at the gearbox side bearing which can be seen by comparing figures . The large peak in vertical frequency response near 60 Hz shown in figure 2.15 has again been reduced approximately 80% by the additional column and the peak in the vertical response has been moved from near 60 Hz to approximately 63 Hz providing some margin.

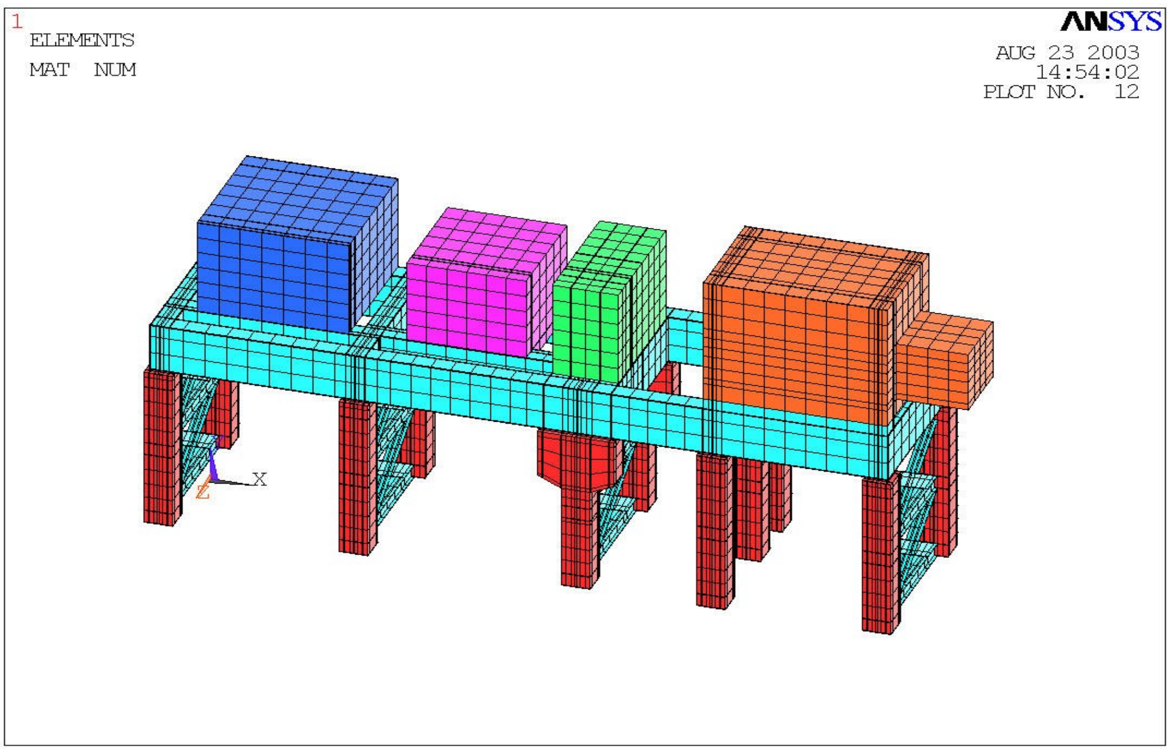

Figure 8. Geometry of Pedestal and Compressor Train with One Additional Concrete Column Added at Gearbox Gearbox End of Motor

The response at the outboard bearing has been reduced approximately 65%. This reduction is slightly smaller than that achieved by adding two columns. Also, the peak at 63 Hz is much larger at the outboard bearing if only one column is added. Thus, it would seem that adding two columns is appropriate. However, it will be seen in the accompanying rotor dynamics analysis that adding one column has some advantages. The primary benefit of adding one column as opposed to two is that the substructure is slightly softer at the outboard end. This allows some structural damping to come into play and affect the rotor response. If the outboard bearing support is more stiff, the overhung section of rotor acts like a cantilever beam with vibrations that are very difficult to control.

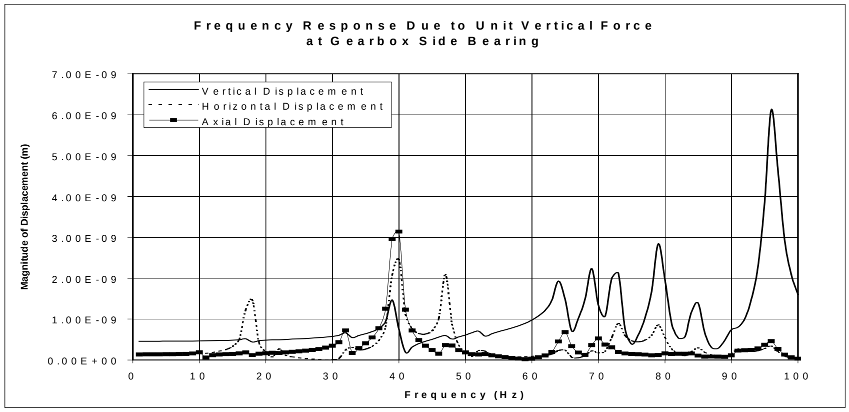

Figure 9. Frequency Response Due to 1 N Vertical Excitation at Gearbox Side Bearing

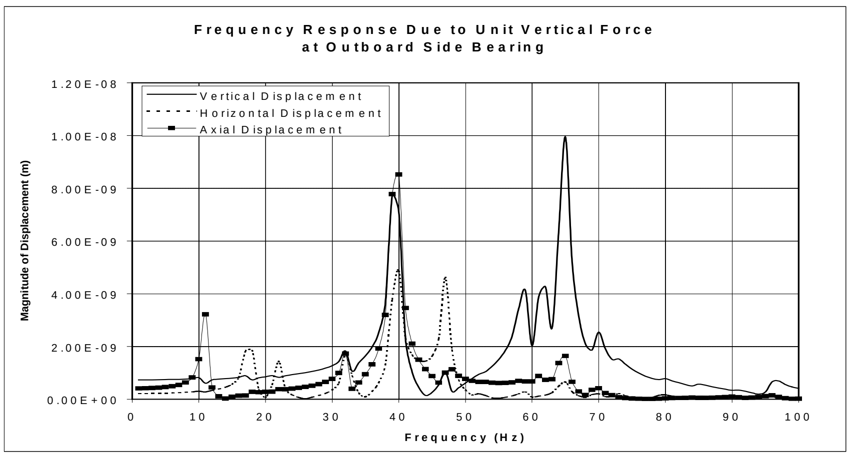

Figure 10. Frequency Response Due to 1 N Vertical Excitation at Outboard Side Bearing

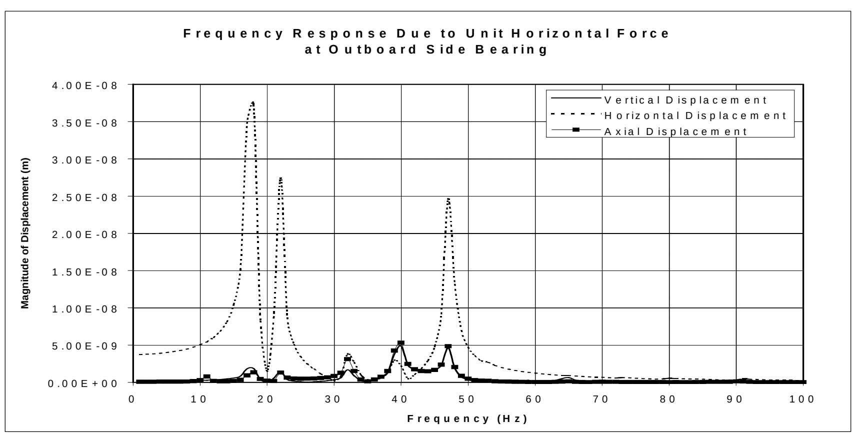

Figure 11. Frequency Response Due to 1 N Horizontal Excitation at Outboard Side Bearing

5. Conclusions

The solution proposed for this problem have been implemented only partialy .

The new bearing (according with proposed modifications) is not yet implemented but the new monocolun solution is already done .

The new bearing solution (not yet implemented) is responsible for modification in pedestal natural frequence wich are excited by the first harmonic .

The implemented new colun (according with **4.2 Proposal B**) is responsible for a great reduction in amplitude for the second harmonic . This colum has the objective of supress the inespected natural frequence that was excited by the second harmonic of the Driver .

The results presented below shows a espectro of vibration of the motor bearing , after modification , where we can see very clearly that the second harmonic was praticaly eliminated from the vibration spectrum of motor gear end bearing .

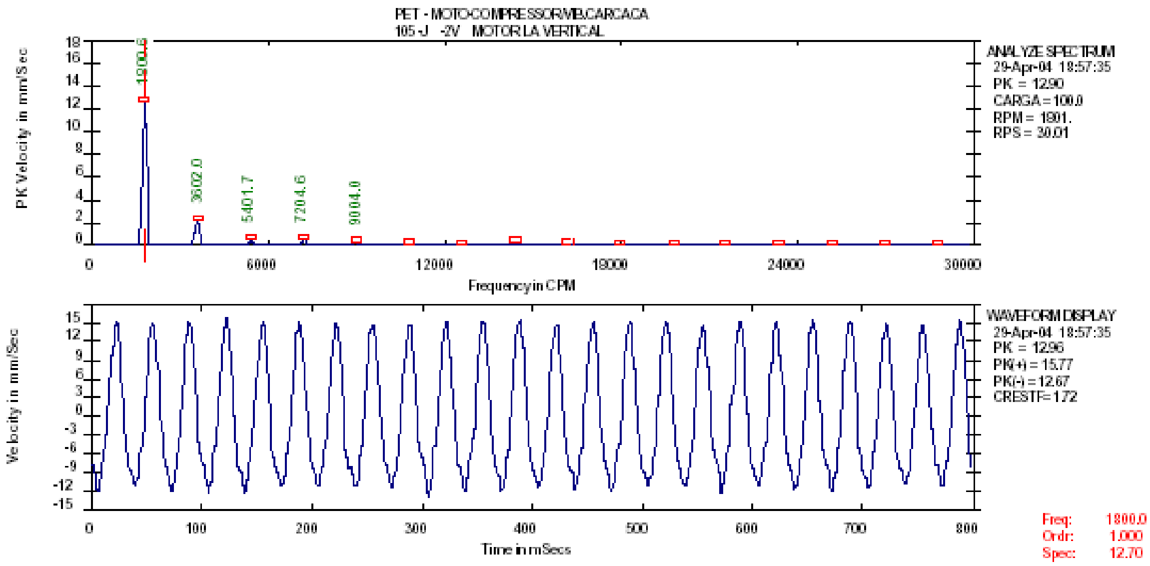

Figure 12. Vibration spectro of vertical acelerometer at Motor Coupling Side .

6. References

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