Dynamic Testing and Analysis for Pedestal Motor Base Upgrade

Jake Zwart
Spectrum Technologies Ltd.
920 Conc 8 West
Puslinch ON Canada N0B 2J0
www.spectrum-tec.com
jake.zwart@spectrum-tec.com

Bill McKevitt
McKevitt Engineering
Vancouver BC

Abstract

In the process of rebuilding a top wire former on a paper machine, the drive requirements increased from 550 HP to 1250 HP, requiring a motor of twice the mass mounted on a pedestal 4.65 m high. The gearbox was moved from the paper machine to the motor base, adding additional mass, and requiring increased platform space on the pedestal to hold the gearbox. To ensure that the redesigned pedestal would not cause any vibration problems, operational and modal testing was performed on the existing pedestal: and this test information was used to validate an FEA model. The validated model was then used to evaluate design ideas for the redesigned pedestal, and determine a practical design alternative by ensuring that the resonant frequencies would not interfere with operating frequencies. Once the dynamic model had been constructed, it could be used for static stress analysis to ensure that other aspects of the design are valid.

Introduction

A paper machine rebuild was planned, where the top former is being rebuilt to increase the quality of the paper. This required a new drive motor of 1250 Hp replacing the existing 550 HP motor, which has double the mass. In addition the existing gearbox on the paper machine was going to be replaced with a new gearbox placed on the pedestal close to the drive motor. These changes added 4775 kg of additional mass to be carried by the pedestal. The pedestal needed to be extended to provide the space to mount the gearbox, again adding additional mass. The base for the motor and gearbox is 4655 mm above floor level and the couch motor at 2480 mm above floor level, hence the pedestal needs to be carefully designed to avoid resonances coinciding with operating frequencies. There was excessive vibration on the current pedestal, and the design team wanted to ensure the new design would operate without excessive vibration.

The design procedure was to measure the current vibration level on the existing pedestal during operation. In addition the natural frequency characteristics were desired.

Then a Finite Element Analysis (FEA) model would be developed for the existing structure, to determine if it could predict the measured natural frequencies. This model would be tweaked to obtain good results. This model would then be extended by changing it to account for the new motor and gearbox, as well as the pedestal extension. The design would then be checked to ensure that any natural frequency was at least 20% away from the motor operating frequencies.

Table 1 The increase in mass due to the motor and gearbox

Item	Mass current [kg]	Mass proposed [kg]
Couch Motor	3990	3990
TWTR motor	3175	6350
Gearbox and soleplate	N/A	1600

Figure 1 Bottom of pedestal showing intermediate level motor with paper machine to the right

Software Selection

The criteria for the software used was that it had to be commercial quality and reasonably user friendly. In my toolbox were Ideas for Test, used for both data acquisition and analysis, and Xmodal, the UC modal analysis software, which is extremely cost effective and has state of art modal algorithms that are migrating into some of the big name expensive software test packages.

My FEA software needed updating, so I investigated CAElinux, a Linux distribution with many open source CAE tools preloaded. From this CD I selected a package called Salome-Meca, which integrates Salome as a pre & post processor and Code-Aster, which is the FEA solver developed and used by the French nuclear industry. This met the requirements of being cost effective and robust.

There was a definite learning curve and the software has its own idiosyncrasies, but in my experience this is consistent with commercial software vendors. What I did miss is the software support contract, though this too is available, but at commercial rates. The online support forums and wiki's are an excellent source of advice, but not as quick as calling a support hotline.

Testing the Existing Pedestal

The vibration of the machine under normal operating conditions was measured with respect to the reference vibration location on the base of the top motor in the machine direction (lateral direction). This reference location was used to process the vibration into a set of operating deflection shapes, giving the amplitude ratio and phase relationship for three directions at each point with respect to this reference location. The vibration at problem frequencies have been processed to visualize the vibration.

A stick figure of the vibration at 9.75 Hz is shown in Figure 2, showing that the vibration is predominately MD with a small CD component. At 21 Hz, Figure 4, the vibration is primarily yawing, and this coincides with the rotational speed of the twin wire turning roll. The results of the modal testing, Figure 6, show that there are resonances at 9.4 Hz. 22 Hz, and 45 Hz.

Figure 2 Operating deflection shape at 9.75 Hz

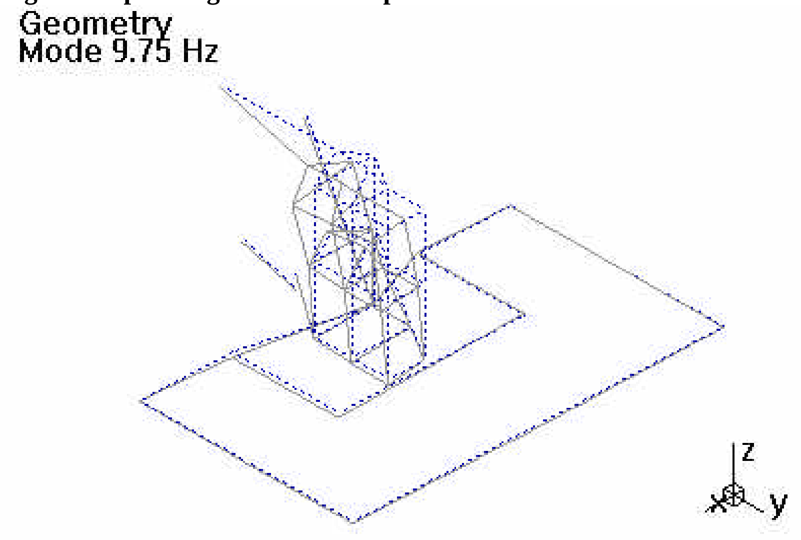

Figure 3 Deformed shape of above figure for final paper

Figure 4 Operating deflection shape at 21 Hz, the turning roll rotational speed

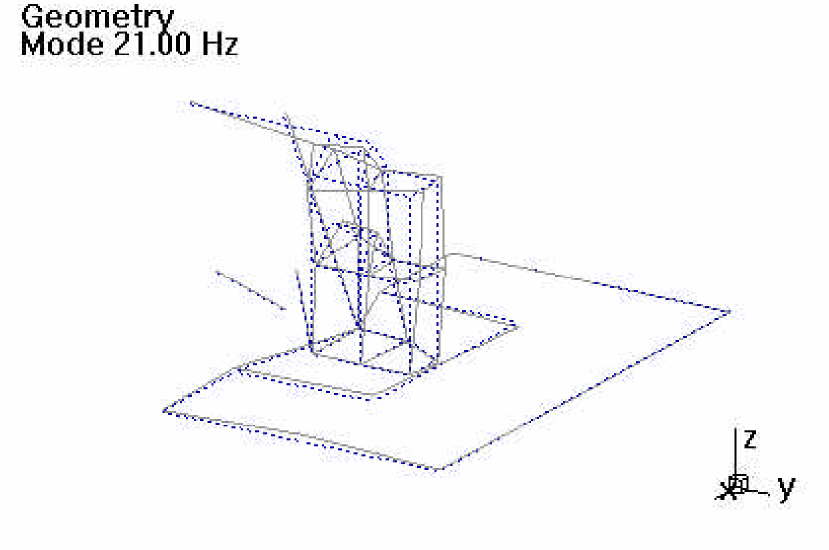

Figure 5 Deformed shape of above figure for final paper

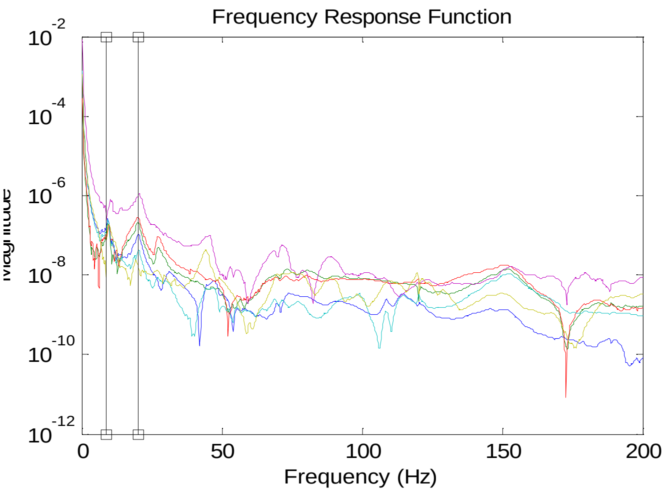

Figure 6 FRF's showing the natural frequencies

Finite Element Modeling

To develop an adequate design for the larger motor and gearbox, a finite element model was developed for the existing structure and motors, and compared with the test results. The model for the existing structure was then modified and the natural frequencies were recalculated along with the mode shapes for different design alternatives. The design was refined until an adequate design was developed. Since space was tight in this area, with existing piping, wiring and ducting, as well as other equipment in the way, the design was challenging. Fortunately, the two pedestals shown in the foreground of Figure 7, were for two drive motors that were decommissioned over the years, and thus available as support columns.

The operating speeds of motors on the pedestal, as well as the one directly beside the pedestal are shown in Table 2. During testing the paper machine operating speed was 1201 mpm, the normal operating speed is 1250 mpm, and 1300 mpm is also shown should a machine speed increase occur in the future. Table 3 shows the predicted natural frequency results for different design iterations discussed in the subsequent sections along with the test results.

Table 2 Operating frequencies of couch and TWTR to avoid

Reel Speed	1201 mpm	1250 mpm	1300 mpm
TWTR current (for reference purposes only)	21.1	22.0	22.8
TWTR new	16.5	17.1	17.8
Couch	17.9	18.6	19.4
Bottom Wire Return (Forward Drive) Roll	21.05	21.9	22.8

Table 3 Frequencies at different design configurations

	Shape	Test results	Current base	First design	Reduced Platform	Reduced Platform
Mode			with motors	with motors	NE columns tied together	NE Col's tied, & west Column extended
1	CD rocking		11.9	10.0	10.5	10.6
2	MD rocking	9.75	13.8	12.7	13.4	13.4
3	Yawing	21	23.1	19.5	24.3	26.6
4	CD rocking			22.6		26.0
5	MD rocking		30.8			33.0
7	Yawing		33.5		26.5	
8	Yawing	45	43.9			

Current Design

The current geometry was modeled as shown in Figure 7. The floor, pedestal and motors were all modeled using solid elements, while the floor beams were modeled with beam elements. The element size selected for the modeling was based on having an element edge length of 200 mm as shown in Figure 8. The elements selected were linear tetrahedral elements, for simplicity in meshing. The beam elements were also linear beam elements to match the solid elements that they shared nodes with.

This model yielded resonant frequencies that were fairly close to the measured natural frequencies, with the yawing being 10% higher than the measured frequency.

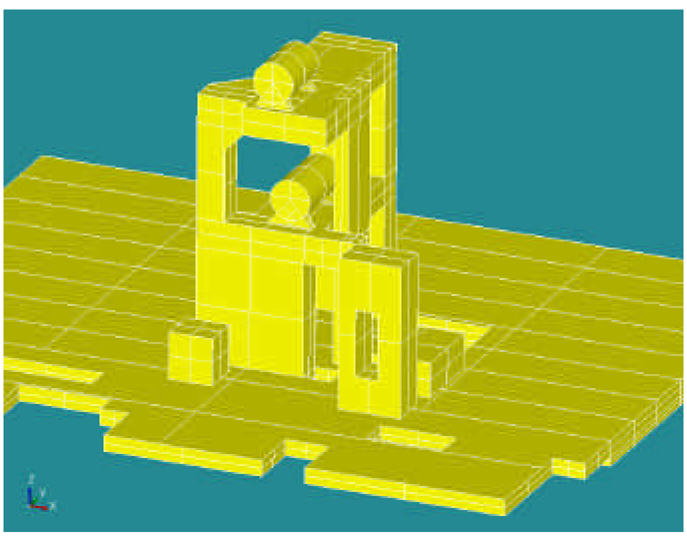

Figure 7 Geometry model of existing pedestal base and surrounding floor

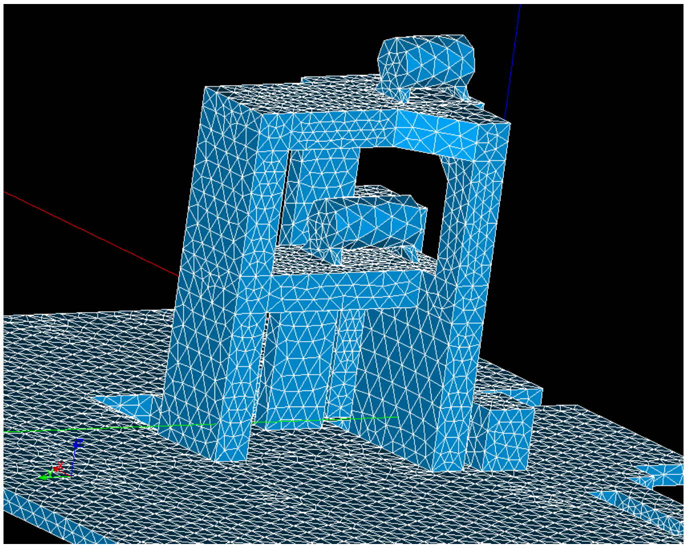

Figure 8 Mesh used for the current platform with existing motors

First Design Iteration

The consulting engineering firm contracted for this upgrade developed the initial design iteration. The modification was a platform extension to accommodate the new gearbox as shown in Figure 9. This is a large platform for the gearbox, and the results showed that there is a resonance about 1 Hz above the couch motor operating speed, well within the 20% band where there should be no resonance. The yawing of the motor base is shown in Figure 10.

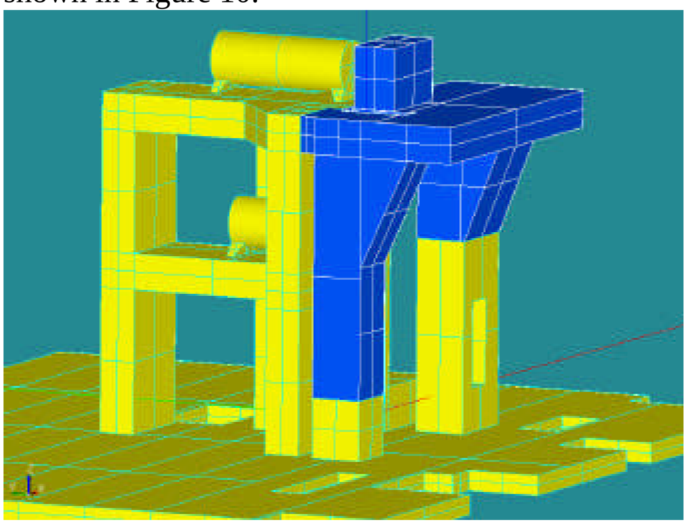

Figure 9 First design iteration with extended platform

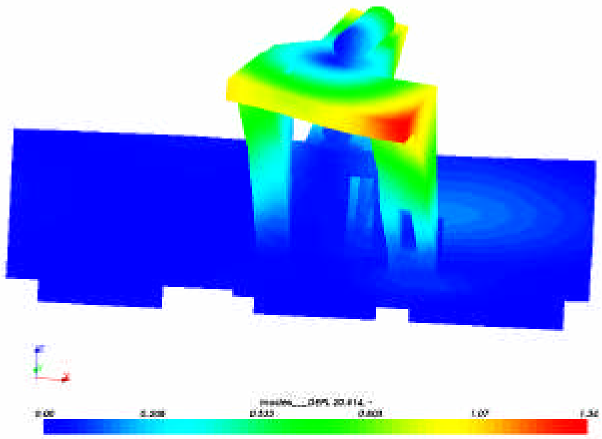

Figure 10 Yawing centered on front of motor at 20.8 Hz

Final Design

To increase this resonant frequency, a variety of design alternatives were tried. Initially the platform size was reduced to decrease mass and thus increase the resonant frequency. This helped but was insufficient. Then some cross bracing was modeled, but when the feasibility of adding this bracing was checked with the mill, it was determined existing piping would prevent its installation. In each case the shapes of the vibration as predicted by the FEA was used as a guide to determine where modifications should be made. Since both yawing and CD rocking have frequencies in the problem range, a stiffness increase for both of these was required.

The proposed new column on the short decommissioned motor base was integrated into the existing column near that location, greatly increasing its stiffness in both the machine and cross directions as shown in Figure 11. In addition, the new platform thickness was reduced to 400 mm. However, since part of the platform went around a column, that portion was reinforced with a steel plate and made 600 mm thick.

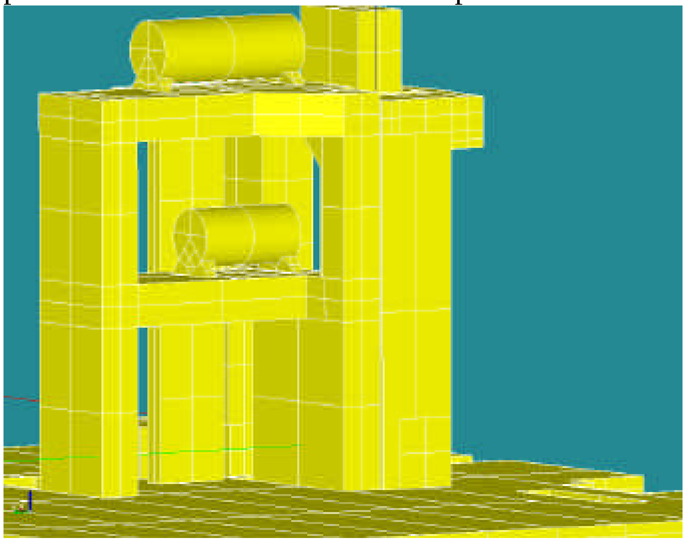

Figure 11 View of the geometry from the southeast

Figure 12 shows the calculated frequency response function of this model, while Figure 13, Figure 14, Figure 15, and Figure 16 show the deformed shapes.
These changes resulted in an adequate design, where the yawing is 30% higher than the operating frequencies of the motors on the pedestal. It is still within 10% of the bottom wire return roll operating speed, but since that motor is on the floor, it is near an anti-node and has a harder time exciting vibrations.

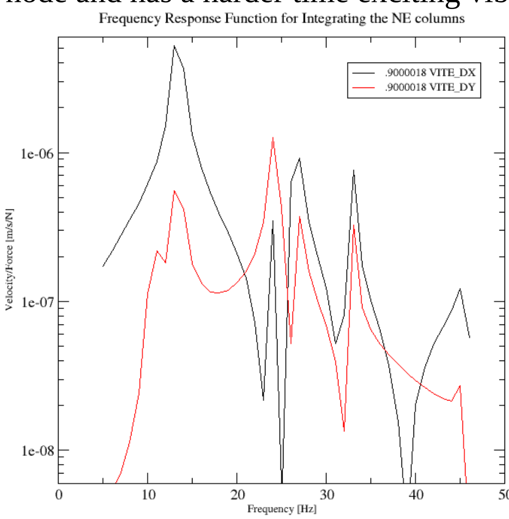

Figure 12 Frequency response function for the columns tied together

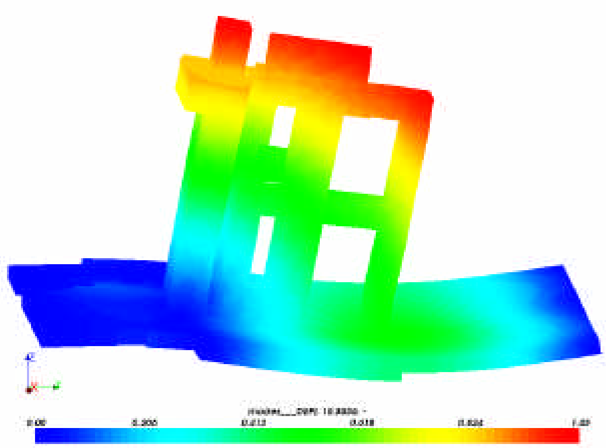

Figure 13 CD rocking at 10.5 Hz for the columns tied together

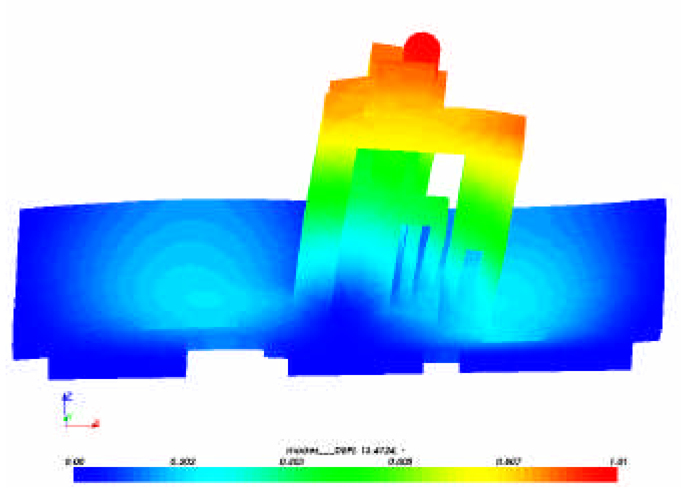

Figure 14 MD rocking at 13.4 Hz for the columns tied together

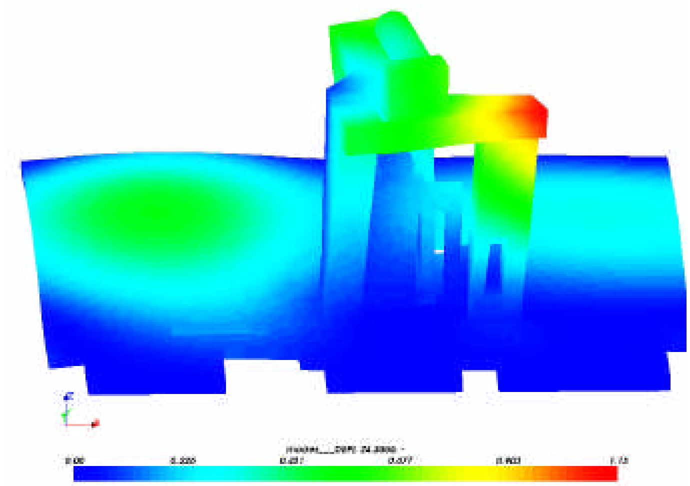

Figure 15 Yawing at 24.3 Hz about NE column

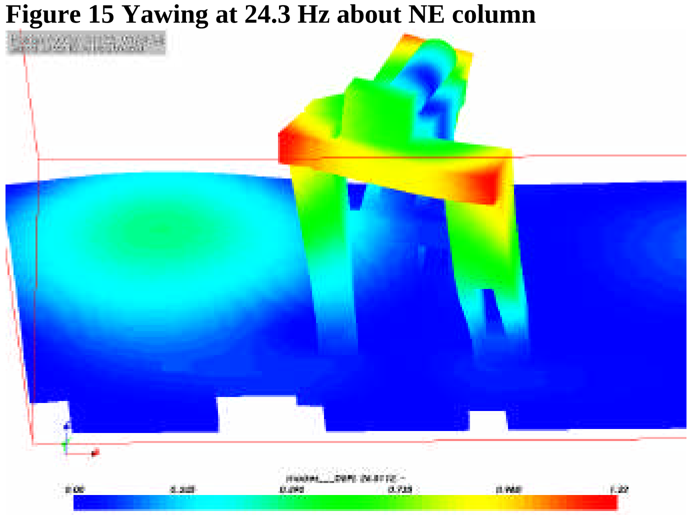

Figure 16 Yawing at 26.5 Hz about the center of the motor

Improvement Possible

This design was deemed adequate, but since Figure 15 showed a yawing about the new column, one final design modification was tried as shown in Figure 17. The existing column towards the wet end was increased in width by 500 mm. This gave a good improvement as seen in Figure 18, where the yawing resonant frequency increased to 26.5 Hz.

When this design was checked for construction feasibility, it was found that the extended column would prevent the electricians from changing the brushes on the couch motor mounted on the first platform. Since the previous design was adequate, it was the design selected. However, if the machine speed will be increased in the future, and the structure's resonant frequency needs to be increased, then we have a basis for selecting a design alternative.

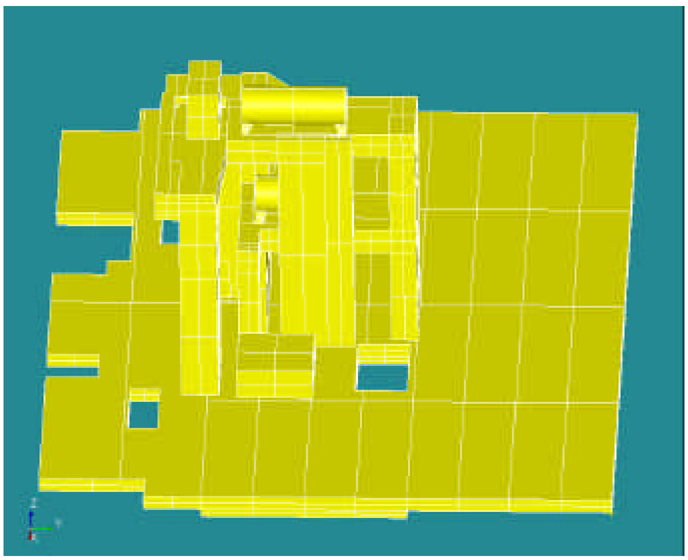

Figure 17 View of geometry from the West showing the extension of the existing west column to the South

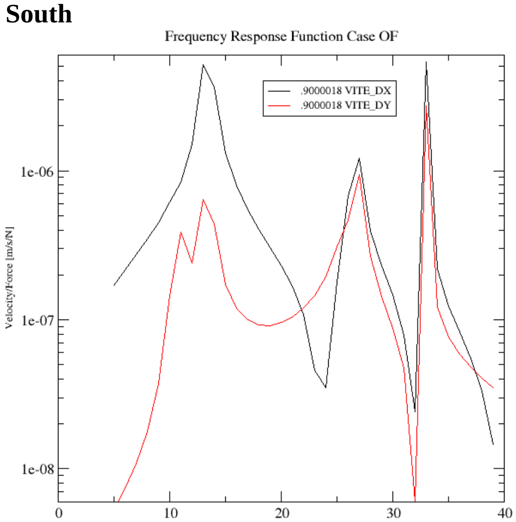

Figure 18 Frequency response function for the columns tied together

Stress Levels

The structural design engineer wanted to see the stress levels in the model. Since the model was already built, it was relatively simple to change it from performing a dynamic solution to a static solution and obtaining the static deformation along with the stress levels. The results were compared at a number of mesh sizes and for the static analysis, a mesh size of 100 mm was chosen.

Initially the forces at the nodes representing the columns that support the floor were determined to see if they looked reasonable. As seen in Figure 19, the only net forces on the structure are the z-direction (vertical) forces. The forces in the other directions cancel out, just as we would expect.

A steel plate was added to the extended pedestal to give it sufficient dynamic stiffness where the pedestal base went around an existing building and crane support column. The stress levels between this plate and the concrete was determined to give the required information for the structural engineer to design the shear supports for this plate, with the shear shown in Figure 20.

The deformation of the resulting structure is shown in Figure 21. From these results, the stress was also determined and was quite low.

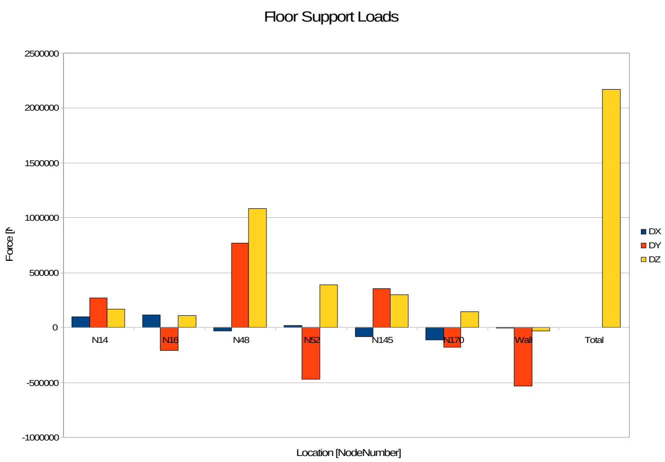

Figure 19 The forces at the support locations

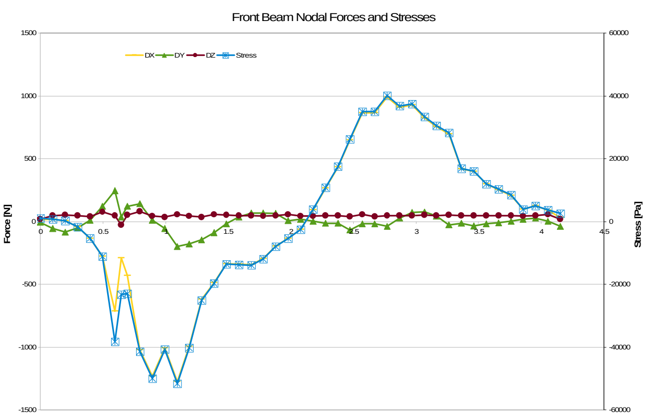

Figure 20 Nodal Forces and stress level on the plate at the bottom of the 600 mm thick portion of the motor base extension

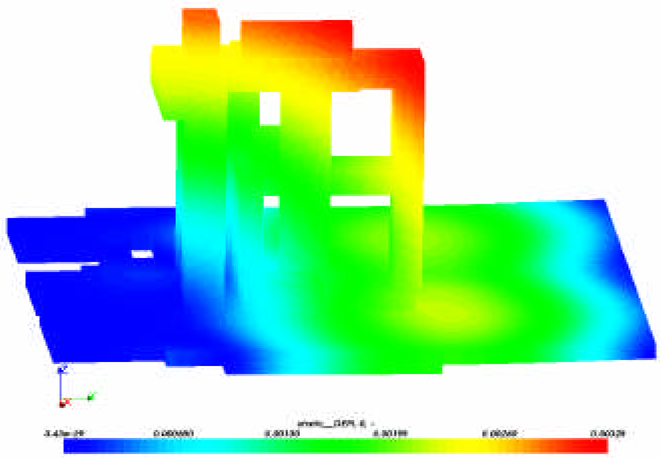

Figure 21 Static deformation of motor pedestal and floor

Conclusion

When there is a redesign that has large changes in mass or stiffness, vibration is always a consideration. This is particularly true for pedestals. Making changes without evaluating the dynamic effects can lead to disastrous results. The ideal way to design effectively is to test the existing structure to determine its resonant frequencies, and then to build a model and calibrate it to match the test frequencies. This model can then be used to predict the resonant frequencies of the modified structure.

In this project many design iterations were modeled to find a design that would both physically fit in the cramped space and that would allow the new motor and gearbox to operate without exciting a structural resonance.