Engine pedestal design – case study and lessons learned

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Abstract

This paper presents a case study of high vibration on an engine-driven reciprocating compressor package. The typical design practice for controlling engine vibration is to avoid resonance of the package (engine, driven equipment and supporting structure) by ensuring package natural frequencies do not align with key engine orders. The usual design practice is to avoid 1x and 2x engine speed. The package in this case experienced high vibration at 1.5x operating speed. The vibration was unusual in that the shape of the vibration was characterized as twisting about a vertical axis through the middle of the engine. Typically, the operating deflection shape is described as horizontal motion of the engine with the drive end and non-drive end moving in the same direction. The package vibration problem described in this paper is unique in the authors' experience and should be of interest to others involved in the design and specification of engine-driven compressor packages.

The paper presents measurements from the initial site investigation, results from the finite element analysis to simulate the field installation, design modifications and measurements after modifications were implemented. It also provides a summary of recommended design practices to avoid similar engine vibration problems on a new design.

Introduction

Many design standards and guidelines give guidance to minimize vibration on reciprocating compressor packages. Most of the emphasis is placed on the compressor, and rightly so, as it generates many dynamic forces that must be considered to allow a safe, reliable system.

Reciprocating engines that commonly drive the compressors are also sources of vibration that must be accounted for in the design of the compressor package. Guidance for skid-mounted engine design is given by Smalley[1,2], Harper[3] and the GMRC Package Guidelines[4]. Reciprocating engines typically generate vibration in the horizontal direction, where horizontal direction refers to perpendicular to the crankshaft and parallel to the ground. The horizontal direction is defined as perpendicular to the crankshaft axis in a horizontal plane. The engine moves as a uniform rigid body with the drive end and non-drive end of the engine moving together or in-phase.

Design studies are often done for the package to ensure the pedestal for the compressor and engine is suitable for the static and dynamic loads. Significant effort is put into generating detailed finite element models of the skid beams and local reinforcement from gussets and other structural members. Improvements in CAD and finite element software has allowed for including comprehensive models of the compressor to simulate the complex dynamics within the compressor, the interaction between the compressor and bottles, and the interaction between the compressor, skid structure and foundation. Experience with previous studies has shown that representing the engine or motor as a simple rectangle block representing the translational and rotational inertia is sufficient. The key characteristic to simulate was the flexibility of the pedestal and skid to which the engine was mounted. The case study presented in this paper represents a unique problem that indicates a deficiency in the typical industry approach for simulating vibration on reciprocating compressor packages.

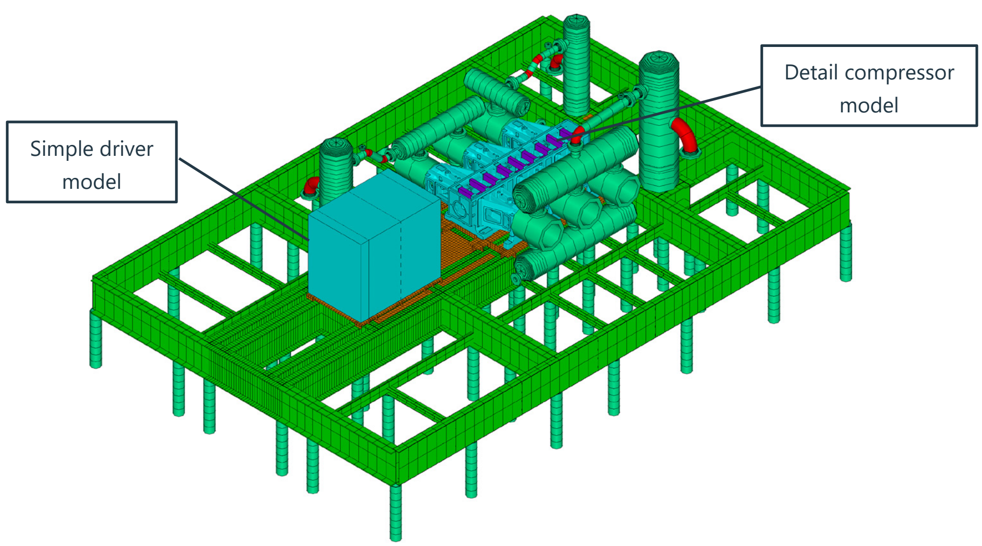


Figure 1: Package finite element model

Problem description

Williams had commissioned four new compressor packages that included Ariel KBZ/6 compressors driven by CAT 3616 engines, as shown in Figure 3. The engine and some of the associated piping and structure for the engine exhaust, coolant and fuel gas piping were reported to have high vibration.

Figure 2: Photo of engine and compressor package – left side

Figure 3: Photo of engine and compressor package – right side

An initial assessment of vibration over a speed range of 800 to 1,000 rpm was conducted. Vibration was measured at the engine crankshaft height at the drive end and non-drive end. Figure 4 shows a sample of the vibration spectrum at the engine non-drive end. The vibration at 1x operating speed is quite low, approximately 0.2 ips pk. The highest vibration was about 0.8 ips pk at 25 Hz. This vibration amplitude is quite high compared to industry guidelines. The frequency of the vibration is also quite unusual with 25 Hz corresponding to 1.5x the operating speed of 1,000 rpm. The engine operates on a four-stroke cycle which generates gas torques and dynamic forces at half orders of engine speed (Pulkrabek[5]). Typically, high half-order vibration is the result of the engine not operating properly (misfire or timing problems). The usual approach is to tune the engine to ensure it is operating properly. As the engines were just commissioned and engine tuning verified to be within specifications, further investigation was necessary.

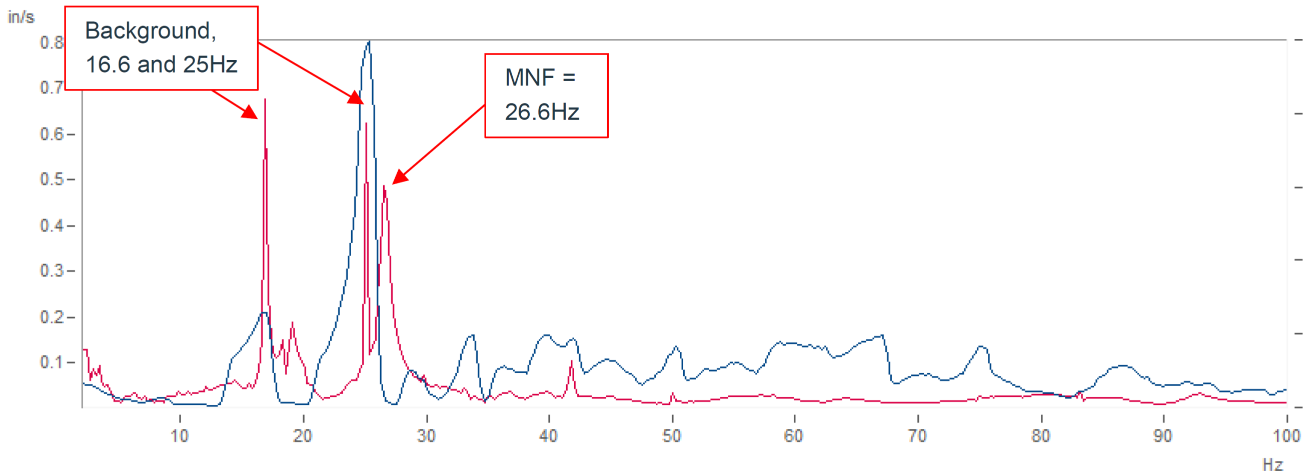


Figure 4 Engine non-drive end vibration (blue) with impact test (red)

An impact test was done on the engine to measure the mechanical natural frequency. One of the impact test measurements is overlaid on the vibration trace in Figure 4. The impact test showed peaks in the measured response at 16.6 Hz, 25 Hz and 26.6 Hz. The results at 16.6 Hz and 25 Hz were determined to be background vibration from other engines operating in the same building. The 26.6 Hz measurement was determined to be the true mechanical natural frequency of the engine. It appeared that the mechanical natural frequency at 26.6 Hz was close enough to the excitation for normal engine operation at 25 Hz to result in high vibration. This high amplitude is not typically seen but is reasonable to expect as the engine does generate significant dynamic torques and forces at 1.5x engine speed even when the engine is operating smoothly.

The next step to understanding the package vibrations measured at the engine was to measure the operating deflection shape. A multi-channel data acquisition system was used to collect horizontal and vertical vibration data on the skid, pedestal and engine. Figure 5 shows a photo of one configuration of the velometers on the engine drive end. Figure 6 is an image of the CAD model showing some of the other engine test points. Vibration amplitude and phase were extracted from the test data for the engine operating at 1,000 rpm. The results from the operating deflection shape are shown in Figure 7. Several important results are shown in these measurements:

1. The non-drive end and drive end are vibrating in opposite directions. The engine vibration at the middle mounting feet is low. The engine is twisting about a vertical axis near the middle mounting feet. This operating deflection shape is quite unusual. The usual operating deflection shape is one where the engine moves in-phase (Rivera et al. [6])
2. The measurements show significant flexibility in the pedestal at the connection to the main skid as shown by the kink in the deflection shape between 5 and 6. There is also some flexibility in the chock (2 to 3) and in the engine block and mounting foot (1 to 2).
3. The engine is well connected to the foundation. Horizontal and vertical vibration on the skid beams and foundation were very low amplitude and similar phase.

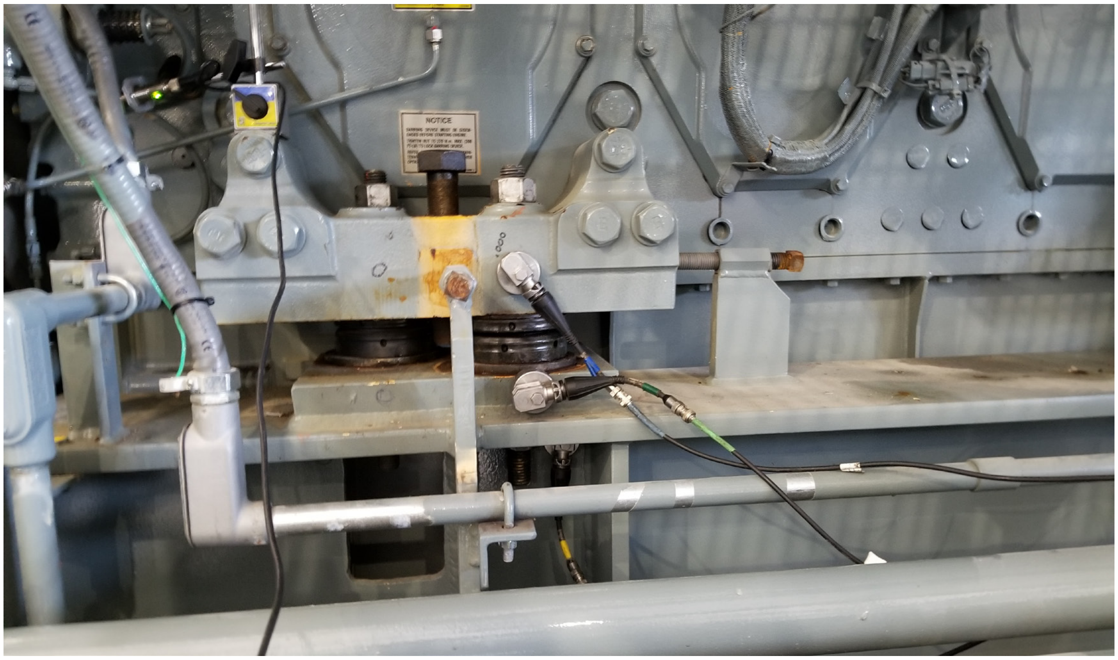


Figure 5: Vibration test points on engine and pedestal (drive end LHS)

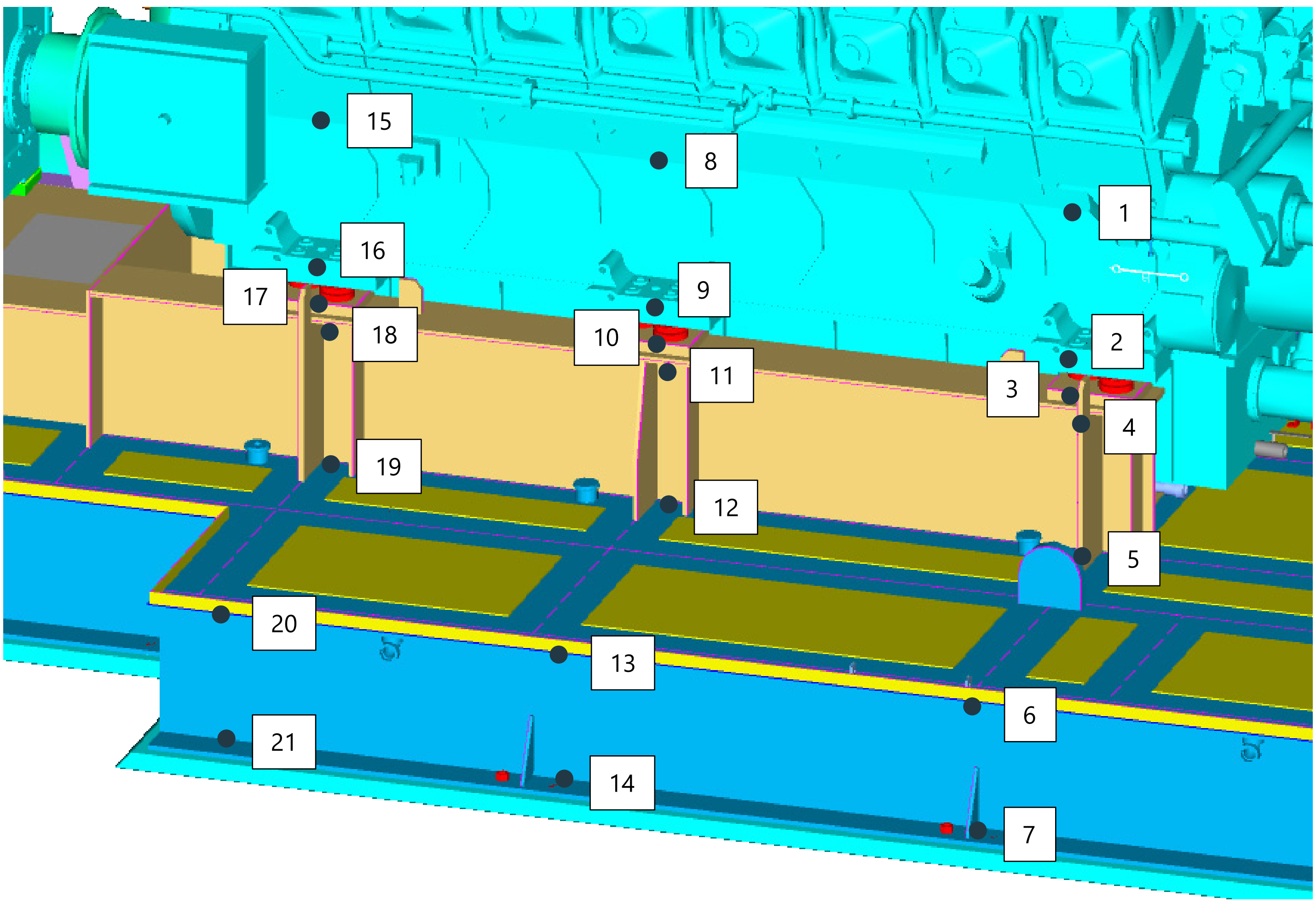


Figure 6: Vibration test points on engine and pedestal

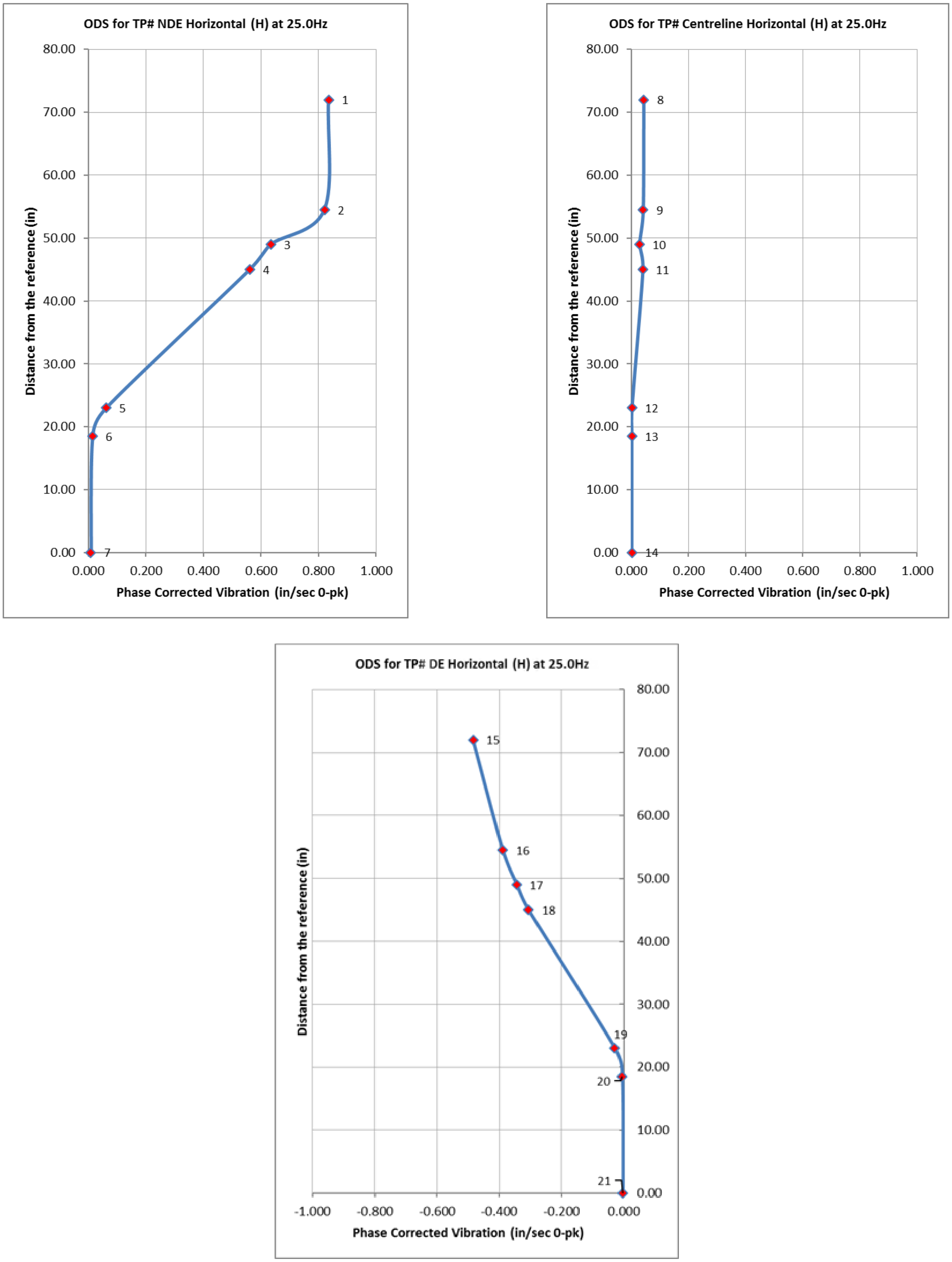


Figure 7: Operating deflection shape at 1,000 rpm

Analysis

The field vibration measurements and mechanical natural frequencies indicated that stiffening of the pedestal and/or engine connection to the pedestal would be required to reduce the engine vibration. The area around the engine pedestal is quite congested, so adding stiffening would be difficult. It was decided to use a finite element model of the package to determine the required modifications to the existing packages. A finite element model of the skid and pedestal was created using shell and solid elements so the local flexibility of the structural components could be calculated. The compressor and engine were simulated as beam elements to represent the mass and general stiffness of machinery accurately. A figure of the finite element model is shown in Figure 8.

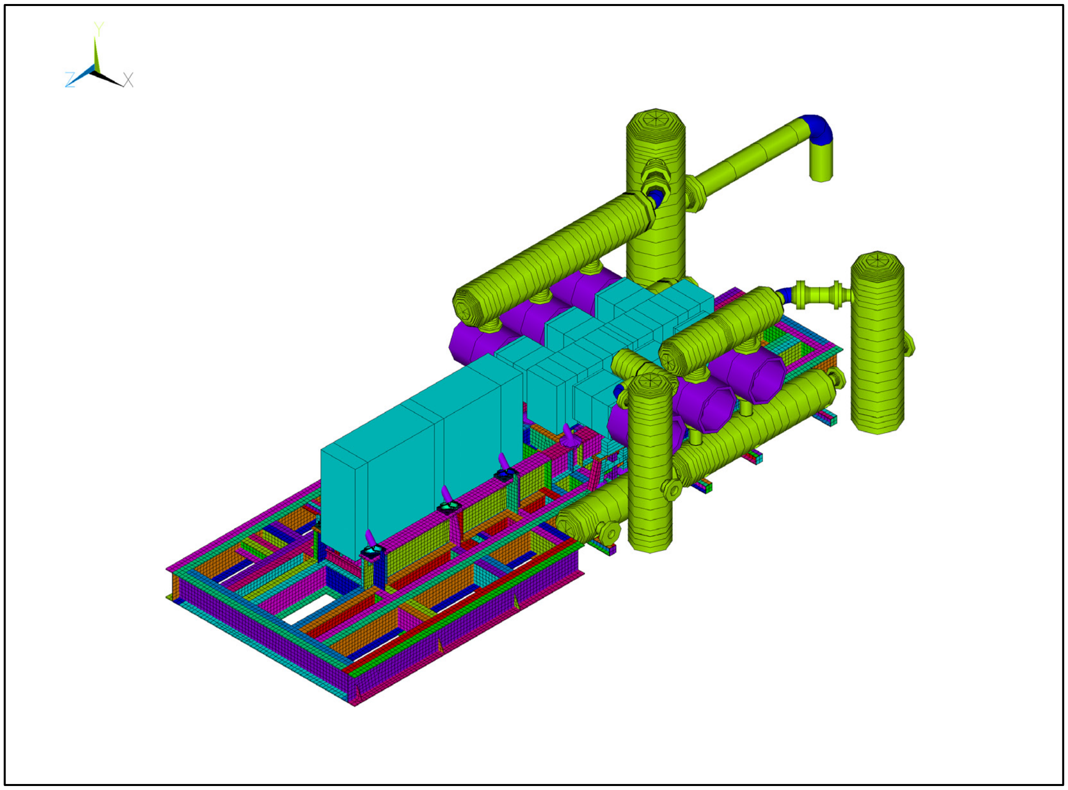


Figure 8: Finite element model of the existing compressor package

The first step in the analysis was to compare the mechanical natural frequency of the model versus the field measurement. The modal analysis determined the twisting mode about a vertical axis, as demonstrated in Figure 7, to be approximately 48.2 Hz. An image from the finite element model demonstrating the modal response twisting mode is shown in Figure 9 and Figure 10.

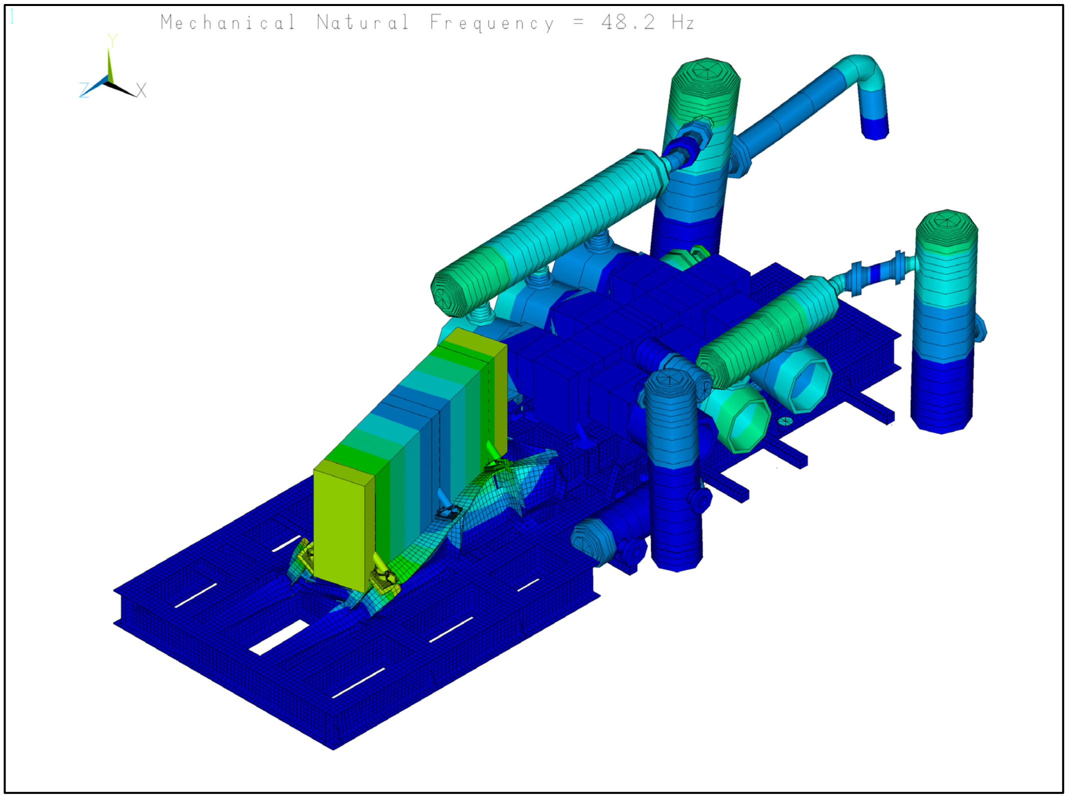


Figure 9: Existing package model original modal result – isometric view

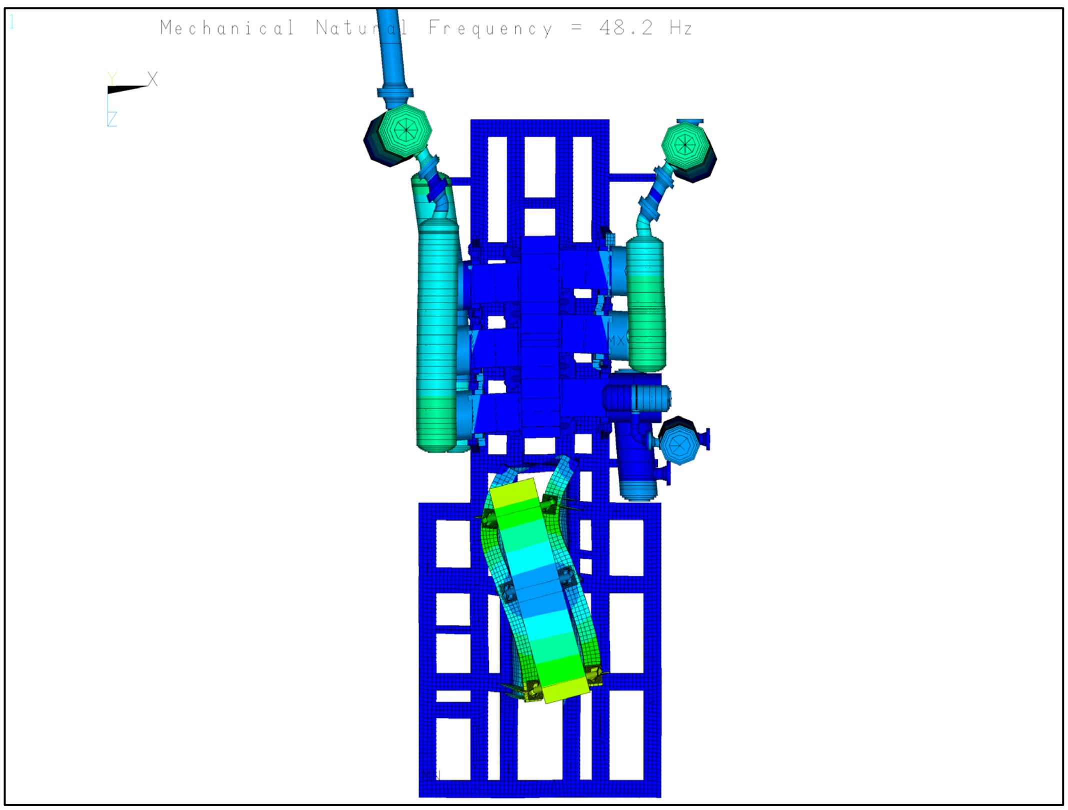


Figure 10: Existing package model original modal result – top view

This modal result demonstrates that the existing package model was too stiff and did not accurately represent the engine twisting response. It is clear from the field measurements that the engine flexibility and flexibility of the engine connection of the pedestal will have a significant influence on the dynamic response. The severity of the vibration problem and risk to the engine and package reliability were such that a simplified approach to simulating the engine and engine support was required to expedite the analysis. The flexibility of the engine and engine support could be adjusted such that the mechanical natural frequency and mode shape of the model matched the results from the field testing.

Several iterations of the model were evaluated. Refinements were made to adjust the flexibility of the beam model for the engine as well as adjust the stiffness of the connection of the engine to the pedestal. There are many parameters to adjust which may result in a twisting mode of the engine at the measured mechanical natural frequency. Judgment was required to review the modal results to achieve a representative model. The result of this calibration step yielded the mechanical natural frequency of 25. 9 Hz, as shown in the finite element model mode shape of Figure 11.

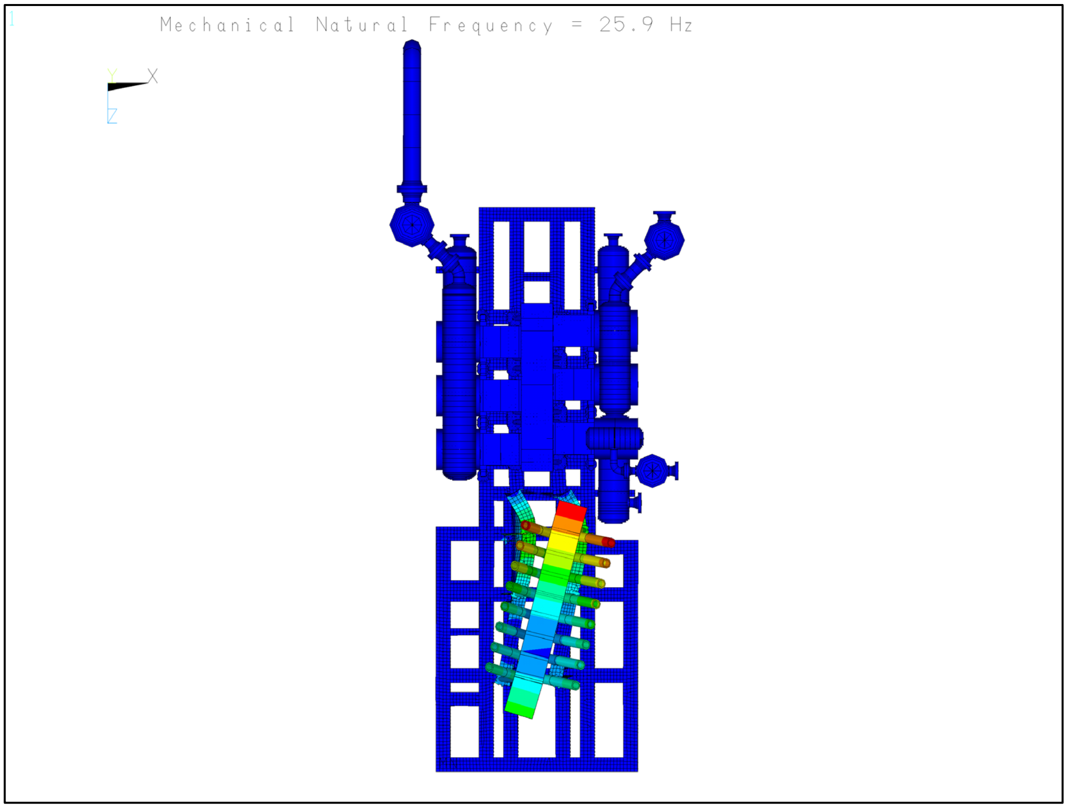


Figure 11: Plan view of engine twisting mode

The next phase of the analysis was to determine a solution to reduce the engine vibration. The typical design approach would be to increase the mechanical natural frequency above 2x operating speed. Initial calculations showed that a significant amount of stiffness would need to be added to the pedestal. Tuning the mechanical natural frequency above 2x operating speed was not possible due to the complexity and limitations of adding pedestal stiffening given the very congested areas adjacent to the engine.

An alternative approach is to increase the mechanical natural frequency to avoid 1.5x. Figure 4 shows that the current system has a mechanical natural frequency just above 1.5x, 26.6 Hz versus 25 Hz. The high vibration is the result of the engine operating on the flank of the resonance curve. Moving the resonance 5% to 10% higher would avoid amplification of the 1.5x forces. However, as shown in Figure 12, the resonance and ±10% interval will overlap with 2x operating speed indicating the potential for exciting high engine vibration as well. The engine will primarily operate at 1,000 rpm, so there is a small frequency interval at about 28 Hz to 30 Hz which may be acceptable to tune the engine mechanical natural frequency.

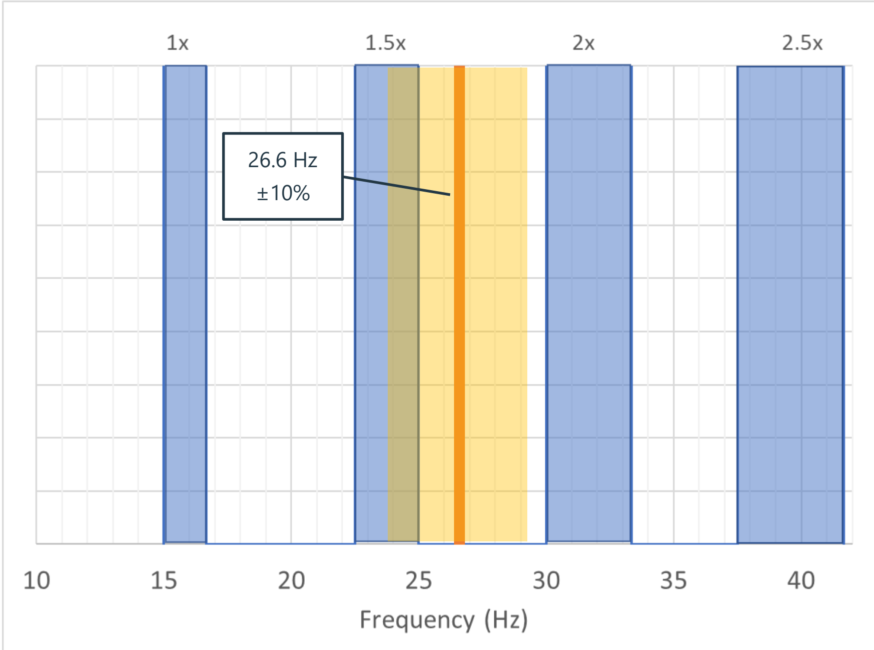


Figure 12: Harmonic intervals for 900 to 1000 rpm

The other factor that must be considered are the engine dynamic forces that will excite the resonance. The combustion cycle of a four-stroke engine generates gas forces similar to those in a reciprocating compressor. The highest force is in the direction parallel to the direction of the piston motion (Figure 13). A force is generated in the gas volume between the piston and head that acts on the head. An equal and opposite force acts on the crankshaft. A force transverse to the piston direction of motion that is analogous to the crosshead guide force is generated as a result of converting the linear motion of the piston into rotational motion of the crankshaft. A transverse force is created at the piston interaction with the liner as a well as an equal and opposite force on the crankshaft. These forces occur at all half orders of engine speed. Figure 14 is a chart comparing the inline and transverse gas forces. The 1.5x engine forces generated due to the normal combustion process are also higher than the 2x engine forces. The inline forces and transverse forces are reduced by 60% and 20% respectively. All things being equal, the engine vibration at 2x would be expected to be lower than 1.5x. There is a small mechanical unbalance force and couple at 2x that does not occur at 1.5x, but the 2x unbalance is much smaller than the inline and transverse forces.

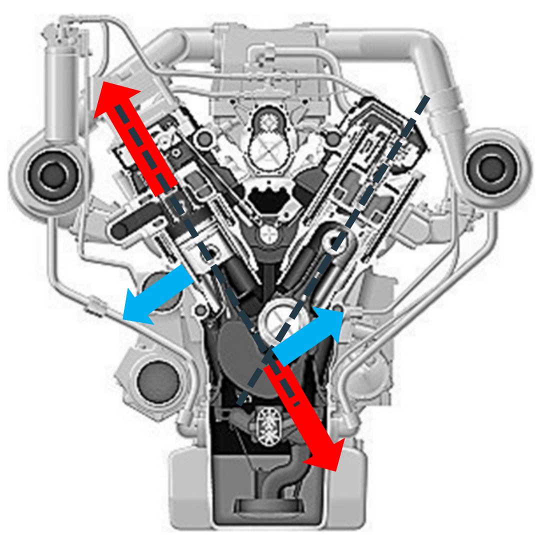


Figure 13: Gas forces acting on one piston

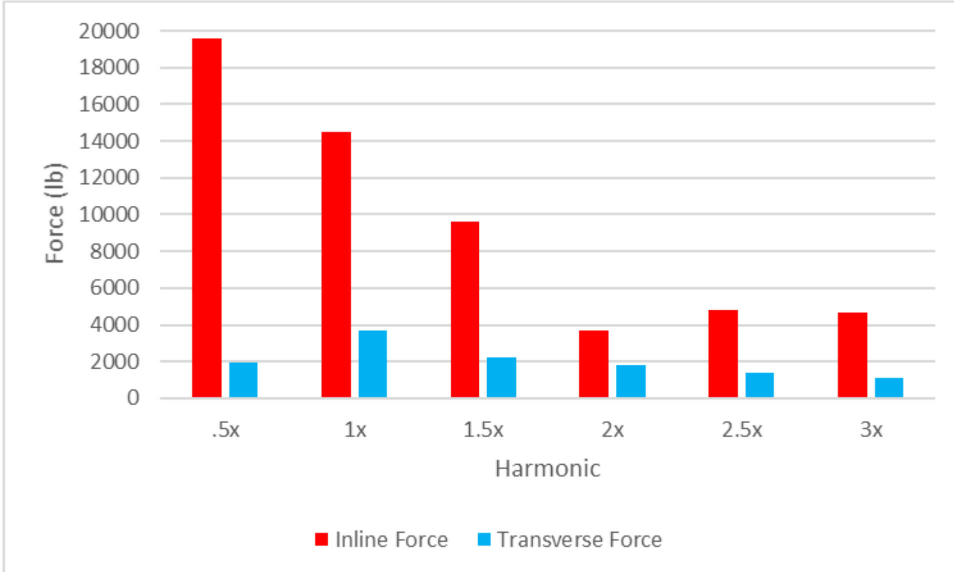


Figure 14: Piston forces

Modifications to the engine pedestal were evaluated with the finite element model to determine design options that would tune the engine mechanical natural frequency to approximately 28 Hz. The design options were reviewed with the fabricator who will be implementing the changes. The sides of the pedestal are quite congested with tubing and auxiliary piping, particularly on the right-hand side. Installing 2" square structural tube braces, as shown in Figure 15 were found to be an effective method of increasing the mechanical natural frequency while minimizing interference with existing equipment and impeding access in the future.

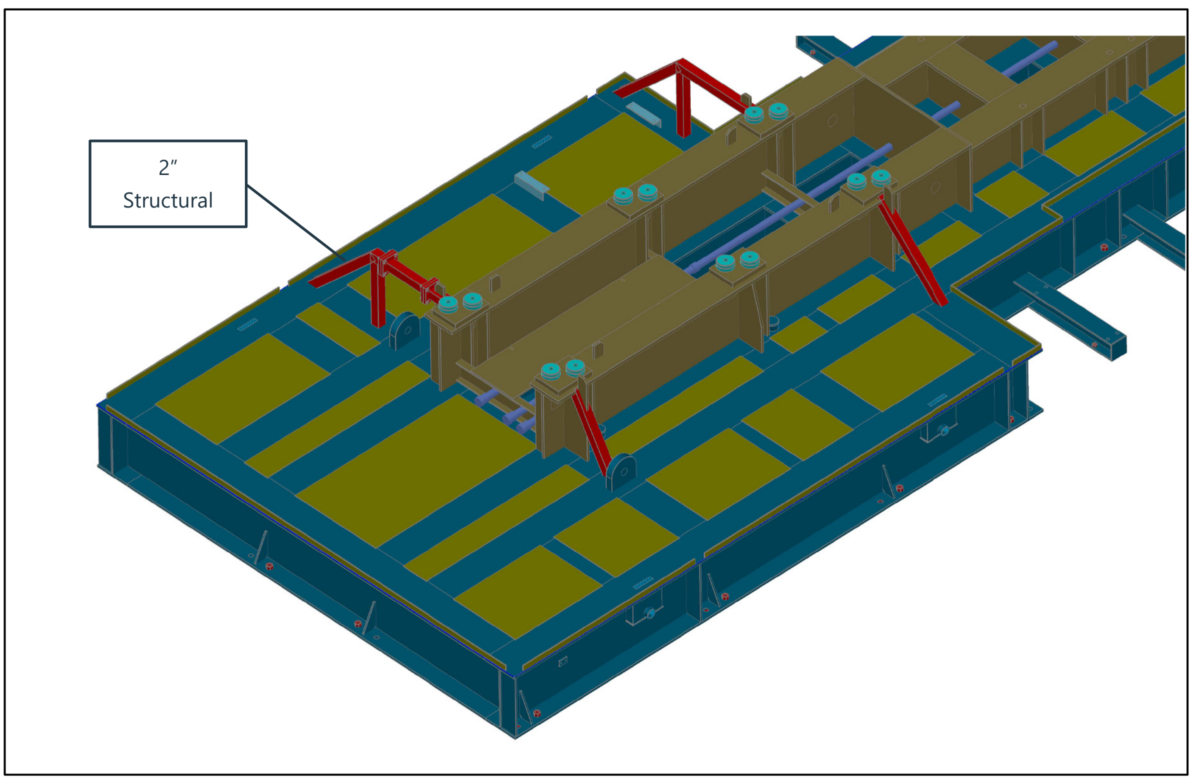


Figure 15: Pedestal braces

The calculated mechanical natural frequency for the engine twisting mode about a vertical axis is 28.1 Hz, as shown in Figure 16. This frequency corresponds to approximately 850 rpm at 2x engine speed. This mode is tuned to the optimal range, as previously discussed.

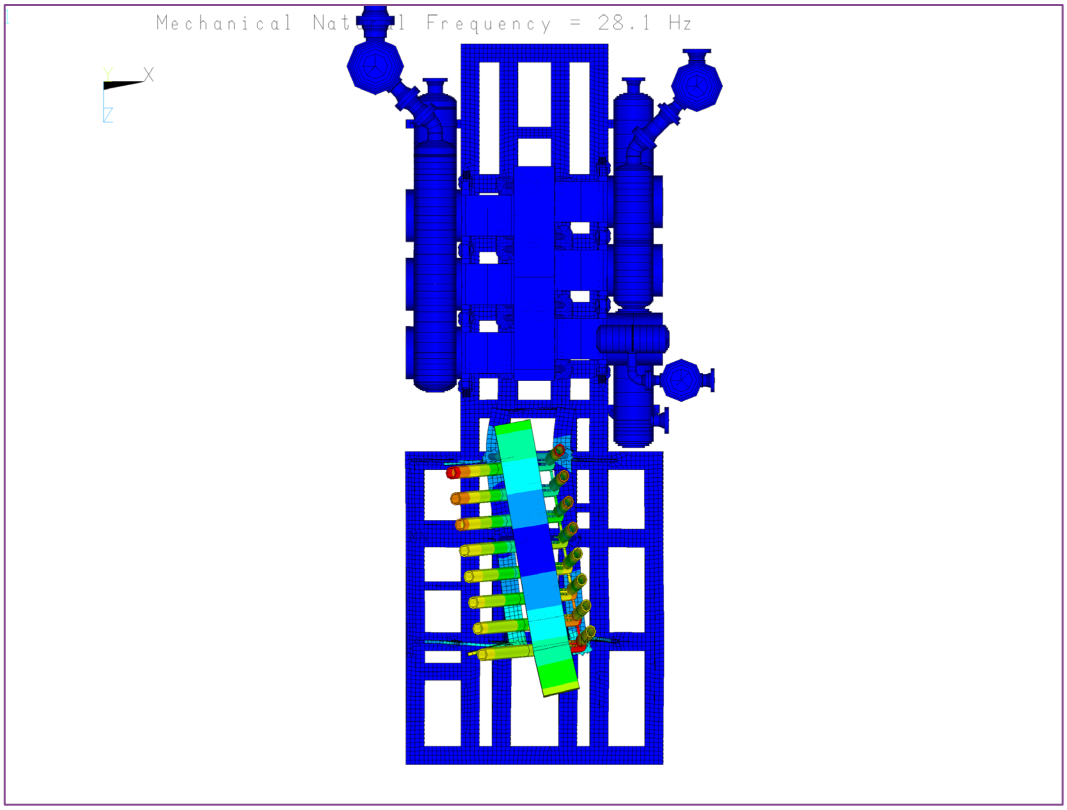
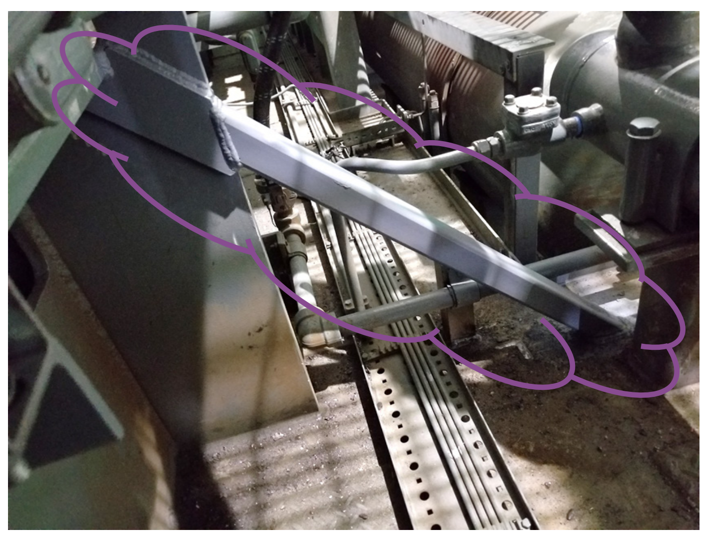


Figure 16 Modal result with braces installed

Field implementation and follow-up testing

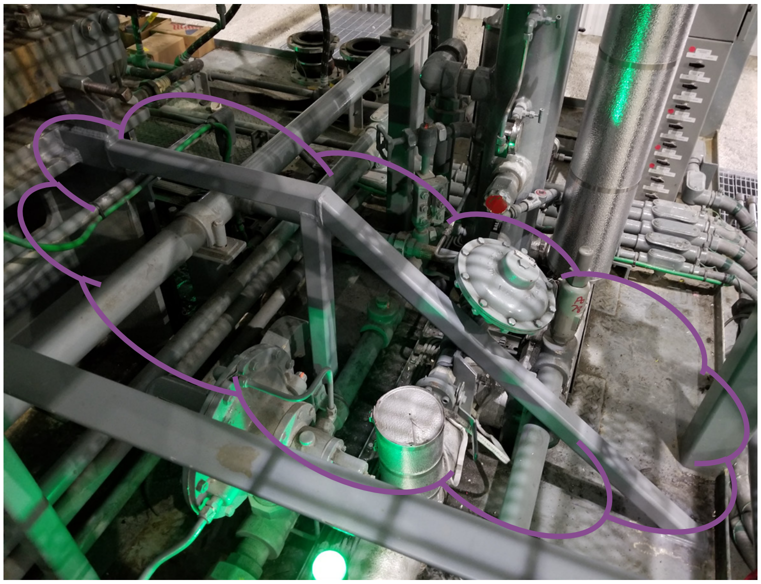
One of the four units at site were modified with pedestal bracing, as shown in Figure 15. See photos of the installed pedestal braces in Figure 17 and Figure 18.

LHS NDE pedestal brace



LHS DE pedestal brace

Figure 17: LHS pedestal braces



RHS NDE pedestal brace RHS DE pedestal brace

Figure 18: RHS pedestal braces

Testing was conducted on the modified pedestals to determine the mechanical natural frequencies and vibrations. The key measurements summarizing the impact test are given in Figure 19. The blue trace represents vibration measurement on the drive end of the engine with an impact on the non-drive end. The red trace is the vibration measurement and impact on the non-drive end. The transfer function indicates a resonance around 28.8 Hz. This result is very close to the model-calculated mechanical natural frequency of 28.1 Hz. The phase difference between the ends of the engine is approximately 172º, so the ends are moving out of phase. The coherence is a measure of the quality of the relationship between the input and output signals. A coherence of 1.0 means the output signal is purely from the input signal. The calculated coherence is 0.98 indicating the signal is a good measurement and indicative of the actual natural frequency.

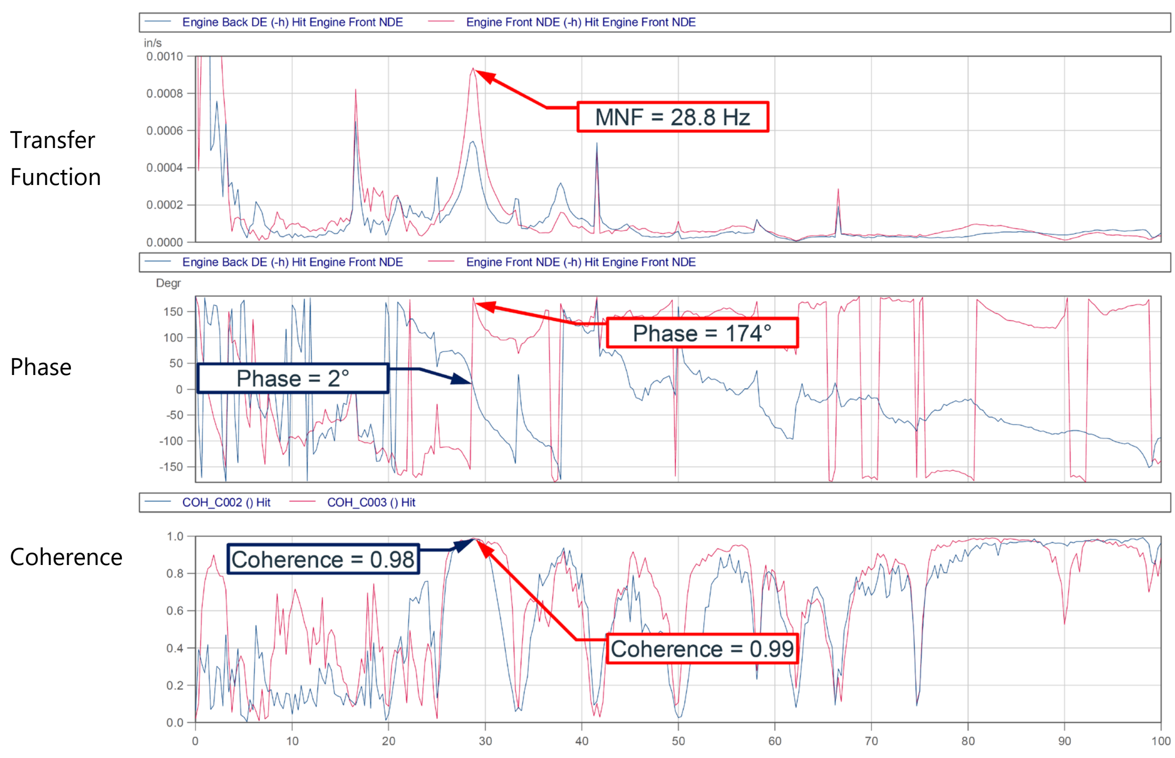


Figure 19: Impact test measurements

The vibration measurements on the drive end and non-drive end of the engine at the crankshaft centerline height in the horizontal direction are shown in Figure 20 and Figure 21. The dark blue trace is the vibration amplitude. The red trace is the Caterpillar vibration guideline. The light blue trace is the transfer function amplitude from the impact test. The vibrations results are a maximum at a frequency of 28.8 Hz in the 2x operating speed range corresponding to the engine mechanical natural frequency. The maximum vibration occurs at 28.8 Hz, which corresponds to an operating speed of 864 rpm. The vibration for the actual operating speed range of 900 to 1,000 rpm as demonstrated by the shaded area from 30 Hz to 33.3 Hz is acceptable and less than the Caterpillar guideline.

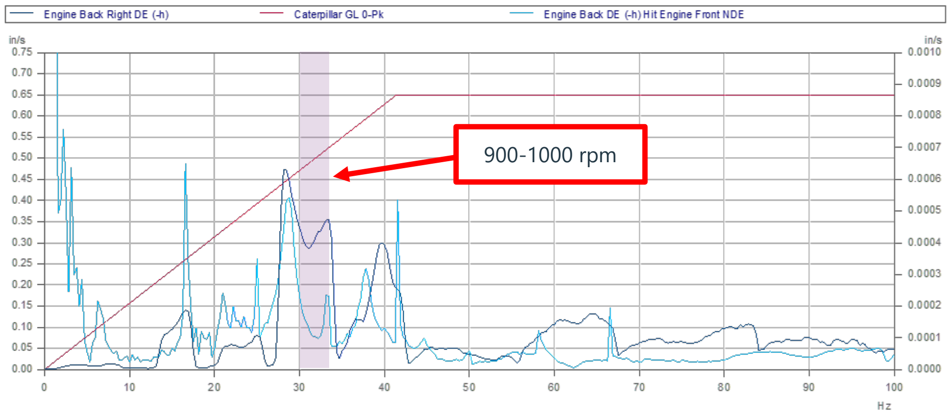


Figure 20: Drive end vibration and impact test results

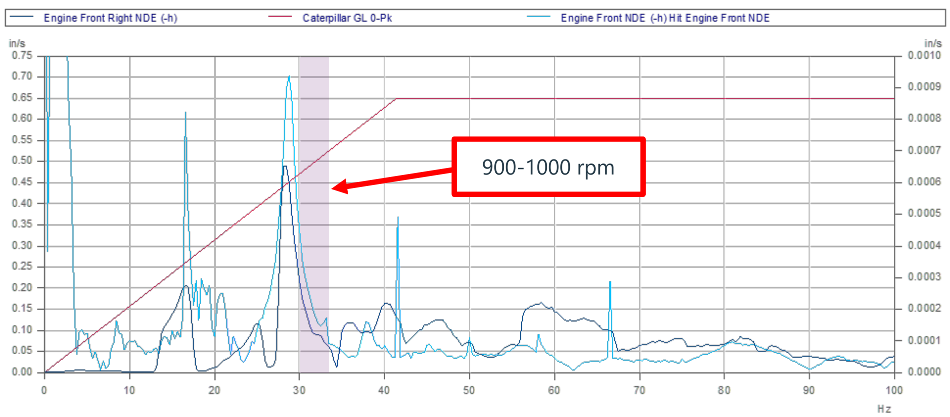


Figure 21: Non-drive end vibration and impact test results

The field testing demonstrated the pedestal modifications were effective in reducing the engine and pedestal vibration to acceptable levels. The recommendations could then be implemented to other existing units that had the same design. New units of identical design were to be fabricated in the future. The pedestal braces, although effective, are not the best option as they limit accessibility and maintenance activities. Additional analysis was done to determine alternative design modifications that would be less obtrusive. Several changes to the pedestal were evaluated, such as using beams with thicker flanges and webs and more gussets. The best option from the fabricator's perspective is to add gussets, as shown in Figure 22, which results in a similar engine twisting mode mechanical natural frequency as the 2" square tube braces. The new design was incorporated in more than eight new units to date and proven to result in acceptable vibration around the engine.

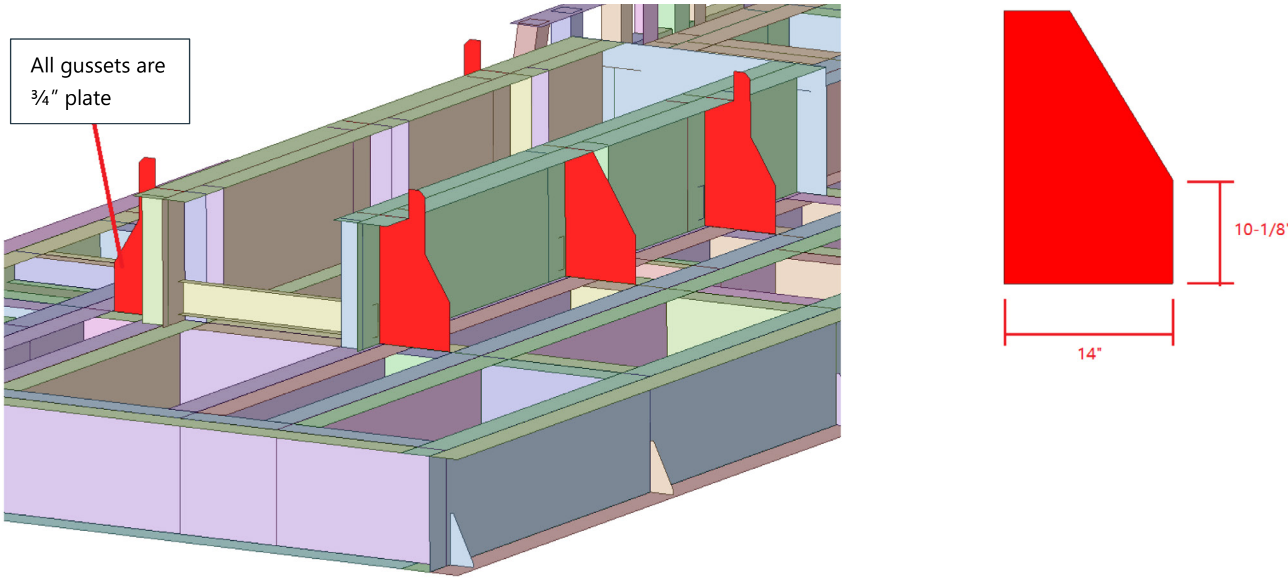


Figure 22: New package pedestal stiffening

Engine modelling

The work done for this case study showed that calibration of the model was necessary to get agreement between the real system and the finite element model. The model required calibration of the engine flexibility, which includes flexing of the engine block as well as the engine mounts used to connect the engine to the chocks. This calibration step is appropriate when measurements of the engine dynamics are available. However, the question arises as to how designers are to manage new skid and pedestal designs or other engine designs.

The engine is a quite complicated component from a structural dynamics perspective. Simulating the dynamic response of the engine structure with the skid and other components on the package was not a scope of work that could be practically done as part of a compressor package design study. The complexity of the engine results in a finite element model that includes hundreds of thousands to millions of nodes and elements. Solving models of this size was not practical as the file sizes and simulation times were much too large. Also, calculating mechanical natural frequencies and vibrations required running modal and harmonic simulations with multiple load cases, increasing file sizes and solve times further as compared to static simulations. Modern finite element solvers in combination with modern operating systems, multi-core solvers, and inexpensive high-capacity memory and data storage have overcome these practical constraints. Simulations of this type that once took one to two days can now be done in two to four hours with affordable computer platforms. The common industry practice of simplifying the engine to a beam model or simple block representation is no longer necessary based on the argument of model complexity and size and time constraints.

The other hurdle that must be overcome is lack of available high-quality engine geometry to support the creation of an appropriately detailed package finite element model. Ideally, a 3D solid model of the engine would be made available to fulfill this need. However, such detailed models may not be readily available for all engines that may be encountered for a number of reasons, including the age of the engine design or the proprietary details of the design that such a model may inadvertently make public. For older engines, producing such models may require a significant investment of time in constructing the model from the existing 2D drawings. Old or new, highly detailed models contain a lot of hard-earned technical knowledge and intellectual property that the engine manufacturer is rightly cautious about sharing openly. For the most part, de-featured solid models can remove details not significant to the structural dynamics of the model. Such a de-featured or simplified model is suitable for this type of analysis.

Adding to the difficulty of availability of a 3D solid model, not all 3D models are compatible across all finite element analysis platforms, and critical details had been lost or altered in the past when using a translator to produce a usable model. Fortunately, significant work has been done to develop platform-neutral formats (such as STEP, ACIS, or Parasolid) that are easily transferred among platforms, largely overcoming this obstacle.

This situation of requiring a more comprehensive engine model became clear for this project soon after the field study had been completed and the initial finite element analysis was done. The flexibility of the engine and engine mounts had a significant influence on the engine dynamic response. Discussions between the parties involved in this project resulted in Caterpillar supplying a defeatured finite element of the engine that could be used for the analysis. The model is quite large, with more than 500,000 nodes and over 100,000 elements. The model was supplied in a Nastran file format. The model was imported into the ANSYS Workbench application with only a few mouse clicks with the only problem being a required redefinition of a few material properties. A model of the engine mounts was added to the generic engine model supplied by Caterpillar using with contact elements to connect the mounts to the block, as shown in Figure 23.

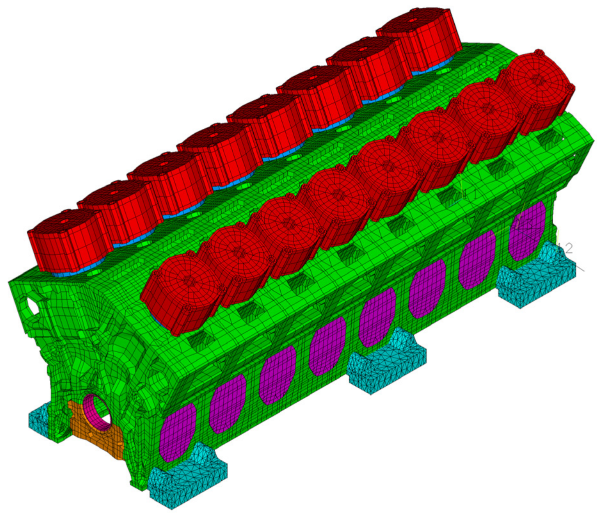


Figure 23: Engine finite element model

The need for determining a solution and availability of suitable field test information allowed for calibrating the finite element model to determine an appropriate design modification. The preferred approach to calibrating the finite element model is to include a more representative model of the engine and other components so reliable studies can be done in the design stage. Additional finite element analysis was done for the package after the field modifications had been made to determine if more accurate results could be achieved with the Caterpillar engine model. Figure 24 shows an image of the finite element model with the detailed engine model. The engine model includes the engine mounts and spring elements to simulate the adjustable steel chocks between the engine and pedestal.

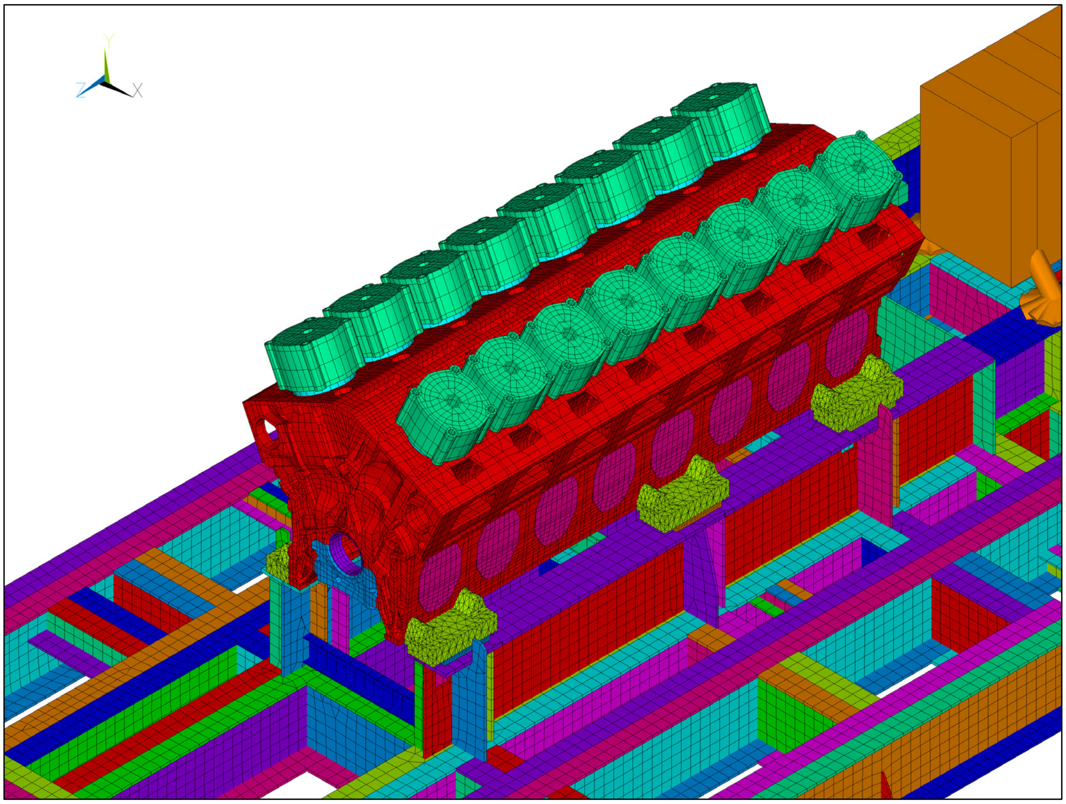


Figure 24 Package model with detailed engine

A modal analysis of the updated package model was conducted. The mode with the engine twisting about a vertical axis was calculated to be 31.1 Hz, as shown in Figure 26. Recall from the field measurements that twisting model mechanical natural frequency was measured to be 26.6 Hz, as shown in Figure 4. The mechanical natural frequency calculated with the finite element is very close to the field measurement with no calibration or adjustment to the model inputs. This result is quite impressive considering the relatively complex model with the engine, engine mounts, adjustable steel chocks and welded design of the pedestal and skid. Flexibility in the engine block and engine mounting feet play a significant role in the dynamic flexibility of the system.

Note there is a second mode of the engine at 28.9 Hz, as shown in Figure 25, in addition to the mode at 31.1 Hz. The 28.9 Hz mode has a twisting component to it but also a large translation component in the z-direction (axial direction). The mode at 31.1 Hz is a purer twisting mode about a vertical axis. The 31.1 Hz mode is more representative of the field measurement as there was low axial direction vibration on the engine. The 28.9 Hz mode, although close to 1.5x operating speed, is not excited by the engine operating forces.

The model results of 31.1 Hz are much closer to the original model of 48.2 Hz but are still 17% away from the field measurement or 26.5 Hz. Reducing the chock stiffness by about 40% and adding vertical flexibility at the skid grout connection reduces engine mode, so it is nearly identical to the field measurements (Figure 27).

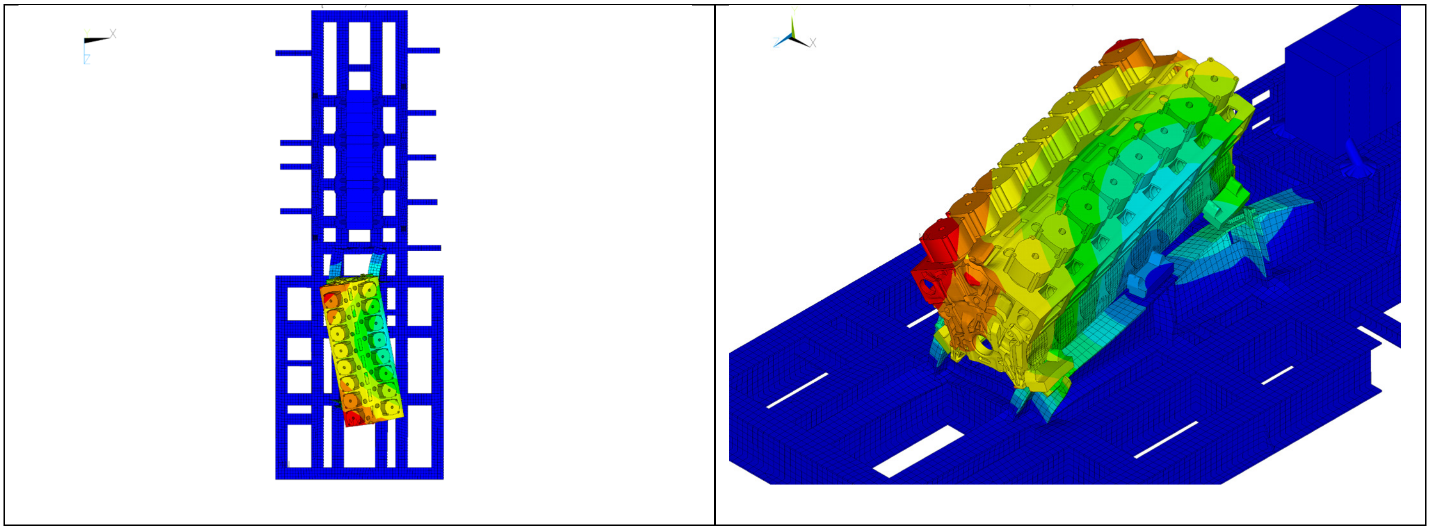


Figure 25: Modal result with detailed engine – 28.9 Hz

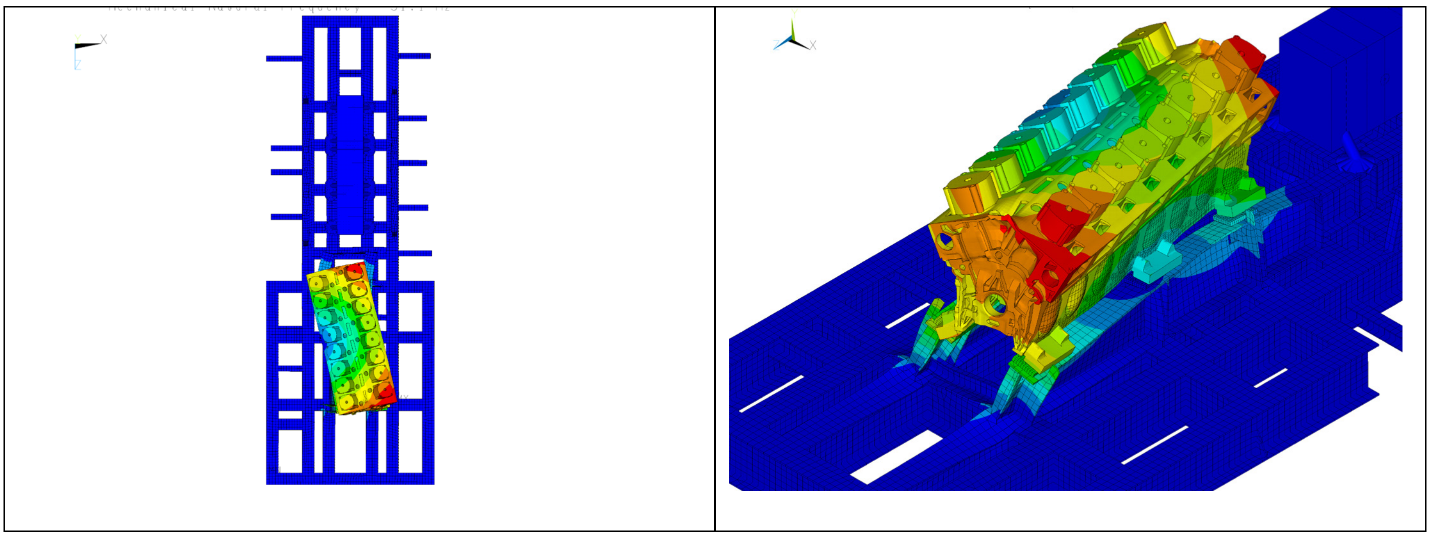


Figure 26: Modal result with detailed engine – 31.1 Hz

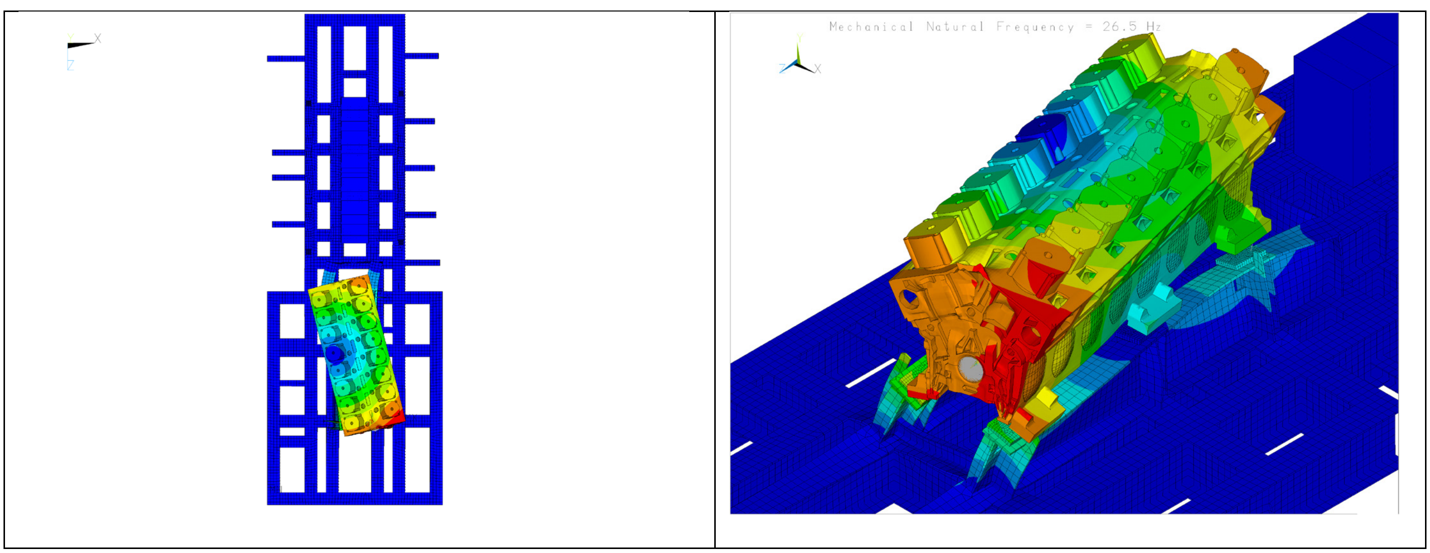


Figure 27: Twisting modal response with adjusted chock stiffness – 26.5 Hz

Engine vibration analysis

The analysis was taken a step further to calculate the vibration due to dynamic forces in the engine. The only forces acting on the engine at 1.5x would be the combustion forces, as described in Figure 13. The inline forces are applied to the head and crankshaft as two separate forces of the same amplitude and opposite direction. The transverse forces are applied as two forces, one on an area of the liner approximately equal to the piston position between top dead center and bottom dead center with equal amplitude and opposite direction force applied to the crankshaft. The location of the transverse force on the liner is a compromise. The real transverse force will be applied at different locations along the liner as the piston moves inward and outward with each cycle of the piston, so selecting a fixed location for the applied load is an approximation. The simulation conducted is a linear harmonic analysis that does not allow for changing the location of the force. A more complex time-based analysis could be done to better reflect the changing location of the force, however, such an approach is much more complex and computationally intensive.

The inline load applied on the head, and transverse load applied on the liner as simply applied to nodes on the engine finite element model. The loads to be applied to the crankshaft require a model of crankshaft. The purpose of the crankshaft is to only automate the transfer of the individual cylinder forces from the crankshaft to the engine block, so a complex model of the crankshaft is not required. A simple cylindrical shaft is used to model the crankshaft, as shown in Figure 28. The crankshaft is connected to the main bearing locations on the engine block webs using a collection springs with vertical and horizontal springs to simulate the bearing stiffness. The springs are linear, constant stiffness springs elements that only have vertical and horizontal stiffness. The actual bearing stiffness is quite complex and varies with the load as described by Bellakhdhar[7], however, the simple spring model was used to reduce the model complexity and simulation time. The bearing stiffness was assumed to be 2E6 lb/in. The actual bearing stiffness should not have a significant influence on the simulation results, as the bearing stiffness is simply a device to transfer loads from the crankshaft to the engine block.

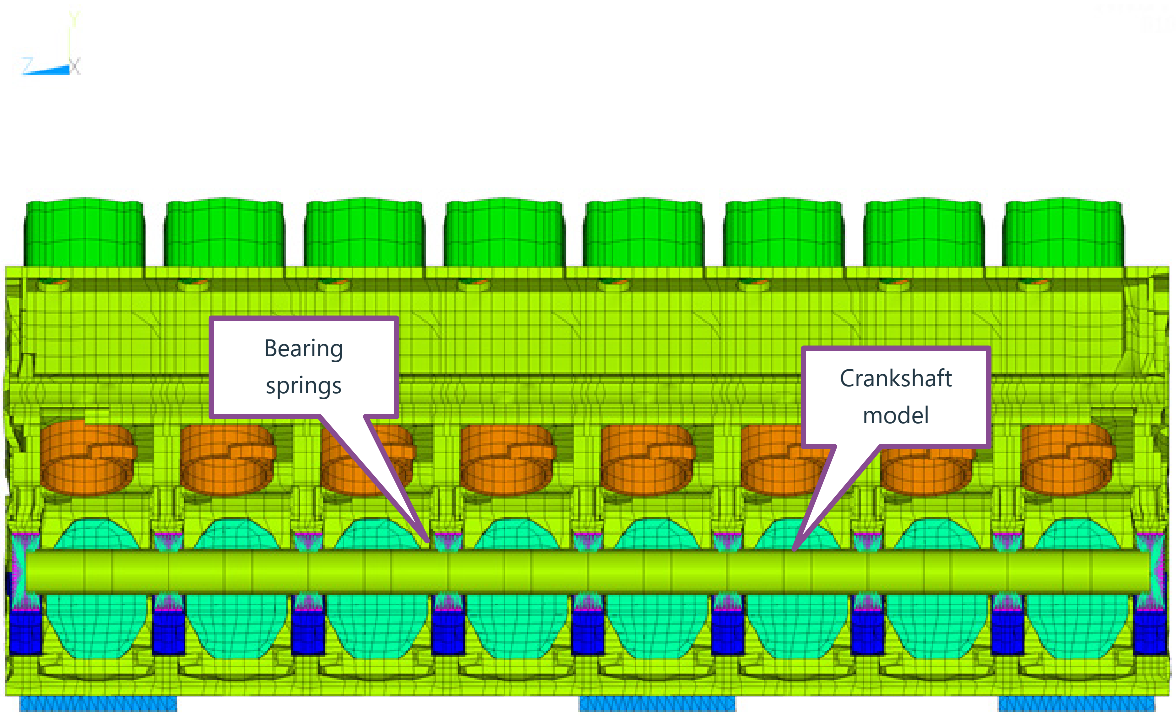


Figure 28: Engine section showing crankshaft

The harmonic analysis, or forced response analysis, was conducted at 1.5x with the engine forces from all 16 cylinders applied to the model. Images of the calculated engine vibration are shown in Figure 29. The overall shape of the calculated vibration agrees with the field measurements. The simulation results showed a vibration of 0.5 ips pk (3.0 mils pk) and 0.6 ips pk (3.6 mils pk) at the drive end and non-drive end of the engine. This vibration is based on an assumed damping ratio of 2%. Modal measurements during the field testing on the engine indicated the actual damping ratio is 1.8%. Scaling the vibration results according to the ratio of the assumed and measured damping ratio gives calculated vibration levels of 0.56 ips pk and 0.67 ips pk. The measured vibration, as shown in Figure 7, shows vibration on DE and ND with approximately 0.5 ips pk and 0.81 ips pk. The calculated engine vibrations agree relatively well with the measured vibration given the approximations and assumptions in the model. Also, the forces applied to the engine model are theoretical forces due to engine combustion. Measurements of the actual pressure-time signatures in the engine were not done. There may be uneven cylinder loading that would account for the variation between the measured and calculated engine vibration.

The forced response analysis and vibration results presented here are due to the combustion forces generated at 1.5x. Dynamic forces are also generated at 0.5x, 1.0x, 2.0x, 2.5x and higher harmonics from combustion forces. It is prudent to calculate the vibration of the engine due to these dynamic forces at other orders. Generally, vibration will be acceptable as long as the mechanical natural frequencies of the combined engine and pedestal system are not coincident, resulting in resonance. Avoiding resonance may not always be possible or practical, so conducting a forced response as described in this section is necessary.

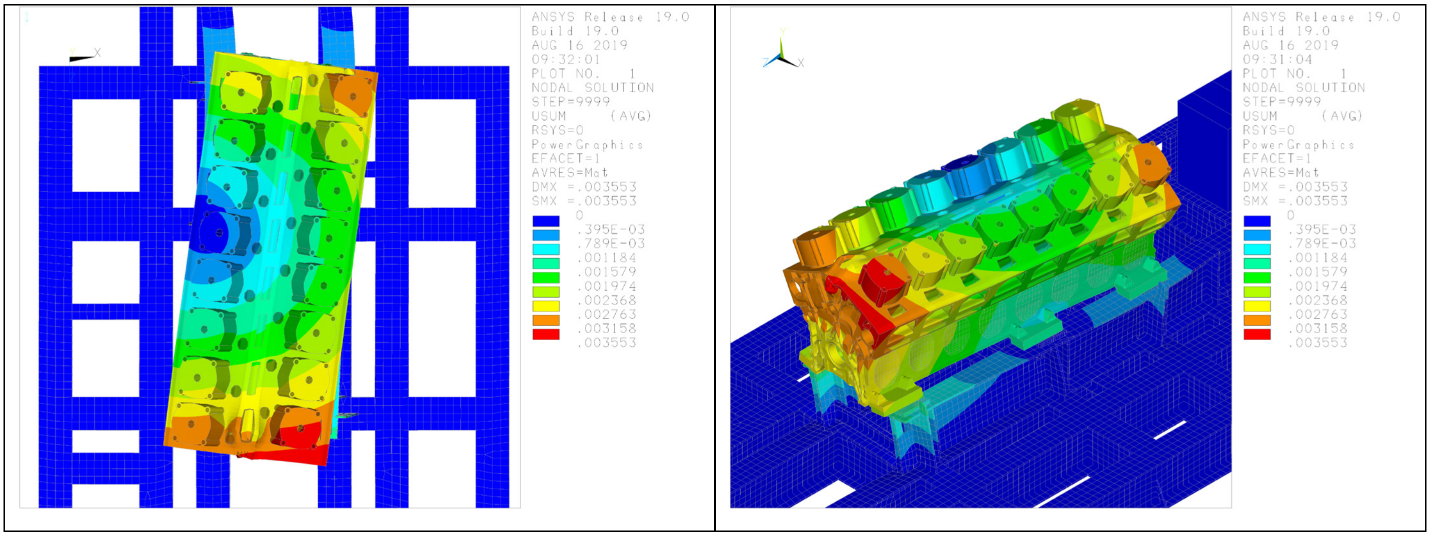


Figure 29: Vibration due to 1.5x engine forces

The previous discussion has focused on the vibration response due to the combustion forces. There are also dynamic forces generated by the engine reciprocating and rotating components. Differences between the weight of the pistons, connecting rods, and small and big end pins for the engine throws will generate forces and couples. The differences in the component weight are due to normal fabrication tolerances. The engine OEMs provide the primary (1x) and secondary (2x) forces and couples in the vertical and horizontal directions. An example of the unbalanced forces and couples for the Caterpillar G3616 engine is shown in Figure 30, as found in the Application and Installation Guide [8]. The stated unbalanced forces and couples are a maximum force or couple based on the worst-case combination of differences in the weights of the engine components. The actual engine forces and moments are not known. Unlike reciprocating compressors where the reciprocating and rotating weights are carefully measured, documented and balanced, the weight components on an engine are typically not documented. The forces and couples provided by the engine OEMs are helpful for design studies on new engine packages. However, the uncertainty in the actual magnitude of the forces and couples is significant when comparing measured engine vibration to simulation results.

Maximum Unbalanced Forces and Moments for G3616 Engine Due to Manufacturing Tolerances **Note:** 30″ Wide Damper (7C-2123) is used in the estimation.				
	Primary Force (kg)		Secondary Force (kg)	
Engine rpm	Vertical	Horizontal	Vertical	Horizontal
700	256	191	29	13
800	335	249	38	17
900	424	315	48	22
1000	523	389	60	27

Figure 30: Caterpillar force and couple information

The unbalanced forces and couples from reciprocating and rotating weights are generated at integer multiples of operating speed. The vibration investigated in this study occurred at 1.5x, therefore, consideration of these unbalanced forces and couples was not necessary for this project. However, for other projects or new package design studies the total engine vibration at 1.0x and 2.0x must be evaluated. The 1x and 2x vibration is the sum of the vibration from combustion and unbalance forces. The suggested approach to deal with the uncertainty in the amplitude and phase relationship of the unbalanced forces is to conduct separate simulations for each load case. The engine vibration from unbalanced forces and couples can be calculated from the loads, as shown in Figure 30. The engine vibration from combustion forces can be calculated from 0.5x to 2.5x. The maximum expected vibration would then be a scalar addition of the vibration results at each frequency from the two simulations. This approach will be a reliable method to calculate the worst-case engine vibration to ensure the package design is acceptable.

Conclusions and recommendations

The case study demonstrates the importance of evaluating engine and package vibration due to dynamic forces from the combustion process. The engine dynamic forces at half orders such as 0.5x, 1.5x and 2.5x must be evaluated. This requirement is missing in industry practices and guidelines such as *API 618* or the *GMRC Guideline For High-Speed Reciprocating Compressor Packages For Natural Gas Transmission and Storage Applications*.

The second major outcome from this case study is the demonstrated importance of including a representative engine finite element model. The engine block and mounts are not rigid. The flexibility of these components must be considered in the dynamic analysis of the package. Modern computer-aided design software, finite element programs and computer hardware enable practical sharing of model files and analysis of complex machinery models for everyday engineering studies. This analysis shows that the engine vibration due to the internal gas forces can be calculated accurately, which will enable proper decisions about the selection of the chock design and pedestal and skid design details.

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