API 618 Forced Response Studies

by
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Abstract

API 618, 4th Edition, Design Approach 3, requires that the vibration and stress levels in compressor manifolds and piping be calculated. Studies M.6 and M.7 are described. However, many would insist that this type of analysis is impractical or ineffective.

In fact, there are uncertainties associated with the behaviour of the final system versus the modelled system. Most of these uncertainties are related to construction and installation variations.

This paper attempts to demonstrate that mechanical modelling is valuable in the design of compressor installations. Good design, however, must be combined with attention to detail in the implementation stage. At best, all vibration problems can be avoided at start-up. At worst, tuning the system, after start-up, to reduce vibrations can be achieved with minimal impact. This efficient tuning of the system is achieved through an understanding of the sensitivity of the system to added mass, and by the strategic provision of places for stiffening braces and supports.

Introduction

API 618 contains specifications for modelling bottles and piping associated with reciprocating compressors. There are many reasons to do such modelling, but there are detractors who believe that the process has no value. We have found, however, that users of compressors who have experienced vibration problems after startup generally agree that forced response studies in the design stage are a good idea.

It is said that all models are wrong. This is true, but at issue is to what degree is a model wrong. Is there sufficient accuracy to provide useful information? There are limitations in the design stage regarding one's knowledge of the exact configuration of a system. After the design is completed, construction does not always conform to the design for various reasons. As a result, there will be differences between predicted and actual behavior of a system.

We see benefits from the modelling process, nonetheless. Examples are given that demonstrate what can be achieved with modelling.

API 618 Specifications

An API 618 4th Edition, Design Approach 3 analysis specifies that the requirements outlined in Studies M.2 through M.8 be met. The most significant change in the 4th Edition from previous versions of API 618, and the level of analysis commonly performed in industry, is the requirement to determine vibration and stress levels in the compressor manifolds and piping (Studies M.6 and M.7).

From API 618 4th Edition, Appendix M:

> **Study M.6 - Compressor Manifold System Vibration and Dynamic Stress Analysis:** Predict vibration and stress levels on pulsation bottles and closely coupled piping due to pulsation-induced unbalanced forces and cylinder gas loads.
>
> **Study M.7 - Piping Dynamic Stress Analysis:** Predict vibration and stress levels in critical areas on piping and vessels away from the compressor due to significant pulsation-induced unbalanced forces. Areas will be defined as critical by the mechanical design and predicted unbalanced force levels.

Predicting vibration and stress levels in the compressor manifold area (Study M.6) is often more complex than predicting vibration and stress levels away from the compressor (Study M.7). Pulsation-induced unbalanced forces can be obtained from an acoustical model. In the immediate compressor area it is important to not only include the pulsation-induced forces across the pulsation bottle or manifold and the close coupled piping, but the acoustical force acting between the pulsation bottle and cylinder gas passage on the suction and discharge side of the cylinder.

Cylinder gas loads that are to be included in a M.6 Study are often more influential than the pulsation-induced forces, especially for a reasonably designed acoustical system. Cylinder gas loads occur at all orders of run speed with the force at the fundamental order being the largest. Generally the lower orders are more significant than the higher orders. The relative magnitude of the harmonic components is a function of not only pressures but of cylinder loading. Whether or not a clearance volume pocket is fully open or closed, or if a cylinder is single acting or double acting can have a significant effect on the amplitude of the gas force at the different harmonics. For this reason it is critical to consider the full range of operating conditions in a M.6 Study and not just the design point of the compressor.

Note that the M.6 Study does not include the compressor residual unbalanced forces and moments. The scope of the analysis can be increased to include these excitation sources, but the model must be extended to include the frame, skid structure, and possibly the foundation.

Goals of Modelling

There are several goals of modelling the dynamic mechanical response of compressor systems. The obvious goals are

- Avoidance of mechanical resonance with known excitations [such as cylinder gas loads, pressure pulsation-induced shaking forces, and unbalanced forces and moments due to reciprocating inertias], though this is often not possible, such as with variable speed systems;
- Avoidance of excessive vibrations [see discussion of vibration problems below]; and
- Avoidance of stresses that could cause fatigue failures.

Less obvious goals are to provide

- suggestions for modifications to permit simple changes after start-up to correct vibration problems,
- a relationship between vibration and stress,
- an understanding of stress gradients in the vicinity of where a strain gauge might be installed,
- feedback to the designer of the pulsation control system when it could be better to reduce pulsations than to change the mechanical system response,
- understanding of the variations in vibrations with respect to operating conditions and load steps on the compressor. This information is particularly useful during start-up verification testing. It is necessary to compare the vibrations at the start-up condition with the vibrations that will occur when the system reaches the “worst vibration” condition and load step.

It would be interesting to predict the effect of pipe strain on vibrations and stresses in a system. However, this is a topic reserved for the installation and start-up verification testing. Suffice to say that pipe strain can have a negative effect on vibration characteristics of a piping system.

Results of Modelling

The results of model studies must be evaluated. A forced response study will provide predictions of mechanical natural frequencies, vibrations, and stresses. However, the designer needs to be prepared to decide if a vibration problem or a stress problem will exist. This process is not as straightforward as it might seem.

Diagnosing a Vibration Problem

In the design stage:

- Use of vibration guidelines can provide an answer. However, vibration guidelines are empirical and are intended to be conservative. The chart shown in Figure 1 has two sets of guidelines that are more or less empirically derived. Inexperienced users need to use guidelines such as these with caution. Generally, stresses will be acceptable even if vibrations exceed the vibration limits.
- Sometimes vibrations are perceived as a problem after a compressor is put into operation even though stresses are not dangerous [e.g., damaged gauges, loosening bolts, uncomfortable to stand on the grating, visually unsettling movement of piping or vessels, etc.] In the design stage, previous experience from measuring vibrations on operating compressors should be used to help assess the need to reduce vibrations even though the associated stresses are predicted to be low enough.
- Appendages are excited by the vibration of the pipe. Absolute vibrations on the end of an appendage may be worse than differential movements relative to the pipe. Elimination of appendages or proper design of appendages will help to avoid problems predicted by the model.
- Relationships between vibrations and stress are geometry dependent. Consequently, no one guideline can be valid for an entire system.
- Stress is proportional to displacement. Overall true peak displacements must be used to compare to stresses, unless a single frequency is present. Use of RMS Overall vibration levels leads to under-predicting actual displacements, particularly around reciprocating machinery, where many harmonics are contained in vibration signatures.

In the operating system:

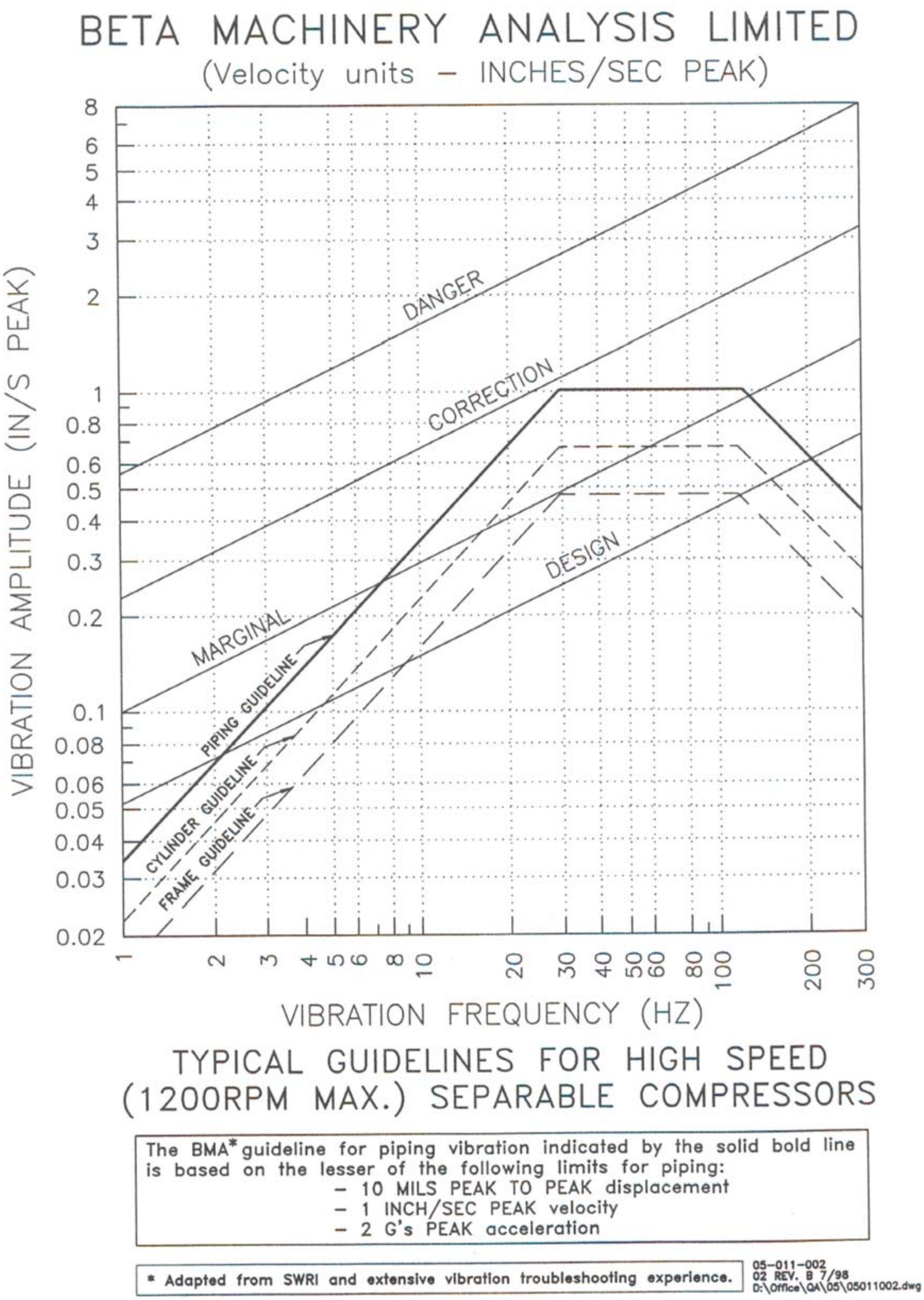
- Vibration guidelines are conservative so that they provide a screening tool. If vibrations are above guideline, consider the stress levels, or reduce vibrations.
- Meeting a vibration guideline does not guarantee the absence of a problem but risk is normally reduced to acceptable levels.
- It is useful to have predictions of vibration operating deflected shapes (ODS) and resultant stress to augment empirical guidelines. Compare measured ODS with the predicted ODS to ensure the validity of the predictions. This is more important at higher frequencies.
- Closely spaced natural frequencies can have distinctly different mode shapes leading to radically different stress levels for the same absolute vibration amplitude.

Diagnosing a Stress Problem

- Guidelines for allowable stress are less empirical than guidelines for vibrations. Assumptions remain related to residual static stresses and stress concentration factors.
- Since residual weld stresses are indeterminate, stress relieving welds reduces the uncertainties associated with predicted stresses by making the endurance limit predictable.
- Stress measurement or prediction includes the effects of macro geometry (unlike vibrations compared to an empirical guideline), but is still affected by micro geometry (such as weld surface treatments).
- Assumptions regarding measurements are still implicit (Is the strain gauge at the point of highest strain? If not, what is the highest strain?)
- One can use empirical guidelines for allowable strains derived from experience. See Reference 1, for example.

Use of Mechanical Natural Frequency (MNF) Predictions

- MNFs are faster and simpler to calculate than forced response studies [see Case History 1]
- Avoiding MNFs at 1X, and 2X shaft speed close to a compressor is generally viewed as a good idea, regardless whether a forced response study is done. (Most compressors have unbalanced forces and moments inherently due to reciprocating and rotating inertia.)
- Limit the analysis to MNF predictions until the system looks acceptable based on good engineering judgement and experience. Then do a forced response study.


BETA MACHINERY ANALYSIS LIMITED
(Velocity units – INCHES/SEC PEAK)
DANGER
CORRECTION
MARGINAL
DESIGN
PIPING GUIDELINE
CYLINDER GUIDELINE
FRAME GUIDELINE
VIBRATION AMPLITUDE (IN/S PEAK)
8
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VIBRATION FREQUENCY (HZ)
TYPICAL GUIDELINES FOR HIGH SPEED (1200RPM MAX.) SEPARABLE COMPRESSORS
The BMA* guideline for piping vibration indicated by the solid bold line is based on the lesser of the following limits for piping:
– 10 MILS PEAK TO PEAK displacement
– 1 INCH/SEC PEAK velocity
– 2 G's PEAK acceleration
* Adapted from SWRI and extensive vibration troubleshooting experience.
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Figure 1

- Determine the sensitivity of MNFs to mass and stiffness variations. This sensitivity information is of use in the field after start-up to de-tune resonances that have crept into the system for whatever reason [see Case History 2 for examples of variations in construction details].
- Provide an understanding of the location of MNFs relative to orders of run speed (using an interference diagram). If an MNF is predicted to be close to an order that has a propensity to cause high vibrations (either due to magnitude of the forces or the phase relationships), then checking ways to avoid such a resonance can be done at this stage. Consider adding welding pads to vessels or providing space for outboard cylinder supports in case additional supports are required after start-up. Changes to the bottles may be an alternative depending on the critical path of the project. Decisions to make such major changes should be made after evaluation of the forced response study's predicted stress or vibration.

<u>Usefulness of Forced Response Studies</u>

- Sometimes the forces and moments at 1X or 2X crankshaft speed are high enough to create high vibrations even though the system is non-resonant [see Case History 3]. A forced response model is the only way to predict such an eventuality.
- Sometimes cylinder gas forces at 3X or 4X are high enough to cause excessive resonant vibrations of the bottles. We have developed empirical rules to estimate the likelihood of such an occurrence, which helps when deciding to do a forced response study. However, only a forced response model will confirm a problem in the design stage. Changes to bottles may be required to solve such a problem, but bottle changes are not usually agreed to without compelling evidence.
- Operating conditions and load steps [hence forces] can vary. At start-up, it may be impractical to test the system over the entire range of conditions that are expected over the life of the unit. Therefore, it is useful to predict the vibrations and stresses for the system over the range of conditions that the unit will see. Due to the complex phase relationships among the many forces in a system, there is no other way to assess the various operating conditions to determine the worst case for stresses and vibrations.
- Tuning analyses can be done for fixed speed units. If vibrations or stresses are predicted to be high, changes in MNFs can be made to eliminate resonances. If a resonance does occur after start-up, the model will have provided an understanding of how to quickly and efficiently get rid of it.

<u>The Modelling Process</u>

- Fast track work makes communications difficult. Some projects have a very short time-line. Difficulties with communication in both directions can occur. Careful planning of the project is required to avoid these problems.
- Detailed models of the vessels are required [brick and plate elements as opposed to beam models]. Beam elements can be used to model piping.
- Predictions of vibration and stress at resonance depend on damping, which does vary between systems. Experience with actual systems is the basis for deciding what value of damping to use in a model. We know of no way to predict system damping.

<u>Case Histories</u>

The first case demonstrates the need for accurate boundary condition information.
The second case shows that small construction variations can make a large difference to vibrations.
The third case demonstrates that forced response models are sometimes required to predict a problem.

<u>Case History 1</u>

The first Case History discusses a situation wherein the assumptions regarding the boundary condition at the base of a pair of scrubbers were wrong in the design stage. It had been assumed that the steel beams under the scrubber would be located such that they would transfer maximum stiffness from the concrete foundation to the base of the scrubber. High vibrations resulted after start-up due to resonance of the mechanical natural frequencies of the scrubbers with moments in the compressor acting at twice crankshaft speed.

Pictures of the finite element analysis models of the actual base are shown in Figures 2 and 3. The skid beams below the scrubber are modeled in great detail. The scrubber is modelled with plate or brick elements, rather than with beam elements.

The results of the new as built model were found to closely simulate the field measured mechanical natural frequencies, see Table 1. Then, after adding more anchor bolts, putting gussets on the skirts and filling the void under each scrubber with grout, the predicted results were acceptable. Refer to the table below for the comparative numbers for one of the scrubbers.

Description	Horizontal MNF (Hz)	Axial MNF (Hz)
Shop test "as built"	10	12
Field test "as built"	11	12.8
Model: scrubber without skid (rigid base)	15.4	15.4
Model: scrubber with skid	10.4	11.3
Model with modifications	15.2	15.3
Field test with modifications	15.3	17.8

Table 1: Predicted and Measured Scrubber MNFs

Discussion:

Highlights from this case study are:

- Actual boundary conditions are required for accurate mechanical models.
- If the actual boundary conditions are modelled correctly, accurate results can be predicted by the model.
- The model results gave everyone involved the confidence to install the changes.
- The first set of changes was sufficient to resolve the problem. Trial repairs were not required.

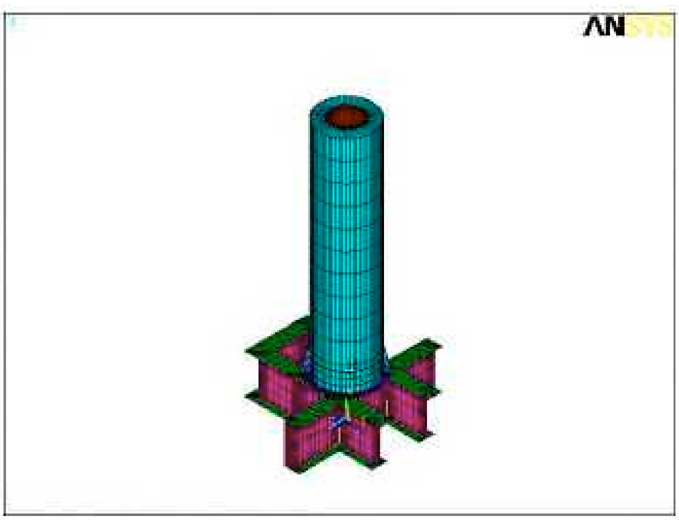


Figure 2: Scrubber and skid finite element model

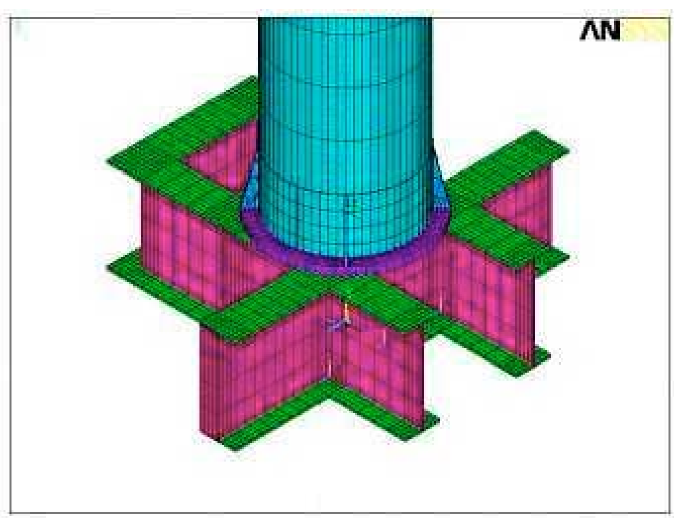


Figure 3: Detail View

Case History 2

This Case History is included to point out the significance of small variations in construction details.

As discussed beside the photographs, there were variations in the construction of the supports (upright location relative to the flange set, and cross-brace details). Superficially, these three support structures are the "same". In practice they behave quite differently.

Tuning of the MNFs was required to obtain acceptably low vibrations. This was done with the aid of tuning masses designed with the assistance of a computer model. A limited amount of experimenting was needed to get the required MNFs.

Photographs of support structures for 3 separate "identical" compressors:

Note the differences in the locations of the cross-braces between the top unit and the other two. Extra flexibility was created by putting the brace in the middle of the upright as well as below the top of the upright.

Another difference among the three units is the distance from the flange set to the clamp. The difference was not large, but was significant.

The top unit required several tuning weights to lower the MNF of the section of the piping to the left of the clamp. It had to be moved from near the top of 4 times run speed to near the bottom of 4^{th} order.

The middle unit required a tuning mass to lower its MNF to just above the top of 4^{th} order.

The bottom unit required no tuning mass to keep its MNF above 4^{th} order.

Case History 3

This Case History is included to demonstrate the need for forced response calculations in some cases. The original model of the frame and pedestal was done in the design stage. The mechanical natural frequency of that system was calculated and found to be above twice crankshaft speed. Avoidance of resonance with the couple at twice crankshaft speed was considered to be sufficient to avoid vibration problems.

In operation, it was found that vibrations were too high at the first order of crankshaft speed. The unbalanced couple about a vertical axis was very large. Vibrations increased with the square of the speed and peaked at the top of run speed.

Stresses were not considered a problem, although some discussions regarding the possibility of failure of the grout did occur. Our opinion was that as long as oil was not allowed to get between the grout and the steel, no problems would have occurred with the grout. A protective layer over the grout was recommended to prevent oil contacting the grout.

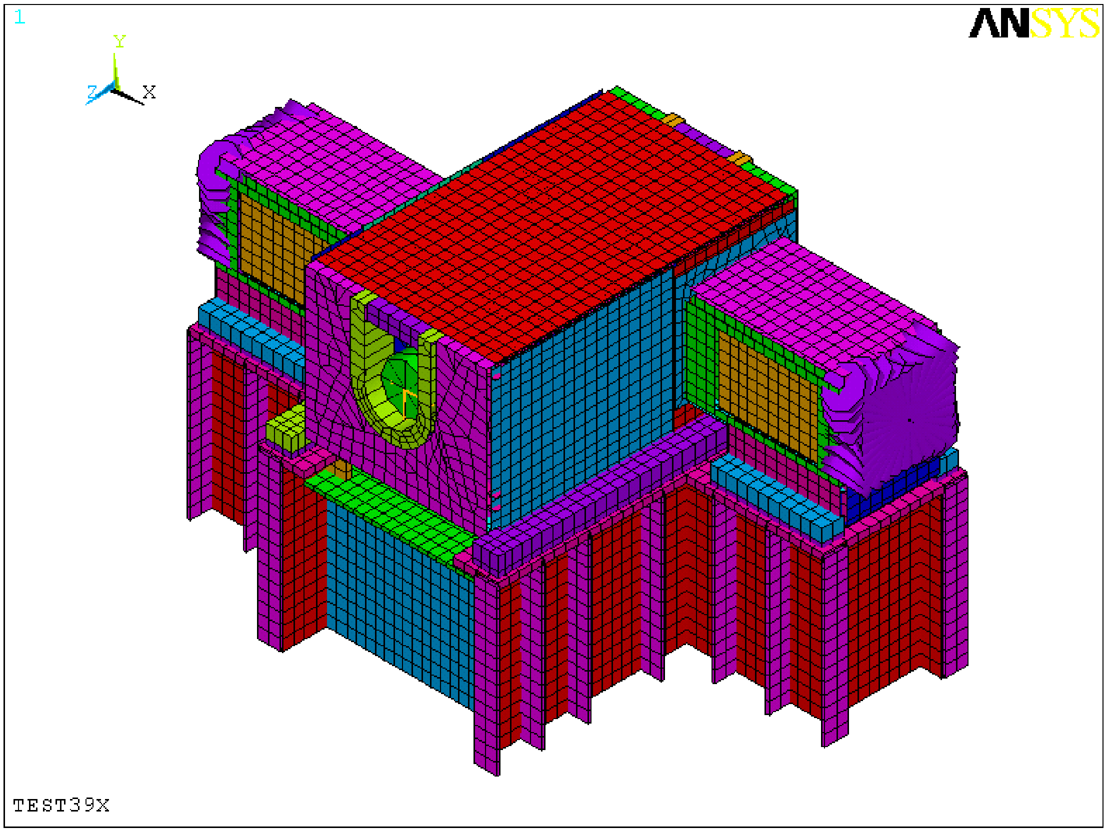
The vibration was, in effect, a quasi-static response. The frequency of the couple at run speed was low compared to the MNF of the frame and pedestal. Inadequate static stiffness of the system was at fault, combined with the large couple in the compressor.

A new model was created of the pedestal and the frame. Forces from cylinder gas loads as well as the couple caused by the reciprocating and rotating weights were used to excite the system. Predicted vibrations were quite close to the field measurements. The need to include gas forces was surprising. If the frame were rigid, the gas loads would cancel. Flexibility in the frame results in additional movements of the frame and pedestal due to the gas loads.

Several changes were modelled. The first set of modifications made (steel plates on the ends of the pedestals and rotating counterweights on the crankshaft to reduce the vertical couple) was successful in reducing the vibrations to the guideline level supplied by the manufacturer.

Several additional modifications (knee braces) had been suggested, but these were not required, as predicted by the model. A large incremental effort would have been required to introduce the additional changes. The model results gave us the confidence to do the limited set of modifications. Without the model, it is likely that the additional modifications would have been installed "just in case".

The model of the compressor frame and the pedestal are shown below. The model included details of the compressor frame, compressor dog-house, compressor crankshaft, the grout layer between the frame and the pedestal, and the frame anchor bolts.



Conclusions

1. API 618 mandated forced response studies can provide valuable insights into the performance of a reciprocating compressor system.
2. Experience has shown that not all risk of a dynamic force excitation related problems will be eliminated by doing a forced response study. The risk of failure due to dynamic force excitation will be reduced to acceptable levels if a forced response study if conducted.
3. One key to conducting a meaningful forced response study is including sufficient detail in the models.

Reference

Acoustic Fatigue Involving Large Turbocompressors and Pressure Reduction Systems, by David E. Jungbauer and Larry E. Blodgett, Southwest Research Institute, San Antonio, Texas.

Author Biographies

Brian C. Howes, M.Sc., P.Eng.
Chief Engineer
Brian graduated from the University of Calgary with a Master of Science in Solid Mechanics. His thesis was entitled *Acoustical Pulsations in Reciprocating Compressor Systems*.

Brian has worked with Beta Machinery Analysis since 1972. In his present position as Chief Engineer for Beta, he has traveled all over the world, troubleshooting as far abroad as India, China and Venezuela.

Brian has many technical papers to his credit. The range of machinery problems they cover includes all manner of reciprocating and rotating machinery and piping systems, balancing and alignment of machines, finite element analysis, modelling of pressure pulsation torsional vibration testing and modelling, flow induced pulsation troubleshooting and design, pulp and paper equipment such as pulp refiners, etc. He has worked on hundreds of reciprocating compressor installations.

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Kelly Eberle is a Senior Project Engineer for Beta Machinery Analysis Ltd., Calgary. His experience includes 14 years of troubleshooting problems and design for a wide range of equipment, with a primary focus on reciprocating compressor installations. He has a Bachelor of Science in Mechanical Engineering from the University of Saskatchewan.

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Project Engineer
Derrick graduated from the University of Saskatchewan with a Master of Science in Mechanical Engineering in 1993. His thesis was entitled *The Effect of Wind on the Air Intake of Cooling Towers*.

Previous employment with Atomic Energy of Canada Limited and contract work with the University of Saskatchewan has developed his experience in experimental measurement, dimensional analysis, fluid dynamics, flow-induced vibration, acoustics, and finite element modelling.

Derrick has worked with Beta Machinery Analysis since 1997. In his present position as Project Engineer for Beta, he works on digital acoustic simulation, thermal analysis, and dynamic finite element analysis of reciprocating compressor packages. Developing analysis techniques and practical application of technology are an integral part of his daily duties.