Case Studies Illustrating Improvements In Reciprocating Compressor Design

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Abstract

This paper uses two real cases to illustrate improvements in compressor system design.

One case involved a compressor with suction and discharge bottles spanning three cylinders. Several nozzle failures were experienced. Measured vibration was dominated by a component at about 70 Hz in the machine vertical and axial directions. Pulsation levels measured in the suction bottle and in the nozzles seemed reasonable in amplitude at the problem frequency.

Detailed investigation revealed a vibration mode involving rotation of the cylinder and bottle assemblies. This mode was excited by out-of-phase forces in the suction and discharge bottles and by vertical forces acting on the cylinders. The solution involved installation of a baffle in the discharge bottle to reduce the pulsation and, therefore the force (really the moment) exciting the mode.

The point here is that, even though the pulsation levels would be regarded as reasonable, the resulting vibration was not acceptable. This illustrates the need to understand and control forces, not just pulsation levels (Note: There are no API guidelines for pulsation levels inside pulsation bottles.)

The second case involved an existing compressor with high vibrations measured on suction and discharge vessels and piping. The dominant components were at 2X, 4X, and 8X compressor speed. Measured pressure pulsations were 200-300% of API limits at these harmonics.

Subsequently, computer modeling was used to develop a cost effective correction. A first step in situations like this is to ensure that the computer model is in agreement with the “as-found” field data. In this case, a state-of-the-art frequency domain model did not accurately predict the problematic 8X pulsations, though the calculated 2X and 4X components were a very good match. Of particular concern is that the predicted pulsation amplitudes at 8X were about 50% too low.

However, results from a time domain model were much closer to the 8X measured pulsation. The paper briefly outlines the differences in the modeling techniques and explores the reasons for the differing predictions in this example.

It is concluded that time-domain modeling is a crucial tool in accurately predicting pressure pulsation in reciprocating compressor installations.

Introduction

Trouble-shooting vibration problems in reciprocating compressor installations has lead to many advances in design-stage modeling techniques. This paper illustrates how, in two cases, severe vibration problems were solved using advanced analysis techniques. Industry standard design practices and American Petroleum Institute 618 guidelines were insufficient in solving these problems.

Case #1 – Residue Gas Service on Six Throw Separable Compressor

Machine Description

Figure 1: Residue Gas System

Six throw separable compressor (4 identical)
2650 hp engine driver
800 to 900 rpm speed range
Five compressor cylinders, two services
Two cylinders for two-stage propane service
Three cylinders for one stage residue service
Level of Acoustical Design: Another vendor acoustical model to API 618 Third Edition, Design Approach 3.

Vibration Problems and Nozzle Failures

The first field analysis of this machine was requested by the owner and the packager because of vibration problems on primarily the propane service side of this compressor. After a thorough vibration analysis, the field representative not only diagnosed and recommended solutions to the problems on the propane service, but also revealed a vibration concern on the residue side of the compressor at about 71 Hz (5 times compressor speed at 842 rpm.) This vibration was attributed to high pulsation-induced unbalanced forces in the residue systems. An acoustical analysis of the residue systems was recommended.

The recommended solutions for the propane service were implemented and those problems were solved. The acoustical analysis of the residue systems was not completed by the choice of the customer.

About three years later after several nozzle failures on the residue system pulsation bottles, Beta Machinery Analysis was asked to evaluate these machines again. The cylinder #5 suction and cylinder #1 discharge nozzles had failed several times. Again the 71 Hz problem was identified and an acoustical analysis was recommended. A detailed operating deflected shape (O.D.S.) was measured to help identify the “shape” of the vibration on the residue system. This type of analysis is useful in identifying the forces involved and identifying stressed components. Figure 2 is a graphical representation of the O.D.S. for the 71 Hz vibration. It was noted that all of the components were vibrating around cylinder # 3. System geometry and constraints along with high vibration produced large stresses in the cylinder nozzles.

There was a system natural frequency measured at around 70 Hz. High forces coincident with this system natural frequency led to unacceptable vibration and stress levels.

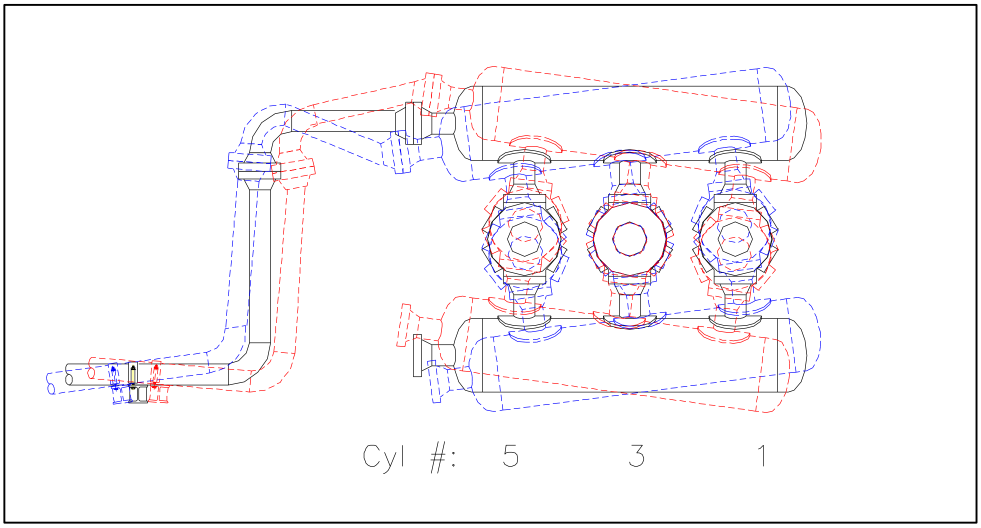


Figure 2: Operating Deflected Shape (O.D.S.) of residue system at 71Hz. The dashed images represent the envelope of the vibration. Note that cylinder #3 is relatively motionless.

Design Analysis and Solution

An acoustical simulation was done for both the suction and discharge of the residue system using acoustic modeling software. Predicted pulsation levels in the piping past the pulsation bottles were well within API 618 guidelines. However, the forces in the bottles and in the cylinder nozzles were above guidelines used by the authors. This, however did not explain why there was such high vibration in the vertical direction on cylinders #1 and 5 and not on cylinder #3.

Further analysis revealed that the phase relationship between the forces in the suction and discharge systems created a strong alternating couple about cylinder #3. This couple matched very well with the measured O.D.S. Figure 3 illustrates the 5X compressor speed resultant couple for the "as-found" and modified system. It was noted that modifying the discharge bottle would have the most impact on the resultant couple and therefore the vibration levels. Modification of the suction bottle was predicted to have little benefit.

A modification to the existing discharge bottle was recommended and implemented on one unit. Vibration measurements were taken for comparison. Figure 4 demonstrates the typical improvement in vibration levels on the modified unit. Approximately an 80% reduction in peak vibration levels was realized. Modification of the remaining units was to be completed at the client's earliest opportunity.

Conclusions

It is insufficient to control line-side (outside the bottles) and cylinder flange pulsation levels when completing an acoustical analysis.

Consideration of pulsation-induced unbalanced forces within the pulsation bottles and piping is critical to a sound pulsation control design.

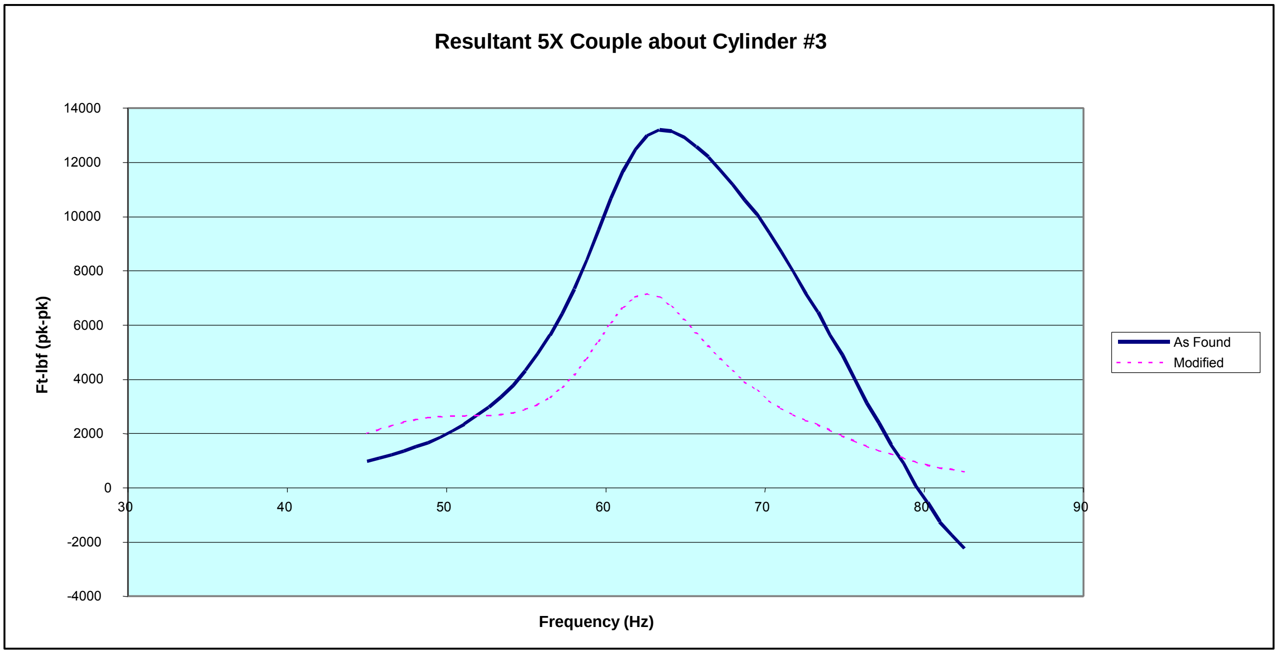


Figure 3: Resultant Couple about Cylinder #3. This plot shows that modifying the discharge bottle should reduce the couple at 70 Hz by 66%.

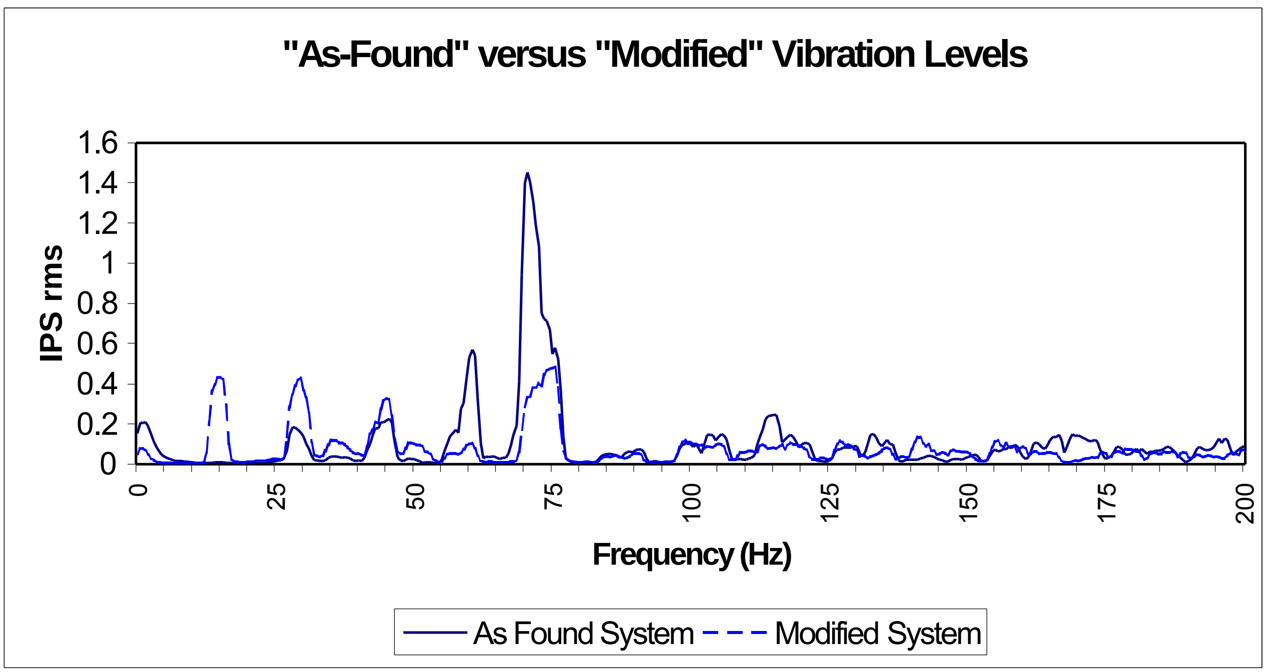
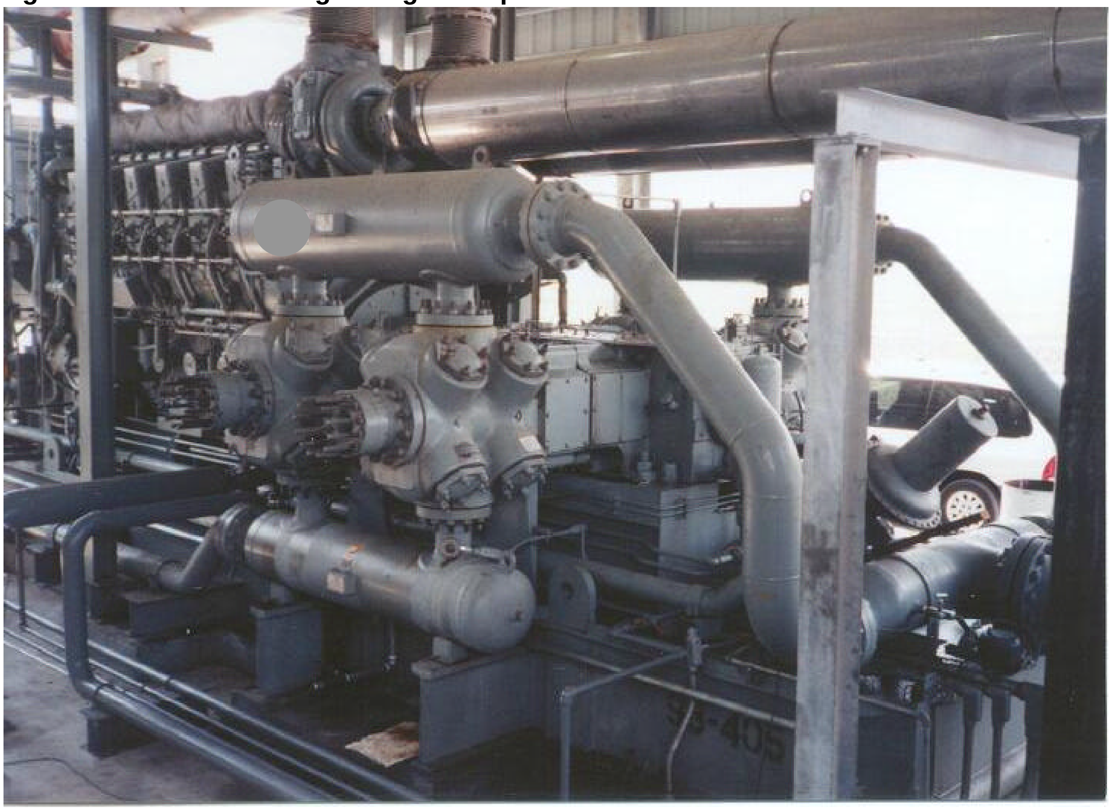


Figure 4: Typical vibration improvement with modified discharge bottle. Vibration level in inches-per-second rms versus frequency in Hz (speed sweep from 800 to 900 rpm.) The “as found” spectrum is the vibration in the vertical direction of cylinder #5 without the discharge bottle modification and the “modified” spectrum is the same measurement with the bottle modified. The reduction in peak vibration at 71Hz is approximately 80%.

Case #2 – Single Stage 4 Throw Separable Compressor

Machine Description

Figure 5: Four Throw Single Stage Compressor



Four throw separable compressors (3 identical at one location.)
3335 hp engine driver
800 to 1000 rpm speed range
Four compressor cylinders on one stage
Sweet Natural Gas, 0.57 specific gravity
Level of Acoustical Design: None

Vibration Problems

Beta Machinery Analysis was asked to help diagnose and solve vibration problems on these new compressor packages. Many component failures on the compressor, engine and piping had plagued these units since start-up. The compressor packager had been, since start-up, making changes to the system including the installation of orifice plates, support changes, and piping changes. These changes made some improvement, but failures continued and vibration levels were not yet acceptable.

High frequency vibration (above 50 Hz) was measured at many locations on the pulsation bottles and piping. Pulsation levels far in excess of API 618 design pulsation limit were measured in various locations. Pulsation-induced unbalanced forces were blamed for the majority of the problems and an acoustical simulation was recommended. Once these forces were under control, it was anticipated that some support modification might be necessary as well.

Design Analysis and Solution

An acoustical simulation was done for the suction and discharge systems using frequency domain acoustical modeling software. Predicted pulsation levels matched quite closely to field-measured data. Very high unbalanced forces were predicted in the bottles and piping. At the request of the customer, modifications were limited to be external to the pulsation bottles. Line sizes were changed and orifice plates were recommended. It was predicted that the system forces were reduced by at least 80%. Some of the predicted forces remained above guidelines used by the authors. As-per the first field visit recommendations, another field analysis was completed.

Though most problem vibrations had been improved dramatically, some areas continued to be a concern. Changes to supports and additional bracing were recommended. An additional field visit was requested subsequent to the mechanical improvements being completed.

The third field analysis showed that vibration levels at frequencies below 100 Hz were now acceptable. Vibration at about 140 Hz, though greatly improved, remained somewhat over industry guideline. More pulsation data were taken at points previously unavailable. These measurements were compared with the frequency domain acoustical model and it was found that peaks in pulsation in the 140 Hz range were being under-predicted by about 50%. That is, the residual unbalanced forces in the system were likely twice as high as originally thought. This seemed a likely explanation for the remaining 140 Hz problems. This discrepancy between the model and the field measurements prompted further investigation using time domain acoustical modeling. The time domain model agreed better with the measured data. See Figure 6 for plots of predicted versus measured pulsation. A solution to the problem, that involved the installation of pulsation bottle internals, was designed. Due to the expense and logistics of this type of solution, it has yet to be implemented and the maintenance of clamps and supports on these units continues to be a problem.

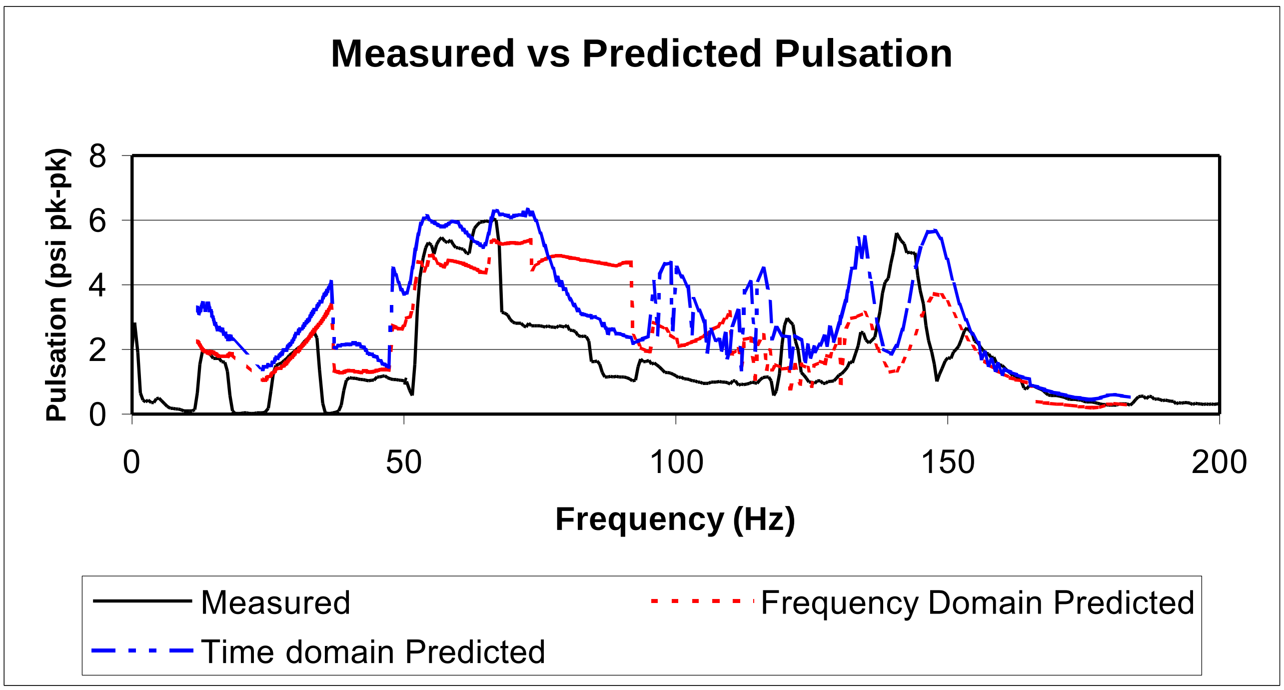


Figure 6: Predicted vs Measured Pulsation It is demonstrated that the time domain model predicts pulsations better than the frequency domain model, especially at higher frequencies.

Differences Between Frequency and Time-Domain Acoustical Modeling

A frequency-based acoustical analysis utilizes discrete harmonic input into a piping model. There is no interaction between the acoustical model and the compressor cylinder clearance volume. That is, the source for the model is located at the line-side of the compressor valves.

Time-domain modeling, on the other hand, provides a time-based input into a piping model. The source of pulsation is at the piston face and input only occurs when the valve is open. Interaction between the varying clearance volume and the piping is considered.

A time-domain model is generally considered to be more accurate than a frequency domain model. This has been shown in several case histories, including Case #2 in this paper.

Conclusions

It is critical to complete an acoustical simulation at the design stage on many types of reciprocating compressors.

All acoustical simulations should utilize a time domain interactive model to check acoustical results.

Overall Summary

These two case studies illustrate the value of design-stage acoustical modeling and underline the fact that this type of analysis is evolving due to lessons learned in real-life problem solving.

Case #1 illustrates the need for considering the forces generated by pulsation during an acoustical design. Forces in one system (suction or discharge) can cause vibration concerns in the other system.

Case #2 emphasizes the value in time-domain acoustical modeling. Though frequency-domain modeling is the mainstay of acoustical design, advancements in computer power and software have made it possible to use an interactive time domain model to confirm the quality of your acoustical system.

References

Both case studies were derived from Beta Machinery Analysis trouble-shooting reports:

- Reference numbers 10177, 10178, 205730, 205730a, and 205776 for Case #1
- Reference numbers 205771, 205763, 205763b, and 205763c for Case #2