

## GMC 2013: Piping Misalignment and Vibration Related Fatigue Failures

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### Introduction

Many vibration problems are amplified by piping misalignment (or pipe strain). In some cases the measured vibration amplitudes are double, triple, or even higher when under pipe strain, compared to after relieving pipe strain.

Pipe strain occurs due to many causes including flange misalignment, and gaps occurring between the process pipe and its supports.

For example, during a recent field visit to investigate vibration on the piping system near a PSV, vibration was measured before and after removing the pipe strain. As shown in Figure 1 below, the “as found” case experienced excessive vibration of 4.5 ips. This is significantly higher than the acceptable guideline of 1.0 ips. Once the pipe strain was removed, the vibration was reduced to acceptable levels (0.25 ips).

Pipe strain has been shown to contribute to significant vibration problems including vibration induced piping failures.

Using field data, this paper will provide examples of pipe strain problems.

Pipe strain vibration problems are related to the combination of static and dynamic stresses. The different stress related issues are presented as background material.

The authors will outline existing standards and best practices to mitigate these design and fabrication problems.

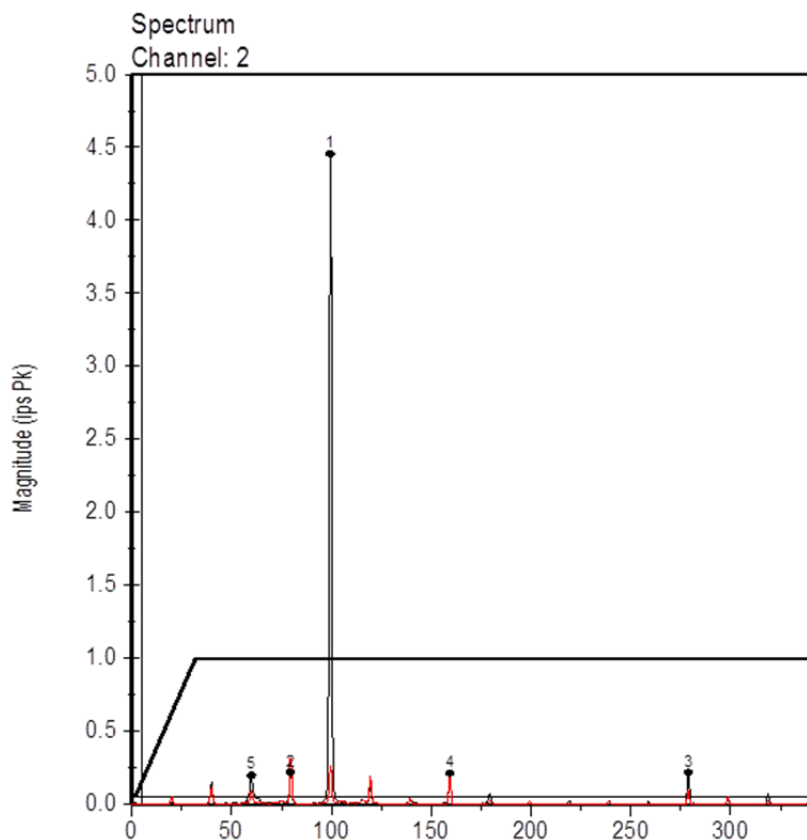


Figure 1 Example vibration spectrum for piping with and without strain

### 1. Overview of Pipe Strain

Pipe strain, for the purpose of this paper, is considered to be the load introduced into the piping due to static deflections. One of the major contributors of controllable pipe strain is piping misalignment. The root cause of misalignment can be attributed to deficiencies in design, fabrication, assembly, and installation practices. By itself, pipe strain may not be sufficient to cause failure, but when combined

with the dynamic stresses related to reciprocating compressors, pipe strain becomes an element critical to the reliability of piping systems. The three figures below show examples of geometries that resulted in pipe strain



Figure 2: Flange Alignment Problem (discovered after removing bolts)

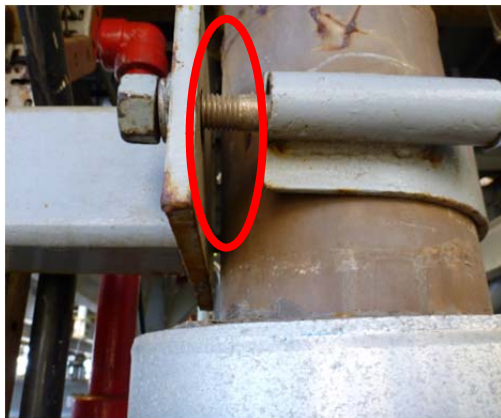


Figure 3: Gap in piping due to improper installation (pipe strain will occur when bolts are tightened)

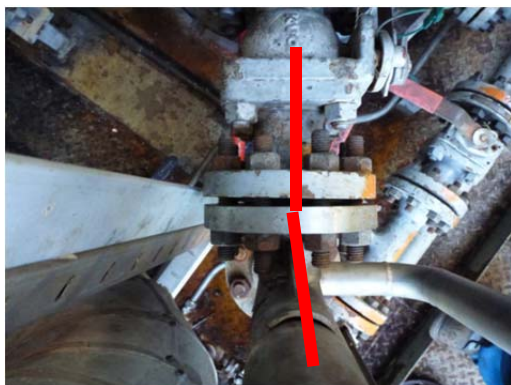


Figure 4 Angular Misalignment of Flange (creating pipe strain)

Two pipe strain examples are presented below to illustrate the magnitude of vibration problems and the types of solutions employed to resolve the issue. These examples include one case relating to static stress problems and the other relating to dynamic stress problems.

### **1.a Static stress related to piping misalignment**

In this case, bolt failure occurred between the pedestal support and the distance piece. The problem was due to increased static loading of the bolts of the support connecting the distance piece to the pedestal caused by:

- Short coupled scrubber and pulsation bottle
- Spool piece (Figure 5) for inlet screen was too short
- Loads transferred to the pedestal support bolts were concluded to have exceeded bolt preloads and subsequent vibratory loosening of the fastener occurred.
  - It is important to note that flange alignment actually met the requirements defined in ASME B31.3

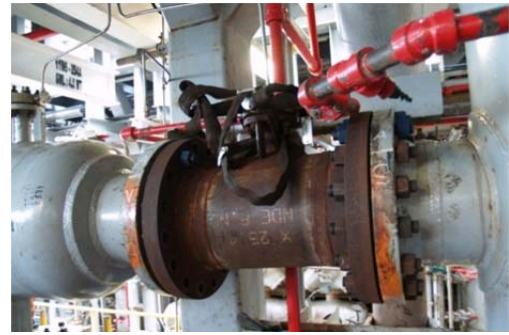


Figure 5 On site fabricated spool piece to address pipe strain concerns

To address the pipe strain, the following solution was successfully applied:

- Custom fabricated new spool piece and custom thickness plate for inlet screen installation
  - Without analysis, standard practice of increasing the pedestal bolt torques would have potentially moved the stresses being caused by misalignment and dynamic loads to compressor components.

### **1.b Dynamic stress on piping system**

Figure 6 shows a gap between the pipe and the pipe support. Once the bolts are fully tightened, the pipe will be strained.

The pipe vibration problem is due to the pipe not being shimmed at the clamp and the resulting vibration response being near the MNF.

A pipe strain condition could be created by tightening the clamp to draw the pipe to the clamp and result in increased static stress, shift of the mechanical natural frequency of the system, and a changing in damping.



Figure 6 Gap between pipe and support prior to shimming

To solve the vibration problem, related to what is shown in Figure 6, shims were added between pipe and clamp. This resulted in reducing vibration to acceptable limits.

## **2. Stresses That Affect Fatigue Failures of Piping**

Sources of pipe stress come from different aspects of a reciprocating compressor's installation and operation. The contributing components to the piping stress are static stresses and dynamic stresses. Analyzing dynamic stresses requires additional considerations, which will also be discussed. The cause of fatigue failures for process piping relates to the combination of the both the static and dynamic stress.

### **2a. Sources of static stresses**

Static Stress: For process piping, loading is dependent on more than simple uniaxial loading. Process piping loads are much more complex. Piping is under pressure and when installed out of alignment it is also under bending loading, which subjects the material to shear stresses. The combination of static loads becomes the mean stress acting on the pipe. In addition to the stress introduced by piping misalignment the following sources also exist:

- Weight
  - For unsupported lengths of pipe, the weight of the pipe itself can introduce static stresses
  - The weight of unsupported process valves introduce static stress

- Internal pressure
- Assembly stresses includes two loads
  - Bolt loading
  - Alignment and fit up loads
- Environmental loading could introduce loads into the piping directly or indirectly through reactions from structures and vessels affected by environmental loads
  - Thermal
  - Wind
  - Seismic
- Skid installation practices could introduce loads indirectly through reactions from structures and vessels if installation causes deflection of the skid that differ from on-skid assembly prior to site installation
  - Gravel
  - Piles
  - Grouted concrete
  - Other

### 2.b Sources of dynamic stresses

Piping in reciprocating compressor service is also subject to dynamic stresses from pulsation induced forces that create an additional component of alternating hoop stress as well as alternating axial loading. Besides pressure pulsations, additional dynamic forces related to the unbalanced forces and couples of the compressor, crosshead guide forces, and cylinder stretch are also present. All of these dynamic forces generate piping vibration. Static stresses combined with dynamic stresses can lead to failures. On their own, the individual stresses would not lead to failure, but when static and dynamic stresses are combined, the cumulative affect creates vibration problems.

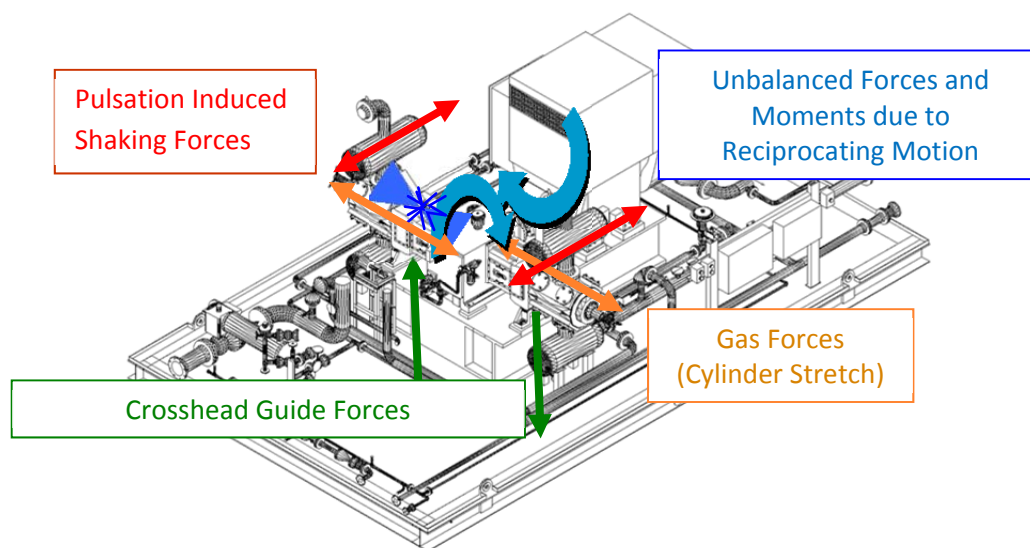


Figure 7 Dynamic forces associated with a reciprocating compressor package

Dynamic forces acting on the compressor and piping system:

- Compressor unbalanced forces and moments from reciprocating motion
- Compressor crosshead guide forces
- Compressor cylinder gas forces (cylinder stretch)
- Pulsation induced shaking forces in nearby vessels and piping (on- and off-skid)
- Lateral force related to torsional response

The dynamic forces affect:

- Main process
- Small bore attachments

### 2.c Important considerations for dynamic forces

Important to the analysis and prediction of the response of piping under dynamic loads are considerations for mechanical natural frequency (MNF) of the system, stiffness of the system, and the system damping.

MNF is defined as the frequency where an object will want to vibrate or oscillate in response to applied force. MNF is a function of spring constant (stiffness) and mass. Dynamic flexibility (the inverse of dynamic stiffness) is maximum at the MNF.

Stiffness is defined as a system's resistance to deflection or deformation from an applied force. It is related to a system's geometric properties and the material or system's inherent flexibility. Static stiffness directly affects the MNF of a system and can either raise or lower it.

Damping can be defined as a force that acts opposite oscillating or vibratory motion. Damping is dependent on material and stress during operation and is not affected by geometry. The material dependent factor is related to the flexibility of the material. Stressed components of mechanical assemblies are affected by slippage and are related to structural or joint factors. High damping will result in decreased response at the MNF, while low damping will result in increased response at the MNF.

Section 3 of this paper will present two examples of field data that suggest pipe strain has affected damping values.

### 2.d Effect of combined static and dynamic stress

The combined stress, which includes both the mean stress and alternating stress, can be represented on a Goodman diagram. The Goodman diagram (Figure 8) shows that the allowable amplitude of alternating stress decreases as mean stress increases. Vice versa, the allowed amplitude of alternating stress is greater as the mean stress is lower.

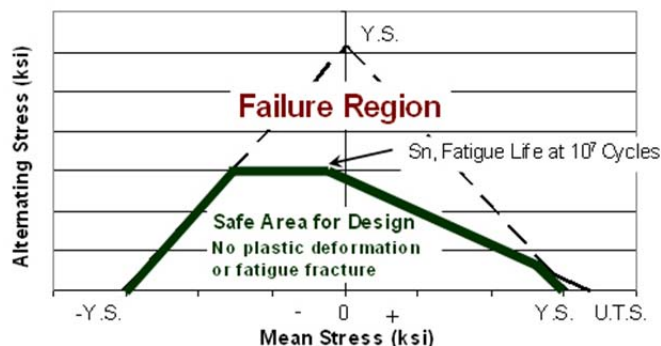


Figure 8 Goodman Diagram for Axial or Bending loads

## 3. Examples of Field Data with Damping Response

Damping is the least understood of the dynamic consideration factors but has the potential to have a large effect to the vibration response. The following two cases illustrate how damping is believed to be affected by pipe strain. The conclusion that a change in damping results from pipe strain is based on data collected where the input forces were not believed to have been reduced.

### 3.a Damping example (suction bottle and piping)

In this case, the problem is vibration on the suction bottle and piping (Figure 9). Initial vibration measurements of the piping on the middle throw between head end unloader and pulsation bottle had vibration over guideline (1.5 ips pk @ 119.4 Hz).

Loosening of the clamp that tied the pipe to the brace showed a gap between brace and piping.

The following solution was implemented to correct the problem:

- Gap between brace and pipe was shimmed and clamp retightened
- Vibrations after shimming were reduced to guideline (.45 ips pk @ 119.4 Hz)
- Figure 10 shows vibration trace in black is “as found” and red is after shimming the gap



Figure 9 Bracing of cylinder head end unloader piping

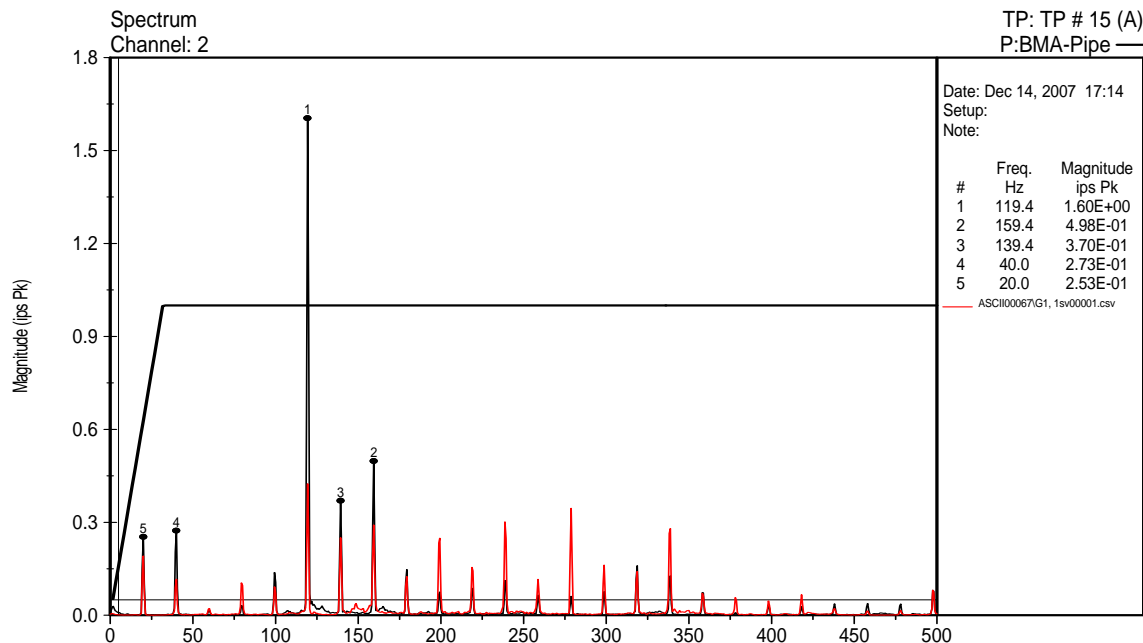


Figure 10 Spectrum plot of before and after measurements showing pipe strain of head end unloader piping

There was a significant reduction of vibration amplitude, but without a shift in MNF. It has been concluded that a change in stiffness of the assembly did not occur, but the reduction in response could be explained by a change in damping.

### 3.b Damping example (discharge piping)

This case illustrates 100 times increase in vibration due to pipe strain. This is another example where a change in damping seems to contribute to the problem.

Initial measurements of a vertical section of piping had vibrations over guidelines (measured 4.5 ips pk @ 99.4 Hz).

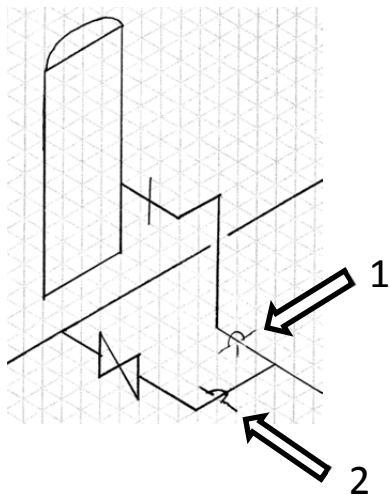


Figure 11 Piping ISO with support locations identified

Vibrations after shimming and reinstallation of the clamps were measured and found below guideline (.25 ips pk @ 99.4 Hz). Based on the reduction of vibration amplitude but without a shift in MNF it has been concluded that a change in stiffness of the assembly did not occur, but that the reduction in response could be explained by a **change in damping**.

Loosening of clamps at the two supports showed no gap to exist at the post support closest to the vertical pipe, identified as 1 in Figure 11, which had the highest measured vibration.

Support 1 was found to be too tall and that pipe was in a bind on the support. When support 2 was loosened, a gap was found between support and piping.

Figure 12 shows vibration trace in **black** is “as found” and **red** is after shimming the gap

The vibration problem was solved with proper shimming of the pipe at the support previously shown to have a gap (marked 2 in Figure 11). The strain at the support marked as 1 in Figure 11 was relieved and allowed for shimming of the pipe to the support and reinstallation of the pipe clamps.

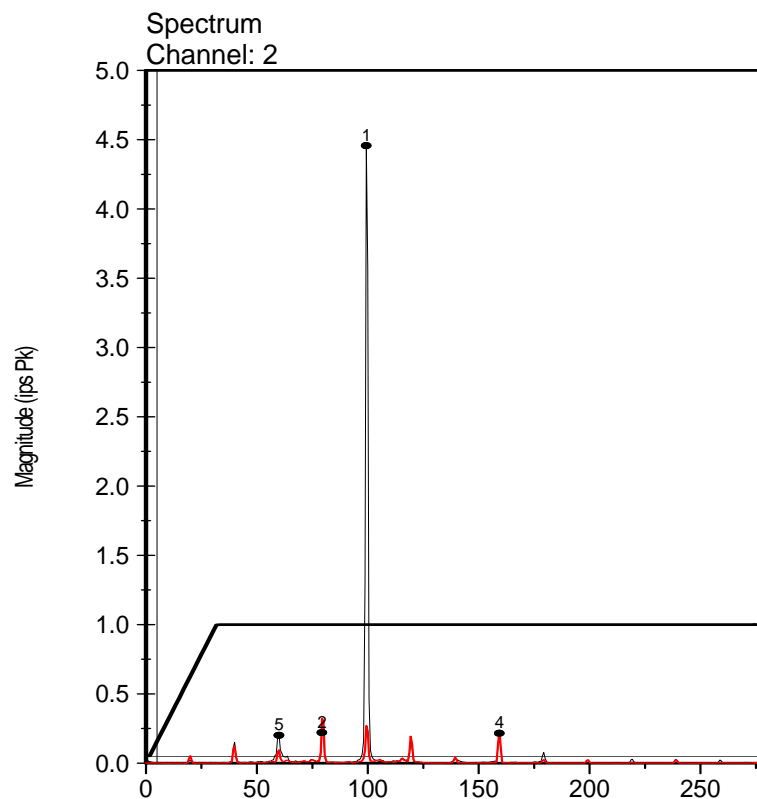


Figure 12 Spectrum plot of before and after measurements showing pipe strain resulting from support installation

#### 4. Existing Standards & Good Design Practices

To reduce the likelihood of piping strain, adherence to existing standards, guidelines, and good design practices is critical.

In the authors' experience, the most commonly referenced standard and recommended practices in the compression industry are ASME B31.3 Process Piping and API RP 686. A summary of the guidance in regards to pipe strain is as follows:

- a. ASME B31.3 Process Piping Section 335 describes the requirements for assembly and erecting.

- i. As it relates to alignment the specification prohibits introducing strain into the piping or equipment by forcing piping into alignment.
  - ii. Specific tolerances for flanged joint alignment are further defined as the following:
    1. Flange faces aligned to plane within 1/16" per foot as measured across any diameter.
    2. Flange bolt holes should be within .0125" maximum offset.
  - iii. The Standard also provides methods for doing a flexibility analysis for a piping system and calculating displacement stresses.
- b. API RP 686: Recommended Practice for Machinery Installation and Installation Design 2<sup>nd</sup> ed. Chapter 6 provides guidelines for piping alignment as well as guidelines associated with measurement of components that could identify pipe strain having effects to connected machinery.
- i. Like ASME B31.3 the Standard specifically prohibits forcing piping into alignment.
  - ii. In terms of piping and flange alignments the following are defined:
    1. Machinery and piping flange faces are considered and should be parallel to within .001 in/in of pipe flange diameter with a maximum allowable of .030" for flanges larger than 10" in diameter.
    2. For flanges smaller than 10" in diameter the requirement is to be parallel with 0.010" or less.
    3. Flange face separation is required to be within  $\pm 1/16$ " taking into consideration gasket spacing.
  - iii. An additional requirement for measurement of pipe strain is provided for verifying that shaft movement of coupled equipment does not exceed .002" in horizontal or vertical directions when in final installation state.
  - iv. Reciprocating compressors have an additional requirement for verifying that piston rod runout before and after installation are within the compressor manufacturer or API 618 allowances.

Concerns with existing standards are that they are not specifically written with consideration for aspects related to reciprocating compressor installations and, as a result, may not be adequate. Specific areas of concerns as they relate to reciprocating compressors are multi-cylinder nozzle pulsation bottles (more so on 3 nozzle bottles), and close coupled scrubber and suction pulsation bottle configurations.

Based on observed conditions and work practices during field analysis, the following best practices are recommended:

- Ensure that piping is not forced into alignment with mating flanges or support locations using force such as hydraulic jacks and chain falls during assembly.
- Design pipe supports with at least ¼" gap between piping and support to allow for shimming after installation is in hot (operating) condition.
- Pipe clamp bolts and pipe supports should line up without forcing piping to line up with the support. Slotting of support holes may be a good practice to allow for variability due to construction tolerances and on-site thermal growth during operation.
- Angular alignment of mating flanges should be as near to parallel as possible.
- Ensure multi-cylinder nozzle pulsation bottles have flange faces on the same plane and are toleranced to mate with cylinder nozzles without stressing the nozzles.
- Implement custom orifice plate thickness if axial misalignment between flanges is too great, rather than drawing the flanges together with bolting.
- Provide support for large process valves.
- Implement sufficient pipe supports at adequate spacing intervals to prevent induced mean stresses from the piping weight.
- Ensure skid is designed appropriately for the installation to reduce the likelihood of deflection of the skid in transit to final site versus the manufacturing location.
- Final fit up of piping near machinery by field welding of pipe spool pieces.



## Summary

- Pipe strain does not appear to change response frequencies.
- Pipe strain causes vibration to increase at some response frequencies as a result of these possible mechanisms:
  - Static stress
  - Dynamic stress
  - Damping changes
  - Some other phenomena
- Good design practices as well as good manufacturing and assembly practices reduce the likelihood of problems.
- Proper installation is critical to ensure low vibration and reliability.
- Identifying root cause is complex and cannot be easily identified without the right tools and experience.
- Further research is needed to accomplish the following:
  - Conduct finite element analysis and field measurements to compare the change in static stresses as it relates to varying degrees of misalignment.
  - Measure vibrations and relate the change in vibratory response to differences in initial static stress.
  - Conduct hysteresis measurements in laboratory and at field installations with and without static stress.
  - Develop improved alignment recommendations for installation of piping systems in strain critical applications.
  - Develop a field calculator able to define allowable alignment tolerances more specific to individual piping joints, with less dependence on generalized tolerances for a very wide range of installation variables.
  - Validate findings and recommendations through laboratory and field testing

## References

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## Authors

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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 performed troubleshooting services all over the world.

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, modeling of pressure pulsation, torsional vibration testing and modeling, flow induced pulsation troubleshooting and design, pulp and paper equipment such as pulp refiners, etc. He has also worked on hundreds of reciprocating compressor installations.

#### Gary Maxwell

Gary graduated from University of Houston with his Bachelor of Science degree in Mechanical Engineering Technology.

Before working for Beta Gary was the Engineering Manager for a compressor OEM. He was responsible for the design of reciprocating compressors and served as a member of the API 618 Task Force. While working as engineer and manager, Gary gained experience with field troubleshooting reciprocating compressor installations, performance, and operations. Gary's prior experience also includes working as Research and Development Test Technician and Engineer with responsibilities including designing test fixtures and test cells, as well as instrumenting and conducting test plans.

As General Manager for Beta's US Operations, Gary is responsible for Beta's US based customers involved in compressor, pump and piping design services, as well as field engineering projects. Gary also manages Beta's Houston-based engineering team, project execution, and other aspects of the business.