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In-field vibration assessment of the piping of a reciprocating compressor plant

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Executive Summary

For a number of years visible vibrations were noticeable in the process piping connected to a reciprocating compressor at a refinery, this was despite a pulsation analysis having been conducted at the design stage. The effects of these vibrations were also visible in the small-bore instrumentation pipes, even though they were braced back to the main run pipe. The operator of the plant was worried that fatigue cracks could occur, especially in the small bore lines, and therefore a study was conducted to determine how the vibration levels could be reduced and whether they were leading to stress levels exceeding the endurance limit.

To calculate the stress magnitudes arising in the piping, including those in the small bore connections, a forced mechanical response analysis was performed using a numerical computer model. As well as using the as-built technical drawings the behaviour of the model was tuned to replicate the findings of in-field vibration measurements taken upon both the piping and the bracing. Tuning a piping model to replicate the dynamic behaviour of an operating piping system is not a trivial undertaking. Within this paper the effect of various factors that were given special attention in tuning (matching) the computational model will be discussed.

Attention was given on how to ensure that the correct mechanical mode shapes were present in the model and that they were excited to the same level as in the field. These mode shapes were identified from the vibration measurements taken using a three-axis accelerometer. Factors such as equipment weights within the piping, and gaps and stiffnesses in the supporting deviate to varying degrees from those envisaged at the design stage in any piping system. Consequently the mechanical resonance modes predicted by the numerical model, initially based on the as-built technical drawings, exhibited some differences from those measured in the field. This was in terms of their shapes but also their response at a given excitation frequency.

In tuning the model the stiffness of the spring loaded guide supports, both laterally and axially had to be varied, as well as the stiffness of the bracing of the small bore branches. Only by modifying these values was it possible to match the vibration amplitudes seen in the field with the computational simulation of the piping system. It is impossible to include these factors at the design stage and they are addressed by the requirement that all mechanical resonance modes should be above 2.4 times the compressor rotational speed. However unintentional installation factors could result in this margin not being met in the field, and thus this additional modelling step with a tuned model is required for determining the stress level and the margin of safety.

The output of the study was a robust set of conclusions to the operator of what changes should be made to ensure there was sufficient margin to prevent cracking in the line. The vibrations in the header lines were reduced using rigid supports where possible, given thermal expansion of the system, which have far fewer unknowns in their installation in the field than supports with pre-loaded springs. Additionally recommendations were given for the bracing and gussets on the small bore instrumentation lines so they were less sensitive to vibrations in the header.

In sharing this study though the intention is to increase the awareness of the factors that need to be considered when tuning a numerical piping model to replicate the field experience under a dynamic loading such as pressure pulsations. Thus improving the robustness of numerical simulations used for assessing potentially critical situations in the field. It is noted that the presented method is not as detailed as an Operating Deflection Shape (ODS) analysis of the system or an analysis in which the mechanical natural frequency and damping were determined directly. The method presented here though is easier to apply and is suitable for indicating relative improvements to the system.

Introduction

During operation noticeable vibrations were observed by the operator of a reciprocating compressor plant. The vibrations had been noted over a significant period of time and there was a concern that they may ultimately lead to fatigue. The vibrations were observed in both the main large bore run piping as well as in a number of the small bore instrumentation branches. Vibrations are always a potential risk in reciprocating compressor piping [4] [5]. The request from the operator was to assess if the observed vibration levels and resulting stress levels were within allowable design limits. The outcome from the study for the operator was a series of recommendations, where necessary, for mitigating the fatigue failure risk.

The aim of this paper is however not to discuss the project and conclusions for the operator, but instead the focus will be on the complexities of tuning a dynamic computational simulation to match the measured vibrations in the field. Here three of the part models used for conducting the study are presented, with the intention of introducing the reader to different factors to consider, and the impact of these uncertainties on the results. It is noted that the presented method is not as detailed as an Operating Deflection Shape (ODS) analysis of the system [6] [7] or an analysis in which the mechanical natural frequency and damping were determined directly. The method presented here though is easier to apply and is suitable for indicating relative improvements to the system. The paper closes with an overview that will helpfully assist an engineer in conducting a robust analysis.

System Overview

The system under investigation had three double acting compressors arranged in series of which two were in use at any one time. Each compressor provided two stage compression with an air cooler located in the inter-stage loop. The reciprocating compressors had a running speed of 298RPM or 4.96Hz. As the compressors were always running at 100% part load the largest pulsation amplitudes were arising at a frequency of 9.9Hz. At the time of installation a pulsation analysis had been performed which showed that all of the piping mechanical natural frequencies were above 15Hz (3 x the running speed), which was predominantly achieved through the use of spring loaded guide supports.

During the site visits, visual inspection revealed observable piping movements especially immediately downstream and upstream of the pulsation bottles, and in the small branch connections. Vibration measurements were made in these regions. The measurements were made using a tri-axial accelerometer, with a sampling frequency of 48 kHz, which was connected by a magnet to either the piping or a pipe support. Given the highly-explosive nature of the process gas and that the isolation had to be removed to permit the measurements it was desired by the operator to keep the number of measurement points to a minimum.

The systems that will be discussed in this paper are as follows:

- Discharge side of the interconnecting line immediately downstream of the compressor
- The second stage discharge line
- A small bore branch located in the interconnecting line

Each of these systems will be introduced and discussed separately and the salient features in matching the vibration measurements will be explained.

Computation Modelling Approach

The dynamic computation simulation has been conducted using the piping stress analysis software CAESAR II [3]. CAESAR II is a FEA package using beam elements, which is appropriate given that the resonance modes at the frequencies of interest are all beam type modes (and not shell modes). The piping model was split into small elements, 3 to 4 pipe diameters in length, to ensure that there was sufficient resolution to capture the shape of the mechanical eigenmodes.

The model of the compressor piping was built according to the received piping isometrics, and the routing and supporting were verified during the site visits to conduct the vibration measurements. The insulation weight was included and the weights of the valves and flanges were taken from typical design data given their nominal diameter and pressure class. Only the structural steel supporting frames were included in the models which in the experience of the authors could not be considered be rigid (for instance insufficient stiffness in the plane of the applied load) and thereby they could have a significant impact on the calculated mechanical eigenmodes.

For this study the modal and harmonic solvers in CAESAR II were used. The former determines the natural frequencies of the piping system whilst in the latter the response of the system to a (series of) sinusoidal load(s), or

displacement(s), of a given frequency and phase angle is evaluated. A harmonic solver calculates the stress level at every phase angle of an applied sinusoidal load, from which the most critical phase angle based on the maximum stress amplitude was automatically selected. When this automatic selection was found not be sufficient then the phase angle at the location of interest was selected manually. The mechanical damping coefficient for the harmonic simulations was 0.03, a value in accordance with the range recommended in piping design practice, for example the design code EN13480.

System 1: Interconnecting line

The system is shown in Figure 1 and runs from the discharge side of the compressor to the inlet of the air cooler. During the site visit it was seen that there were noticeable vibrations in the piping close to the discharge bottle and near the support frame, as highlighted in Figure 1. Measurements were taken at these two locations, where Location 12 was on the support frame just below the pipe show and Location 11 was on the rest support underneath the flange connection with the bottle exit nozzle. The measured vibration amplitudes are shown in Table 1. It is seen that the rms velocity exceeds the Energy Institute [1] T7.2.2 guidelines. It is noted the compressor bottle is not included in the model as the focus of the study at the request of the operator was the piping, and the directional anchor immediately downstream of the bottle nozzle meant that the bottle flexibility had no impact on the mode shape at measurement location 12.

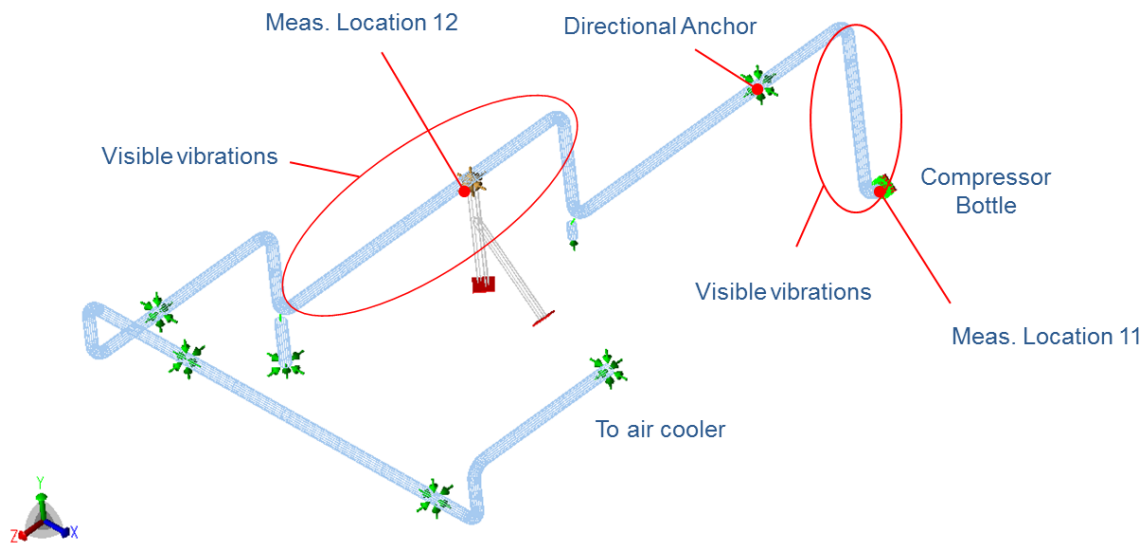


Figure 1: Overview of System 1 with measurement locations.

Table 1: Vibration amplitudes in System 1, peak at 10Hz is shown.

Location	Axes	p-p disp. Measured (mm)	rms velocity (mm/s)	EI 'problem' rms velocity* (mm/s) [2]
Meas. Loc. 11	X	1.8	38.7	23.8
Meas. Loc. 11	Y	1.6	35.3	23.8
Meas. Loc. 11	Z	1.7	36.2	23.8
Meas. Loc. 12	X	0.5	10.7	23.8
Meas. Loc. 12	Y	0.5	10.7	23.8
Meas. Loc. 12	Z	1.0	21.8	23.8

*allowable at 10Hz.

Applying the displacements to the model

The first stage in trying to match measured vibrations was to apply the displacements shown in Table 1 with all the spring loaded guide supports as stiff rigid supports which fully restraint dynamic motion. On running the model it was seen that the stress levels remained within the fatigue design limit at all locations, however measurements were only possible for two discrete points and it is needed to extrapolate these results to other locations for example downstream of the support at location 12.

To extrapolate the results it was required to determine the magnitude of the underlying unbalanced forces that are causing the vibration. Here the unbalanced forces were calculated using the worst case pulsation amplitudes at 10Hz per straight pipe section (between elbow pairs) from the earlier third party pulsation analysis of the system. The unbalanced force was then modified if the length between elbow pairs was shorter than the wavelength of a 10Hz pulsation.

Running the model with these pulsation amplitudes did not result in any significant vibrations at location 12, with those in the axial direction (Z) being an order of magnitude smaller than those in Table 1. It could be that the compressor pulsation amplitudes were higher than calculated, but as displacement is linear with applied force it is unlikely to provide the full explanation as they were unlikely to be ten times larger than the calculated pulsation amplitude at the design stage.

Investigation therefore moved to the spring loaded supports and the stiffness of the support at measurement location 12. As the vibration measurement at location 12 was taken on the structure (and not on the pipe) the spring support at this location must be providing a reasonable degree of axial restraint. It was found however that if the spring support further downstream was made free then a resonance mode existed in that section with a frequency of 9.5Hz. This mode shape is shown in Figure 2. When the pulsation loads were applied for this case the observed displacement at 10Hz matched the displacement in the Z direction at location 12 from the measurements.

It can be seen in Figure 2 that the compressor bottle has not been modelled, and has instead been replaced by an anchor for the modal analysis. This was done as the directional anchor (rather than spring support) located between the compressor bottle exit and measurement location 12 (see Figure 1) meant that the mode shape at measurement location 12 was independent of flexibilities in the compressor bottle.

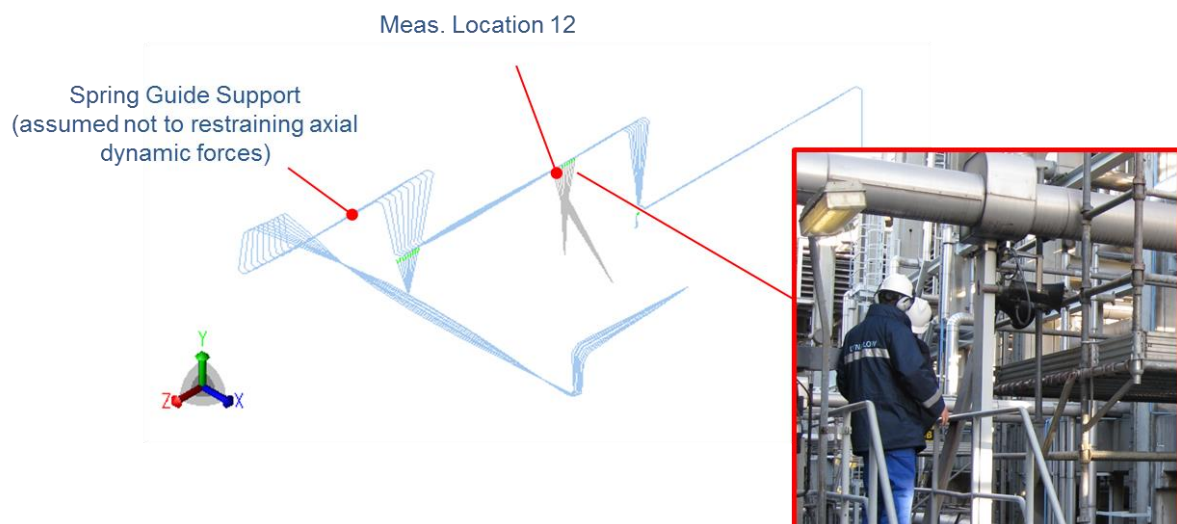


Figure 2: Mode arising at 9.5Hz if spring guide support assumed not to be restraining dynamic axial forces, photograph shows the structural frame at measurement location 12 (note angle of photograph is mirrored).

Conclusions from Model 1

This example has shown that the spring guide support cannot necessarily be presumed to be providing full restraint against dynamical axial loads. It is unlikely that this was providing no restraint to the axial forces and given the large displacements seen at the compressor bottle discharge (location 11) it was suspected that the pulsation amplitudes were also higher than those simulated at the design stage. Thus the measured vibration appeared to be a combination of larger than designed pulsation amplitudes and a non-ideal spring guide support. This concept will be taken further in the following model.

System 2: Compressor Discharge Line

This model is the discharge line downstream of the compressor bottle up until the connection with the general discharge header between the three compressors. An overview is shown in Figure 3, with the three locations at

which the vibration levels were measured. The vibration amplitudes are listed in Table 2 and these are compared to the 'problem' rms velocity limit from the Energy Institute guideline.

As for model 1 the displacements shown in Table 2 were applied as harmonic displacements at the measurement locations. In this case though it was found that the stress amplitudes near measurement locations 22 and 23 exceed the allowable value for the fatigue design curve. Additionally the displacements at measurement location 21 lead to significant stresses (which would have caused fatigue failure) in the compressor nozzles. The harmonic displacements from measurement 21 were thus applied as a boundary condition as the vibrations arose partially across the entire compressor skid and not only the bottle and piping. It is reminded here that the operator wanted to keep the focus of this study on the piping only.

Table 2: Vibration amplitudes in System 2, peak at 10Hz is shown.

Location	Axes	p-p disp. Measured (mm)	rms velocity (mm/s)	EI 'problem' rms velocity* (mm/s) [2]
Meas. Loc. 21	X	1.5	32.6	23.8
Meas. Loc. 21	Y	2.3	49.5	23.8
Meas. Loc. 21	Z	2.8	60.7	23.8
Meas. Loc. 22	X	0.8	16.3	23.8
Meas. Loc. 22	Y	0.8	16.7	23.8
Meas. Loc. 22	Z	1.5	32.6	23.8
Meas. Loc. 23	X	2.1	46.7	23.8
Meas. Loc. 23	Y	0.6	12.6	23.8
Meas. Loc. 23	Z	0.8	17.5	23.8

*allowable at 10Hz

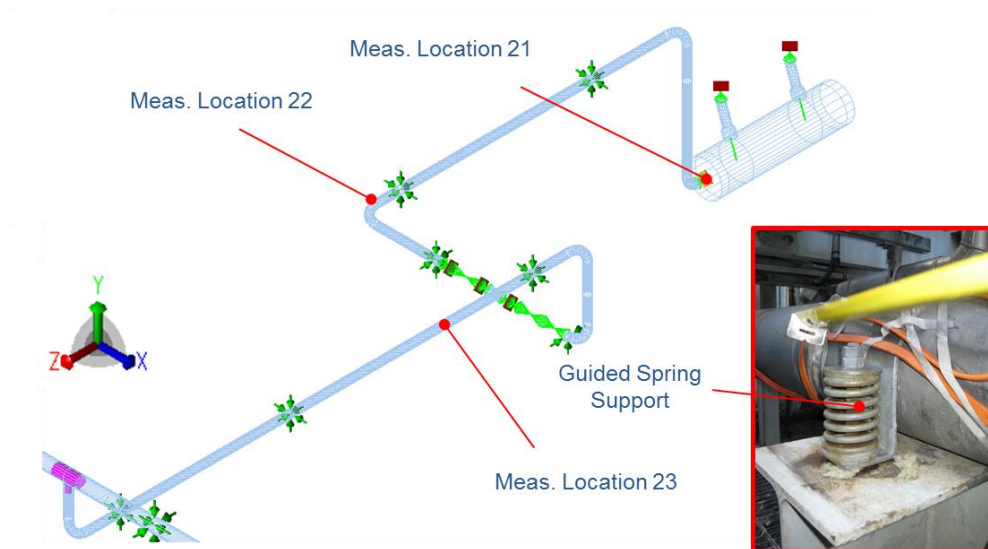


Figure 3: Overview of System 2 with measurement locations, photograph shows the design of the guided spring supports.

Matching the displacements

To tune the model, the design stage pulsation amplitudes, as in System 1, were used to provide the unbalanced forces acting between all elbow pairs. By doing so the vibration amplitudes at points other than the measurement location could be estimated. The unbalanced forces were all taken to act in phase, as detailed phase information was not available from the earlier third party pulsation study.

The simulated displacements at measurement locations 22 and 23 were compared to the measurements considering the spring guide supports as perfectly stiff or flexible. However in both cases the calculated displacement

was lower than that from the measurements. Reviewing the mechanical resonance modes when the spring loaded support was assumed to provide no axial restraint it was seen that there was a mode at 8Hz (shown in Figure 4) that could be causing the measured vibration amplitudes.

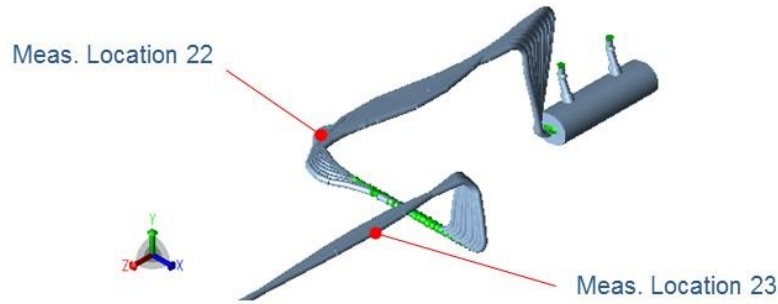


Figure 4: Mode arising at 8 Hz if spring guide support assumed not to be restraining dynamic axial forces.

In Figure 5 the impact of varying the stiffness and the axial restraining capacity is shown, here it is seen how the frequency of maximum response varies as the axial stiffness is increased. In the computational model this was done by introducing the stiffness friction factor (FF). This is an arbitrary calibration factor for a dynamic system used in the following formula to create the dynamic frictional stiffness ($K_{fric,dyna}$) [3]. Where μ is the static friction coefficient at the applicable support location, F_{vert} is the vertical static load and δ_{static} is the calculated static displacement at the support location.

$$K_{fric,dyna} = FF \cdot K_{fric,static} = FF \cdot \frac{\mu \cdot F_{vert}}{\delta_{static}}$$

The effect of varying the friction factor is to stiffen the system and the resonance frequency increases. This can be seen in the two graphs below for measurement locations 22 and 23. Here it is seen that frequency increases as the friction factor is increased. Here it is seen that increasing the friction factor (FF) to 10 means that the displacement measured at location 22 is the same as that measured. At location 23 however the situation is more complicated as increasing the resistance of the spring support to a FF of 5 provides the best match in frequency response but the displacement is lower than measured.

A further option was to review the stiffness of the guide supports. Initially as for the limit stops these were also modelled to be stiff, however calculating the stiffness of these using a shell FE model this was found to be 1.6kN/mm. As shown in Figure 5, applying this value at all of the guide supports means that the resonance frequency changes and now the displacement at location 23 exceeds that measured.

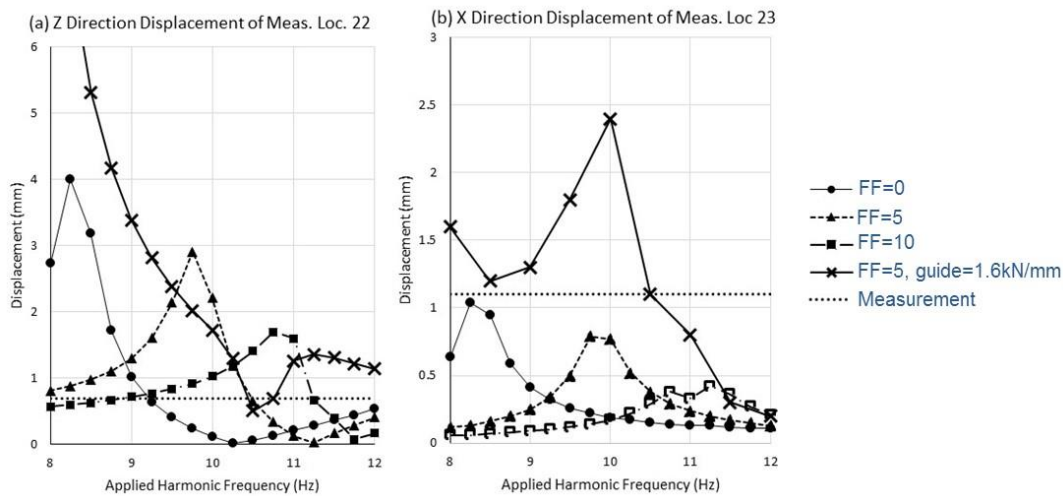


Figure 5: Displacement at measurement locations 22 and 23 for varying stiffnesses in the limit stop and guide.

Conclusions from model 2

Here it has been shown that it is necessary to modify the support stiffnesses (rather than take them to be rigid) to obtain a best fit with the measured vibration levels. However, as for system 1 this is complicated by the fact that the unbalanced forces in the system are based on design pulsation levels. The solution offered to the client for both model 1 and model 2 was to increase the stiffness by ensuring the stiffness of the spring supports and where possible, given thermal expansion, to introduce directional anchors. The sensitivity of the system in this approach clearly justifies the criteria of API 618 to ensure that all resonance modes are above 2.4 times the compressor running speed.

System 3: Small bore connection

Subsequent to the measurements on the header the vibration levels in the small bore lines were checked. A typical example of one of these small bore lines is shown in Figure 6(a). The small bore connection has a ND of ½”, is gusseted to the header and has two 1500lb valves. Between the two valves it is restrained to the steel bracing by means of a U-bolt, which was taken to restrain lateral movement only. The measurement locations are shown in Figure 6. Here it is seen that the location 31 is found on the header and 32 and 33 are on the steel frame.

The amplitudes of the measured vibration for this small bore connection are shown in Table 3 for both the compressors that were in service at that time (B and S). These are the displacements recorded at 10Hz, there is also amplification between the header and the branch which indicates that a resonance mode within the branch is being excited.

Calibrating the model

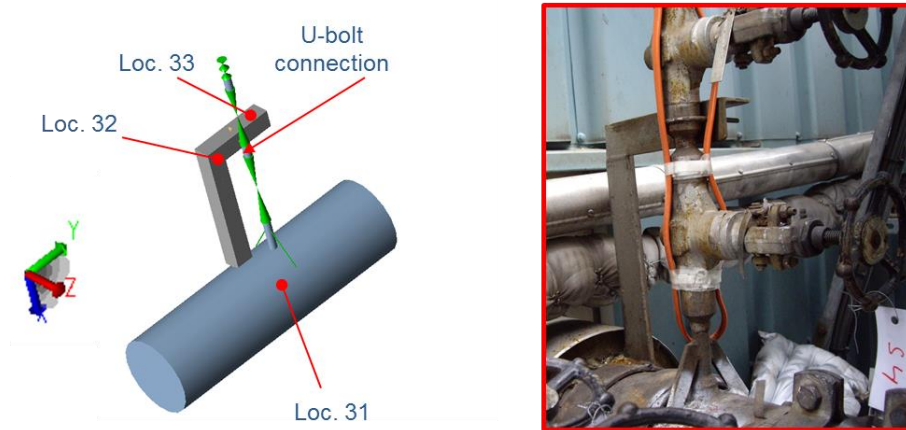
When calibrating the models for the measured vibrations what is important is the combination of the absolute amplitude and the amplification compared to the header. For instance a large amplitude on its own does not mean a large stress in the small bore connection as if it is moving in-phase with the header then no bending stress is generated. Similarly a large amplification is irrelevant if the displacement amplitudes are small. By reviewing the mechanical response when applying different frequencies in the computational analysis it was concluded that the header and branch excitations were in-phase unless the mechanical resonance frequency was exactly matched.

To determine which mechanical resonance mode was being excited the modal solver was used to determine the in-plane and out-of-plane resonance modes. These are shown in Figure 6 (b) and (c), and both have a frequency of 31Hz, which is significantly above the response at 10Hz seen in the vibration measurements.

The resonance frequency of a branch connection is a function of $\sqrt{k/m}$ where k is the stiffness of the branch and the mounting of the bracing on the pipe, and m is the mass of the components such as the valves in addition to the bracing and piping. To establish the effect of different parameters on the resonance frequency a number of these parameters were varied as listed here:

- Case 1: Baseline, all valve weights are as per typical information.
- Case 2: As Case 1, but with branch U-bolt assumed to work as a three way stop (also restrains axial movement).
- Case 3: As Case 1, but doubling of the weight of the valve (for example due to uncertainties such as the control equipment).
- Case 4: As Case 3 but with increased flexibility of the connection between the bracing and the run pipe, (for example due to a loose buckle).
- Case 5: As Case 4 but with the flexibility of the gusset/weld connection reduced.

(a) Measurement locations and photograph of the small bore connection



(b) Out and in-plane resonance modes

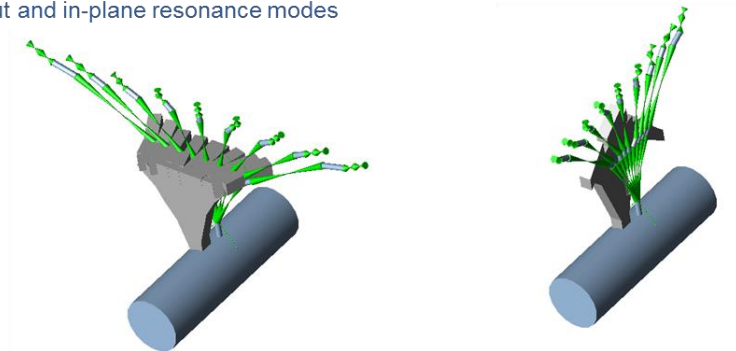


Figure 6: Overview of System 3 with (a) measurement locations and (b) showing the resonance modes.

Table 3: Vibration amplitudes in System 3 at 10Hz

	Frame displacement amplitude (mm)*	Increase compared to header (-)
Comp B – in plane	0.18	2.7
Comp S – in plane	0.04	11
Comp B – out of plane	0.13	1.4
Comp S – out of plane	0.28	2.0

*Mean of points 32 and 33

The effect on the system response for a given excitation frequency is shown in Figure 7. In Figure 7(a) the response of the branch connection to in-plane amplification is shown whilst in Figure 7(b) the response to out-of-plane amplification is shown. The location of the maximum amplification indicates the resonance frequency. It can be seen from Figure 7 that introducing flexibility into the branch and frame connections leads to a large reduction in the resonance frequency. The amplification for Case 5 (and for Case 4 out of plane) is now similar to that shown in Table 3

for 10Hz as it shows approximately a two-fold increase.

Conclusions from model 3

It has been shown here how the response at 10Hz could be matched, however it cannot be certain that this accurately models the branch at all frequencies. Given the uncertainties the choice was taken to provide a robust solution as the current vibration levels in the gusset weld toe were close to the design fatigue limit. The solution was to introduce additional in-plane bracing and additional out of plane bracing. This had the effect of increasing the resonance frequency (for Case 5) to 30Hz for both modes and thereby significantly far away from the excitation frequency.

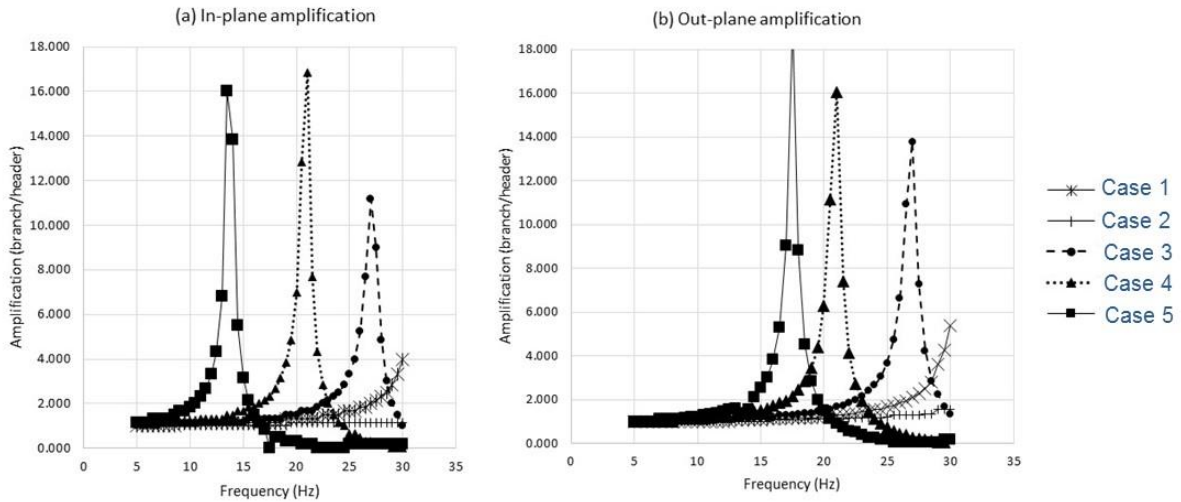


Figure 7: Effect of different factors on the resonance modes.

Conclusions and Recommendations

As stated at the beginning of this paper the aim here is not to explain the solutions for this specific reciprocating compressor plant, but more to demonstrate and explain the factors involved in trying to match vibration measurements in the field to a computational model of the system. Of course in a perfect computational model the vibration amplitudes would match perfectly, but a perfect model relies on exact knowledge of the system, which given uncertainties including corrosion, support stiffnesses, small clearances and equipment weights is not readily available for a practical study.

A possible method to avoid this uncertainty is to take sufficient measurement points through the system. The arising modes shapes and associated stresses can then be calculated and if they are within the fatigue allowable then no further analysis is required. In this the stiffness of the supports (including the spring guide supports) are all rigid and that the measurement needs to be made at the location of maximum vibration amplitude. However as shown in this study this could lead to large stresses as the flexibility of the modelled system is significantly reduced compared to the case with some flexibility in the supports.

If the stresses are excessive or if intermediate points need to be calculated then the forces acting on the piping (due to the pulsations) should be estimated. Even if a computational pulsation study is available there is no certainty that this is an accurate representation of what is occurring in reality, as appeared partially to be the case in the analysis presented here. The measured vibration is a function of the product of the applied force and the DAF (Dynamic Amplification Factor). If the applied force is unknown then calculating the DAF and the precise mode that is being excited is not possible. This difficulty is implicitly addressed by the requirement in API 618 [2] that the mechanical resonance frequencies should be greater than 2.4 times the rotational speed of the compressor, as the DAF is minimal if there are no resonance frequencies to excite.

The question then arises how do vibration measurements assist in solving a vibration issue? In the cases shown here the DAF was not minimal as the vibration amplitudes could only be achieved if a mechanical eigenmode

was excited. The methods shown here, where uncertainties in support and connection stiffnesses have been varied, provides a simple method to identify the resonance mode that has been excited. The authors note that more detailed analysis, such as Operating Deflection Shape or Mechanical Mode Shape analysis could have been possible if significantly more vibration measurements and post processing had been performed. The current method though is easier to apply and was more desirable to the operator in this instance given the explosive nature of the process fluid. The method was able to provide targeted recommendations for the operator to reduce the relative vibration levels.

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