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Considerations for Flexibility Analyses of Piping Systems in Vibration Applications

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Presented at:

Gas Machinery Conference 2017

Pittsburgh, PA

Abstract

Designers and engineers of piping located near rotating or reciprocating equipment should aim to prevent failures from excessive vibration, which can be mitigated if a proper design philosophy is adopted. However, vibration issues are often ignored in static analyses because codes do not address vibration risks in a detailed manner and many designers are not aware of possible piping vibration issues and how to avoid them.

The source of vibration typically varies from one or more of the following: impact (such as water hammer or sudden valve closure), pressure pulsations, flow vortices, compressor resonance and wind. Compared to non-vibrating piping, there are fundamental differences in performing a flexibility analysis for piping systems subject to vibration. Special considerations and techniques are required to account for the unique characteristics of vibrating piping and to fulfill the code requirements for a flexibility analysis. Important elements of such an analysis include pipe support spacing, pipe support type, support structure stiffness, design pressure, temperature range and interactions between piping and its support.

This paper discusses several considerations for performing a flexibility analysis of piping systems subject to vibration and is of interest to designers, engineers and pipe stress analysts working with rotating and reciprocating equipment design.

1. Introduction

The primary goal of a piping flexibility stress analysis is to ensure safety. It aims for an optimal design for both piping and structure. A piping stress analysis is necessary in order to limit stresses below code-allowable limits (within the safety zone). A piping stress analysis is performed to avoid:

- Excessive flexibility
- Leakage at joints
- Resonance at any imposed vibration frequency
- Excessive movement due to thrust
- High loads on the pipe support and structures
- Limiting nozzle loads of the connected equipment [1]

Pipe vibration and pipe stress analyses are engineering disciplines that require the balancing of a contradicting set of solutions. Piping vibration occurs at resonance condition, which is the periodic excitation of the system when periodic external forces are at or near the natural frequency of the system. Therefore, piping subject to vibration requires increased stiffness in order to raise the mechanical natural frequency (see API 618 requirement of 2.4 of running speed) by engineering support spacing and selecting support types [2]. On the other hand, adding flexibility (reducing stiffness) is often required in order to meet code requirements and allowable nozzle loads by adding thermal expansion loops or designing flexible joints (Figure 1).
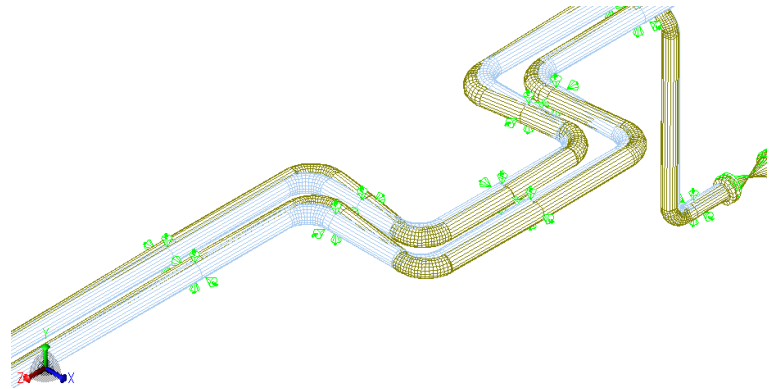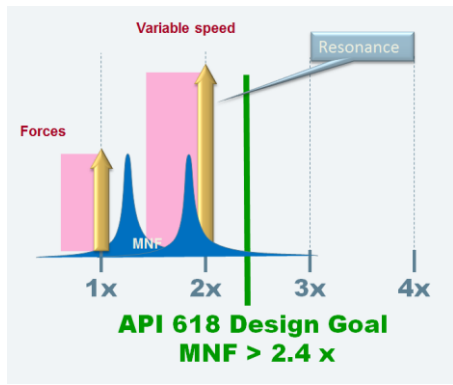
Figure 1. Design requirements of vibrating piping system; increasing dynamic stiffness (left), increasing flexibility (right)

When engineering personnel performs a piping stress analysis for vibration piping, special considerations are involved because of the unique characteristics of the vibration, code requirement, techniques in the analysis. This paper will provide over view of some components for piping stress analysis such as vibration, pipe support spacing, support type, pipe-clamp interaction, shaking forces.

Because reciprocating compressors or pump packages are major sources of piping vibration, a flexibility analysis is required for piping stress analysis. This paper will mainly focus on the above ground piping around the reciprocating compressor package and flexibility analysis.

2. Characteristics of vibrating piping

To describe the unique elements involved in piping stress analysis for vibrating piping, it is necessary to first distinguish vibrating piping from non-vibrating piping. Since the piping designer has numerous other factors to consider when determining a piping system layout, it is not suggested here that all piping systems should consider vibration analysis during the design stage. The engineer should spend time instead to ensure that the piping system will not be in the neighborhood of vibration excitation.

There are two main types of vibration behaviors, free and forced. In free vibration, the system is excited by external transient impulses and vibrates under no external forces. In forced vibration, the system vibrates under the external periodic forces; this type of vibration is commonly encountered in piping vibration situations in the oil and gas industry [5].
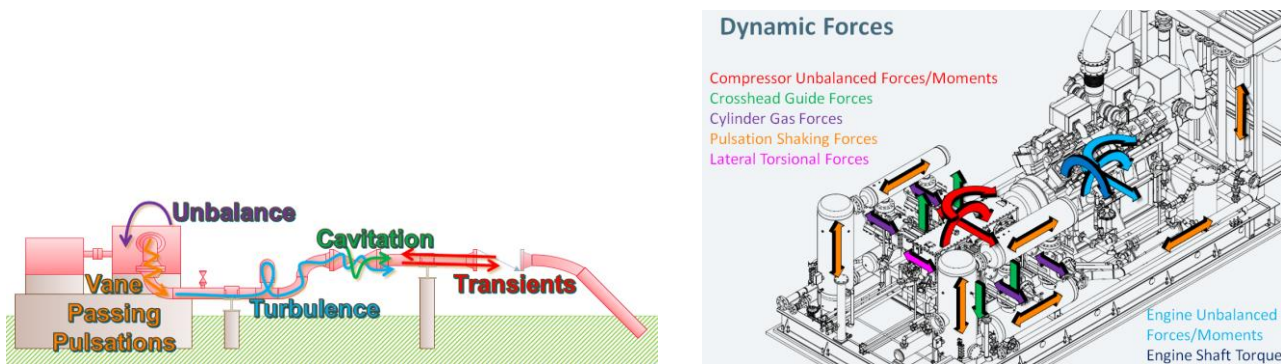

Figure 2. Examples of excitation forces from rotating or reciprocating machinery

A typical example of forced vibration is a piping or structural system exposed to periodic excitation forces from rotating machinery or a reciprocating compressor (Figure 2). Rotating equipment is a significant source of vibration due to the unbalanced mass in its rotating parts. If the rotating speed is near the natural frequency of any nearby piping system,

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resonance will occur and potentially lead to failure of the piping system. Reciprocating compressors also generate a periodic pressure pulsation at the frequency of its running speed (single-acting cylinder operation) or two times the running speed (double acting). If this frequency approaches the acoustical length of the piping system, large pressure pulsations will occur and act as periodic forces on the pipe elbows and other components. These forces can be transmitted directly to other vessels, structures or foundations.

Balancing rotating parts, adding pulsation dampers to the system or other techniques can reduce the vibration excitation. In addition, to avoid resonant vibration, the natural frequency or acoustical length of the piping system should be designed so that it does not match the speed of rotating machinery or the pressure pulsation frequency of reciprocating compressors. Optimizing the line configuration and choosing the right support types and spaces are critical steps to control potentially excessive vibration in the piping system; the piping stress analysis plays a major role in this process.

Figure 3. Examples of support location with thermal flexibility

3. Requirements of a piping stress analysis

Engineers are tasked to select an appropriate pipe size, thickness and material for the safe design and construction of a piping system under worst-case operating conditions. Available codes provide guidance in form of a simplified approach for determining stress levels and other piping design criteria [3]. Designers need to understand the code requirements (standards or regulations) in order to restrain a piping system properly. These requirements are often the technical basis of the piping system design and analysis.

For reciprocating compressor packages, API 618 section 7.9.4.2.6 'Mechanical review and piping restraint analysis' states that thermal flexibility effects should be considered in note 1. The commonly-used standard which governs a flexibility analysis is ASME B31.3. ASME B31.3 '301.5.4 Vibration.' It states that "piping shall be designed, arranged, and supported so as to eliminate excessive and harmful effects of vibration which may arise from such sources as impact, pressure pulsation, turbulent flow vortices, resonance in compressors, and wind."

Designers must have a fundamental understanding of the different types of stresses to perform a piping flexibility analysis accurately. The codes break down the types of stresses into primary, secondary and peak stress.

Primary stress is generated by imposed mechanical loadings. Primary stress is not self-limiting. Therefore, as long as the load is applied, the stress will be present. It will not diminish with time or deformation. Typical examples of imposed loadings are internal or external pressure and gravity.

Secondary stress is caused by constraints of a structure against displacements. The characteristic of secondary stress is that it is self-limiting. A piping system complies with imposed strain rather than imposed force. Distortion of a piping system subjected to a temperature increases is an example of secondary stress. When a piping system reaches its operating temperature, the bending strain in the elbow will increase until the distortion is over. High stress at the elbow will be reduced as the piping system experiences local yielding and deformation. It is related to temperature cycle conditions such as the plant starting up or shutting down.

Peak stress occurs at concentration points such as a pipe fitting or a weld by local discontinuities or abrupt geometry changes. Peak stress does not cause any distortion but can initiate a crack, which can cause a fatigue failure. These types of stresses can cause a piping failure with various loadings. Therefore, engineers perform a flexibility analysis to make sure that the design of the piping system meets the code requirement.

4. Code for flexibility analyses

Stress calculations for piping systems require many complex mathematical equations which can be solved with specialized computer software. The biggest challenge for the user is to provide the right data to the program. This includes physical properties such as layout and materials, stress intensification factors, piping component weights, support types and various loads. The user must have a working knowledge of the analysis software and understand several key aspects of how the analysis is to be performed. In addition, engineering personnel needs to understand the difference between the failure mechanisms involved in high- and low-cycle fatigue. Low-cycle fatigue is caused by significant plastic strain during each cycle, whereas high-cycle fatigue occurs within the elastic range, and its maximum allowable stress level is referred to as the endurance limit.

A flexibility analysis addresses excessive plastic deformation or instability due to low-cycle loads in the plastic range, ie, high stresses which cause low-cycle fatigue. ASME B31.3 302.3.5 (2014) defines a basic allowable stress at maximum metal temperature expected during the displacement cycle under analysis as S_h :20 ksi (maximum). According to B31.3 302.3.4.(c), stress should be considered under operating condition. S_c (allowable stress at the minimum metal temperature) is define in the same section. The ASME B31.3 allowable displacement stress range S_a is calculated by the following equation, based on the parameters S_h and S_c with f being the stress range factor, as shown in Figure 4.

$$S_a = f(1.25\,S_c + 0.25\,S_h)$$

The stress range factor varies from 1.2 to 0.15 depending on the number of cycles. In a flexibility analysis, the most commonly used value for loading cycles would be 7,000 ($f = 1.0$), which is about one cycle per day for 20 years. Therefore, the allowable stress range is based on the number of thermal cycles (shut down and operating) during the service life of the piping system.
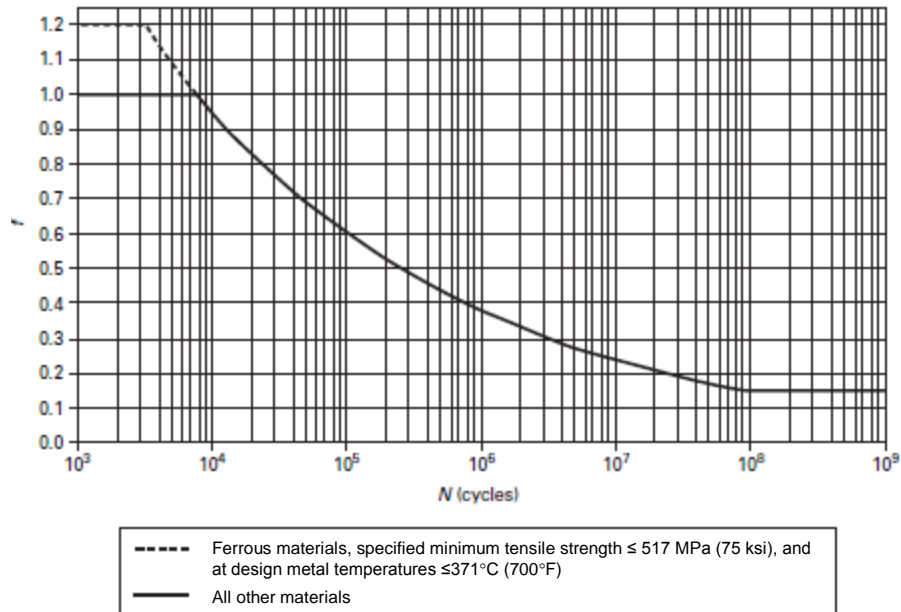
Figure 4. Stress range factor, f, from B31.3

In a reciprocating compressor system, the piping easily experiences more than 10^6 cycles in a single day. Some engineering approaches use a stress range factor below 0.15, however, this is not a practical approach because the allowable displacement stress range S_a is not designed for high-cycle fatigue. A high-cycle fatigue curve, typically known as the S-N curve, can be used to correlate stress levels with the number of cycles to failure in a vibrating piping system. For example, ASME boiler code Section VIII, division 2 specifies how to compute the number of design cycles. A fatigue curve for various materials can be generated empirically. The API 618 design approach step 3b suggests performing a forced mechanical response on the system to apply the allowable cycle stress criterion. Section 7.9.4.2.5.2.5.1 states that vibration shall not cause a cyclic stress level in the piping system in excess of the endurance limits of a material during design approach 3. The peak-to-peak cyclic stress range is presented in 7.9.4.2.5.2.5. The cyclic stress range for carbon steel with an operating temperature below 700 °F (371°C) shall be less than 26,000 psi (pk-pk), which does not consider a stress intensification factor (SIF). Therefore, it is important for engineers to have an approach that will allow for a practical use of this value in the context of a real analysis. Typical safe vibratory stresses would be below 1,500 to 2,000 psi (0-pk), once the SIF is considered during the design. The stress range factor for vibrating piping should be selected based on the temperature cycle from installed temperature to operating temperature such as plant start-up and shutdown.

In summary, the allowable stress range in ASME B31.3 does not address high-cycle fatigue in piping systems. The purpose of the flexibility analysis is to check the piping system for low-cycle fatigue.

5. Design conditions

The intended design and operating conditions are basic input parameters for the analysis and design of a piping system. ASME B31.3 301 lists various components for these design conditions, such as operating temperatures and pressures. Determination of these parameters varies from company to company. Experienced designers should include these conditions in their piping system designs.

Pressure mainly contributes to the hoop stress defined earlier and is the main design factor in determining the wall thickness of the selected pipe outside diameter and material. ASME B31.3 states that the design pressure shall not be

less than the pressure at the most severe condition of coincident internal or external pressure and temperature (maximum or minimum) expected during service. Therefore, design pressure shall be based on the highest expected operating pressure. For piping systems exposed to pulsation pressure conditions such as those associated with reciprocating pumps and compressors, some companies consider increasing the design pressure by more than the expected peak operating pressure as a safety factor.

Temperature is also an essential component for a flexibility analysis. The design temperature in a piping system is the temperature at which, under the coincident pressure, the greatest thickness or highest component rating is required. The maximum and minimum design temperature should cover the full range of anticipated operating temperatures. ASME B31.3 301.3 states that when establishing a design temperature, at least the fluid, ambient, solar radiation, and heating or cooling medium temperatures should be considered.

The most common method to control the safety factor for a piping flexibility analysis is changing the design temperature. It is commonly believed in industry that high design temperatures correspond to a more conservative assumption for a given analysis. However, this is not necessarily the case, as high design temperatures create overestimated strain, leading to high stress. As a result, more time and material for stress-reducing features such as expansion loops and special supports are required to meet the code-allowable stress level. High temperature can also easily overestimate the thermal growth and nozzle loads near sensitive equipment and machinery. The spool between suction bottle and scrubber generally has a short piping length and is typically within allowable stress limits under maximum operating temperature (Figure 5). However, using an unnecessarily high temperature will require a thicker nozzle repad in order to increase the allowable nozzle loads or, in the worst case, require a thermal loop which will then also require an additional support structure for the pipe. This is non-conservative and may lead to high pipe vibration due to low dynamic stiffness.
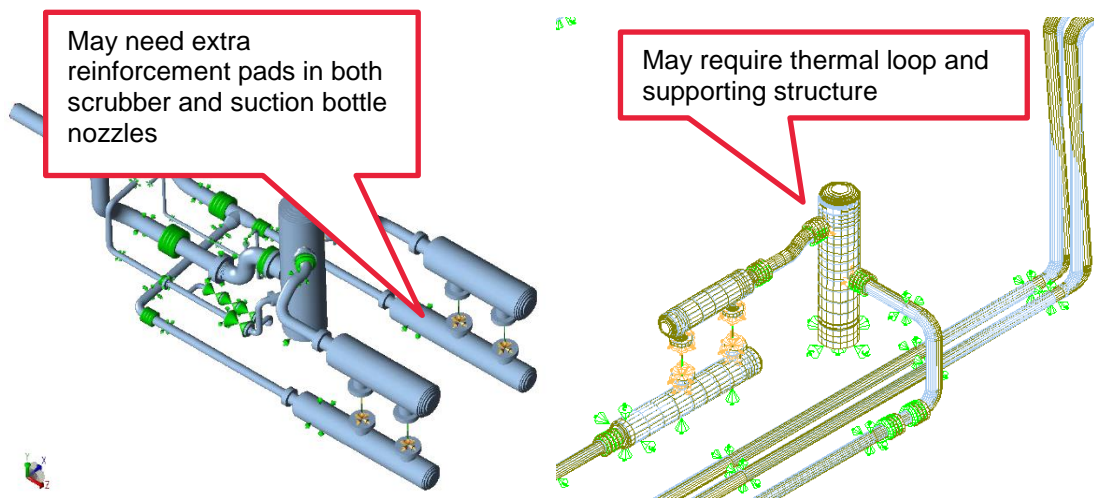

Figure 5. Examples of flexibility analysis models in reciprocating compressor

The design minimum temperature is the lowest component temperature expected in service. It is a commonly-held belief in industry that low temperature is a conservative assumption and sometimes, the material minimum temperature without impact test from a given code is used as the minimum design temperature for the flexibility analysis without considering the real operating temperature range of the equipment. This will create unnecessarily wide temperature ranges for the thermal analysis.

In a piping flexibility analysis, one more temperature is required for the analysis, that being the install temperature, which is based on the geographical location of the plant. The ambient temperature can be defined as the pipe metal temperature at the time of initial fabrication such as bolt-up of the flange at the nozzle or closure weld on vessel nozzles. A thermal flexibility analysis uses this temperature as the starting temperature for the evaluation of thermal stresses of piping systems, which experience either maximum or minimum temperature during the service.

A key word in these design conditions is "coincident," when considering which combinations of pressure and operating (or service) conditions to use in the analysis. As a general rule of thumb, the maximum design temperature should be set to no more than 5% above the maximum operating temperature of the fluid and should not include temporary conditions such as those developed during start-up, shutdown or steam-out. The minimum design temperature should be not more than 5°C below the minimum operating temperature for continuous operation. When operations are expected to be intermittent or involve shutdowns for extended periods of time, the minimum temperature can be the minimum ambient temperature. The installed temperature is established by a plant (or fabrication) location in a hot or cold climate.

6. Support type and modeling technique

A key component of a piping stress analysis is the selection of the type of supports to be used in order to control the various design and operating behaviors of the piping system as well as loads caused by pressure pulsations, wind, earthquakes, shocks, thermal movement and gravity. The main criteria for selecting support types are the function of the support, the magnitude of the loads, the type of fluid and the existence of piping insulation and space limitations. ASME B31.3 provides some guidance for pipe support type and material in Table 326.1. Engineering personnel can use MSS SP-58 to select the pipe support suitable for their application (Figure 6).

Rubber Gasket Joints for Ductile-Iron Pressure Pipe and Fittings. . . . . AWWA C111
Flexible Metal Hose [Notes (1), (6), and (7)]. . . . . BS 6501, Part 1
Pipe Hangers and Supports — Materials, Design, and Manufacture. . . . . MSS SP-58
Brazing Joints for Copper and Copper Alloy Pressure Fittings. . . . . MSS SP-73
Standard for Fire Hose Connections . . . . NFPA 1963

Figure 6. Table 326.1 , ASME B31.3

The most common types of support in a typical piping system are weight supports and rigid restraints (Figure 7). Weight supports provide support against pipe loads in vertical direction. The most commonly used weight support is a shoe or rest type support, which supports only downward forces due to gravity. The supports lose their function when thermal expansion or vibration causes the pipe to uplift since shoe type supports do not stop the pipe from moving upwards. When the vertical thermal movement is expected to be negligible or when the piping does not experience any vibration, a simple rest support can be adequate. A spring-type support is another commonly used type of weight support. In many cases, spring supports are used to compensate for movement differences between operating temperature and the installed temperature on load-sensitive equipment such as pumps or turbines.

Rigid restraints such as line stops or guides are designed to mainly resist thermal loads and movement at the support location. The most common design for a line-stop support is a welded lug on the pipe shoe or pipe. Lugs transmit loads to the support steel from the pipe in the resisting axial direction. Therefore, these lugs must be sized according to the loads with pipe strength. Guide supports are for resisting the piping movement in lateral direction. It is also necessary to consider the friction between the lug and the steel due to thermal movement.

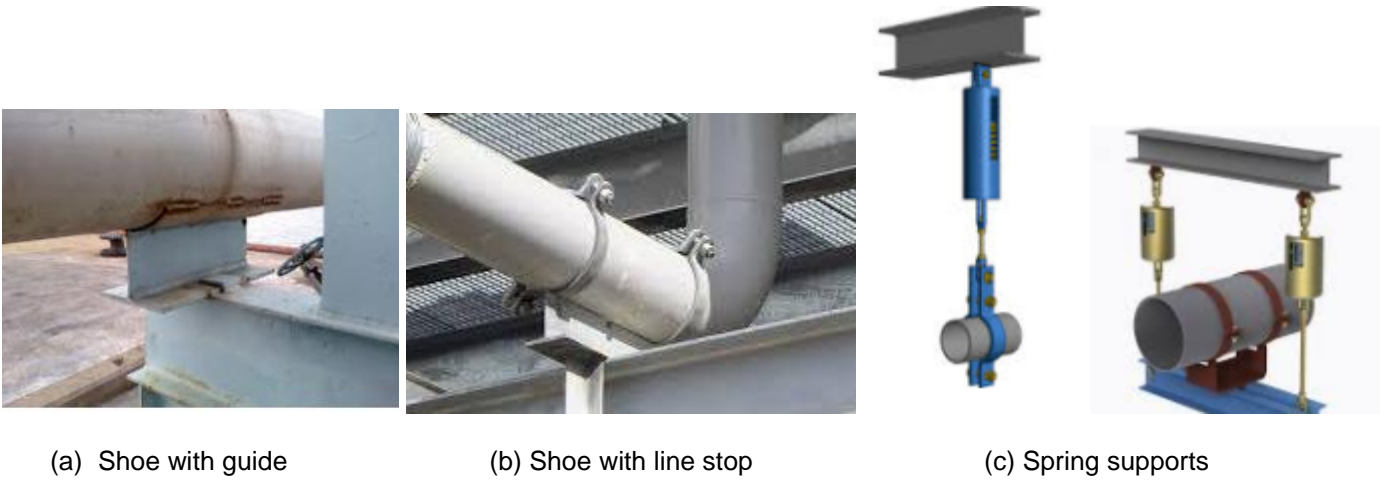(a) Shoe with guide (b) Shoe with line stop (c) Spring supports

Figure 7. Examples of typical support for piping system

However, to avoid piping vibration, the supports must dynamically restrain the piping. Note 2 in API 618 7.9.4.2.3.6 states that piping supports must have enough mass or stiffness to enforce a vibration node at the restraint location. Simple supports, hangers, springs, line stops and guides often cannot achieve this requirement. Instead, a clamp-type support can provide resistance in three directions (vertical, horizontal and axial direction) by supporting the pipe directly with a u-shaped steel strap (flat bar) (Figure 8). Vertical and horizontal forces are transmitted through the steel from the piping. Frictional and clamping forces restrict movement in axial direction.

U-bolt

Omega clamp

Standard vibration clamp

Vibration clamp with specialized damping material

Vibration clamp with damping material for high-temperature applications

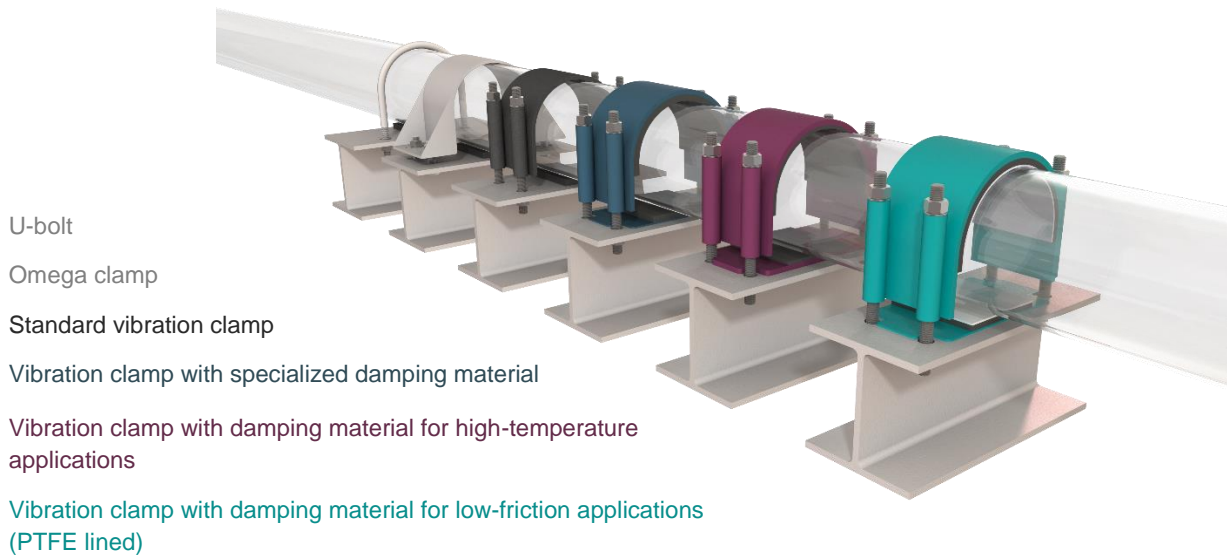Vibration clamp with damping material for low-friction applications (PTFE lined)

Figure 8. Various types of pipe clamps

Another similar type of clamp is the U-bolt. A U-bolt provides restraint in two directions: tensile and shear, with tensile capacity being much higher than shear capacity. It should be used where the lateral load is minimal since it cannot restrain the piping in three directions (Figure 9).

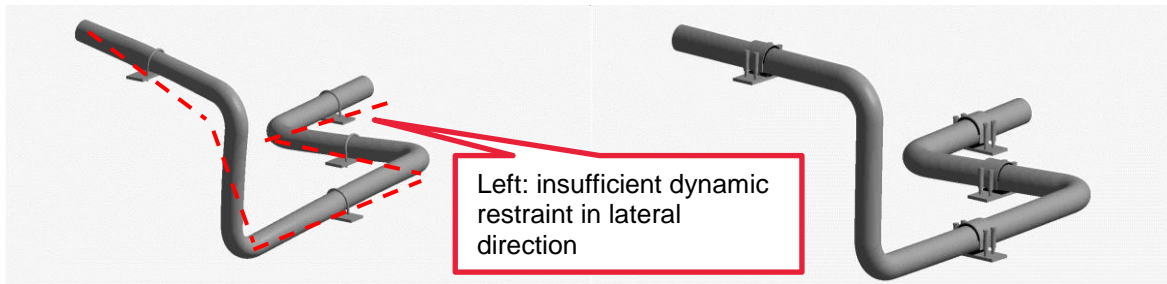Left: insufficient dynamic restraint in lateral direction

Figure 9. U-bolt vs flat-bar type clamp

Another drawback of the U-bolt is fretting, which can generate a significant groove into the pipe when a U-bolt loosens under vibration (Figure 10). Due to their small contact area, they tend to loosen and aggravate the fretting in areas of high vibration. Frequent inspections to identify and clamp loose U-bolts before they can damage the piping would be acceptable in most cases.

Figure 10. Fretting of U-bolt

One of the most important parts of piping flexibility analysis is the accurate modeling of boundary conditions. A support is the boundary condition of the piping system. For example, shoe supports restrain the pipe in the downward direction only. Restraints may act in more than one degree of freedom, depending on the type of support. Typical restraint types such as shoes, anchors, line stops and spring hangers are generally well defined. Clamp-type supports in a flexibility analysis also require accurate modeling of the boundary conditions. Each company has their own method to model clamp-type supports in their analysis. Therefore, this is also an area with potential to generate errors or unnecessary flexibilities due to a large variety of restraint methods. When conducting a flexibility analysis, analysts should consider the effects of the clamp support type and model on the piping system.

A clamp is used to hold down the pipe, and will allow a certain amount of pipe slip in axial direction once the axial force exceeds the hold-down (clamping) forces. The piping design will depend on whether the analyst considers pipe slip, which adds flexibility to the piping system, or not. The magnitude of the pipe slip is controlled by the number of bolts and the prescribed bolt torque at the clamp. The required bolt torque and numbers should be engineered for different nominal pipe sizes. The engineering firm that performs the piping stress analysis should have a clamp force table depending on the type of support (see Table 1 for example).

Table 1. Example of design loads for various clamp types

Nominal pipe size	Bolt diameter (in)	Bolt torque (ft-in)	Friction loads by clamp type				
			CL-1 (N)	CL-4 (N)	CL-8 (N)	SCL-8 (N)	CL-11 (N)
2	0.5	20	12,842	2,675	25,684	5,351	12,842
3	0.5	20	12,842	5,351	25,684	10,702	12,842
4	0.5	20	12,842	5,351	25,684	10,702	12,842
6	0.75	25	10,702	8,026	21,404	16,053	10,702
8	0.75	35	29,965	10,702	29,965	10,702	29,965
10	1	55	35,316	13,377	35,316	13,377	35,316
12	1	55	35,316	13,377	35,316	13,377	35,316

Along with the clamping forces, the stiffness of the clamp also has a significant effect on the piping flexibility analysis. Non-vibrating pipe systems typically assume rigid stiffness. However, this assumption can lead to significantly conservative designs with dynamic restraints and result in unnecessary modifications to the system. Assuming rigid stiffness may not always produce conservative approaches in a complex piping system where the supports are far from rigid [6]. Therefore, it is necessary to estimate the clamp stiffness for the piping stress analysis accurately. The true stiffness of a clamp will depend on the stiffness of the supporting structure as well as the clamp itself. A clamp that is located high above the ground may be less stiff than one that is located on the ground (Figure 11).

Figure 11. Example of various piping support structures with clamp-type restraints

Performing the analysis based on a more realistic stiffness will produce more reliable results. Using structural information, some piping analysis methods incorporate a partial structural model of the supporting structure in the piping system. However, it is often difficult to know the design of the structures during the piping stress analysis and incorporate the structural design into the flexibility analysis model. Analysts also need to consider time and cost, in terms of balancing detail and accuracy with the project schedule requirements (Figure 12).
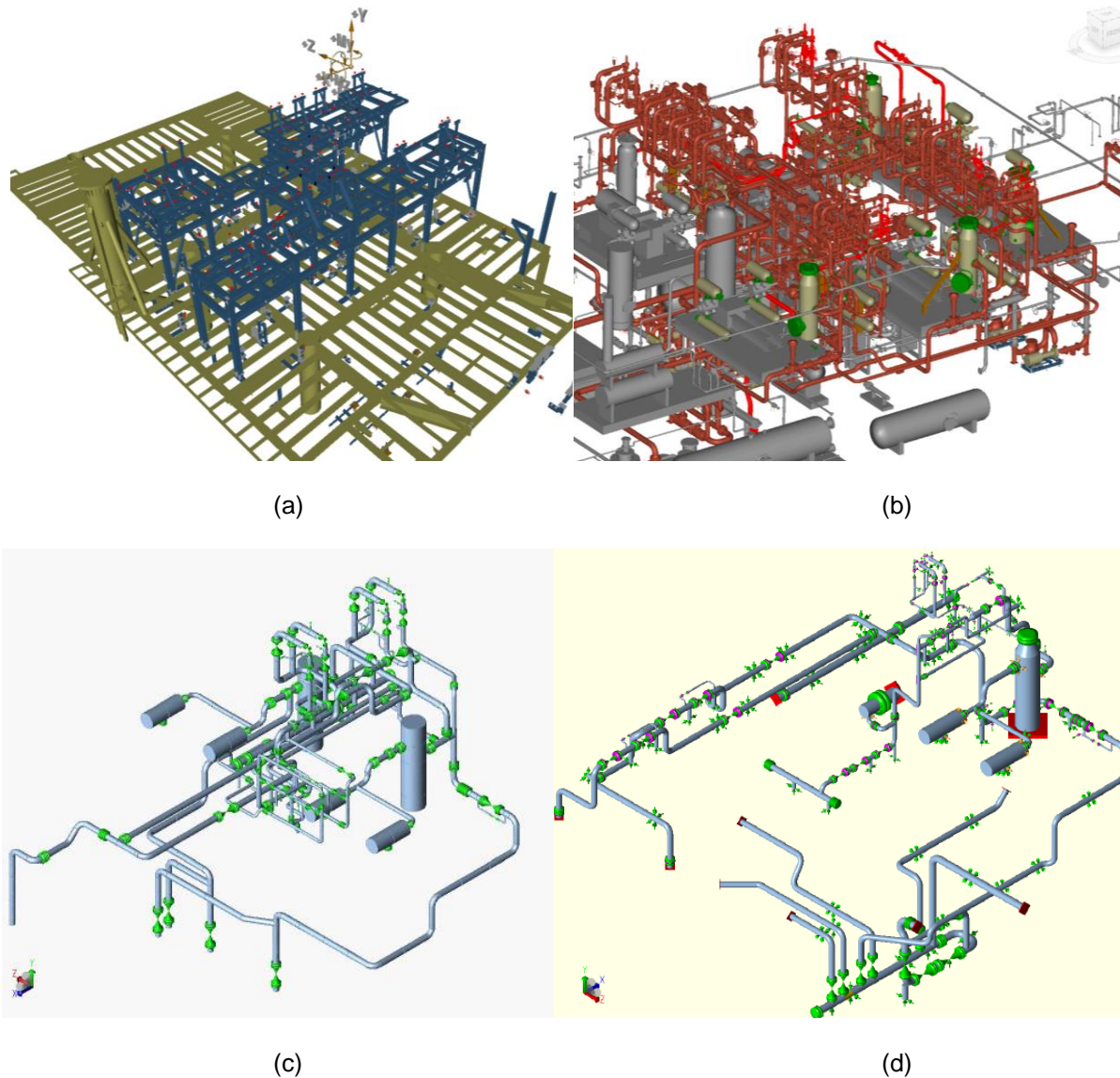
Figure 12. Example of piping flexibility analysis, (a) structure model, (b) piping model, (c) flexibility analysis model, (d) flexibility analysis model

Determining a reasonable value close to the true stiffness of a clamp support is an important factor in achieving a practical modeling accuracy. The stiffness of the clamp can be calculated manually. For example, the minimum required static stiffness of a pipe support based on API 618 5th edition can be used for the stiffness. The equation for minimum static support stiffness determined from API 618 5th edition is shown below.

$$minimum\ K_s = C_{ks} * A^{0.75} * I^{0.25} * f_{n,T}{}^{1.5}(n - \frac{1}{n})$$

Where

C_{ks} is the constant dependent on support stiffness units (SI units: 1/130; USC units:25)

A is the pipe cross-sectional metal area in mm²

I is the pipe cross-sectional area moment of inertia in mm⁴

$f_{n,T}$ is the minimum transverse natural frequency in Hz

n is the number of active supports ($n = 2$ as a minimum)

Analysts must review the calculated stiffness values to ensure that these values provide reasonable boundary conditions to the piping system without compromising the dynamic restraints and providing too much flexibility. As a rule of thumb, at least two times the calculated API 618 stiffness can be considered a good starting point in the piping flexibility analysis. Table 2 shows the pipe support stiffness using a maximum rated compressor speed of 1000 rpm.

NPS	Schedule	OD	ID	Minimum Static Stiffness (lbf/in)	Minimum Static Stiffness (N/cm)	Axial (2x API 618 minimum) (N/cm)
2	XXS	2.375	1.503	7,040	12,329	2.47E+04
3	160	3.5	2.624	13,933	24,401	4.88E+04
4	160	4.5	3.438	24,913	43,629	8.73E+04
6	120	6.625	5.501	49,668	86,981	1.74E+05
8	100	8.625	7.439	79,847	139,833	2.80E+05
10	120	10.75	9.064	155,561	272,430	5.45E+05
12	120	12.75	10.750	238,351	417,417	8.35E+05

Table 2. Example of restraint stiffness for the flexibility analysis

7. Support spacing

There are many tables available for determining the required spacing between supports. The formula used to determine maximum spans is generally used to limit sustained loads based on a uniformly loaded beam with both ends fixed or simply supported at both ends, and based on a several other, general criteria such as horizontal pipe runs, the existence of concentrated loads (ie, valves, flanges and any special items) and standard weight. In practical applications, the maximum allowable pipe deflection between supports should not exceed 1 inch (2.54 cm) or half the nominal pipe diameter; whichever is the smaller [3]. For example, ASME B31.3 established limits for sustained load stresses. Pressure and weight on any component in a piping system can cause stresses along the axis of the pipe. These stresses are not self-limiting, and therefore will not diminish over time as long as the loads are present. The limits are the hot allowable stresses of the material at a certain temperature. These limits do not consider vibration conditions (Figure 13).

Figure 13. Example of piping design without piping vibration consideration

When a piping system has serious vibration issues, a separate analysis should be undertaken to ensure that the mechanical natural frequency of the pipe segment is greater than a certain value. API 618 defines a limit for the mechanical natural frequency of reciprocating compressor piping systems. It states that the analysis should be performed to avoid mechanical resonance using span and basic vessel mechanical natural frequency calculations. This maximum allowable span considers the compressor operating speed and the support type. The natural frequency should be greater than 2.4 times compressor run speed, and the support must dynamically restrain the piping.

As an example calculation, we can consider a 10" NPS schedule standard seamless pipe (106 Gr. B) with the following conditions:

- Pressure: 300 psig
- Temperature: 300°F
- Corrosion allowance: 0.0625"
- Compressor operating speed: 1000 rpm

In addition, a heavy piping component (300 lb) is located 15 ft from the pipe support (assuming 30 ft between adjacent pipe supports). B31.3 gives the formula for calculating the hoop stress due to pressure as: $S_{hoop} = \frac{PD}{4t}$. In this equation, P = pressure, D = Diameter (10.75 in), and t = wall thickness (0.365-0.0625=0.3025 in). This calculation gives S_{hoop} as 2,665 psi. The bending stress caused by gravity can be calculated as: $S_{bend} = \frac{WL}{Z}$, where W = weight of the overhang, (400 lb in total, assuming that 100lb is pipe overhang weight), L = length from point of support to W(15 ft), and Z = section modulus of pipe (25.22 in³). These values give a value for S_{bend} of 2,855 psi. The total calculated sustain stress is 5,520 psi (total of two stresses), which is below the allowable stress of the material at the operating temperature. Therefore, the given 30 ft piping span is acceptable according to B31.3 criteria.

In addition, the pipe needs to meet the mechanical natural frequency guideline, which, using the formula (2.4*RPM/60), gives a minimum natural frequency of 40 Hz. Using API 618 Appendix P, the maximum allowable pipe span to meet this natural frequency is about 14 ft for a double-acting reciprocating compressor. In reality, additional supports are also required near elbows, flange sets, valves and other concentrated masses. Furthermore, if supports are not rigid, such as those used for elevated pipe supports, the user must consider reducing the spacing further.

8. Nozzle load (or load checking)

The first step in a piping stress analysis is to check if the sustained, occasional and expansion load cases pass the code-allowable stress or not. Next, the displacements and loads at the support locations must also be checked, in order to make sure that the results are reasonable. Large displacements may indicate that either data entry was incorrect or the piping is too flexible. High loads might indicate that the piping system is too stiff. Lastly, loads and displacement at the equipment nozzles need to be checked to make sure that nozzles are designed to withstand forces and moments from the thermal expansion or contraction of piping in service.

Equipment interacts with the other aspects of the plant, especially piping. The nozzle loads are the net forces and moments acting on the nozzles from the weight and thermal expansion of connecting piping and equipment. Increased nozzle loads can cause misalignment and can occur due to inadequate equipment and piping support. Piping stress engineers use FEA software to ensure that piping reactions at the nozzle are designed within the limiting criteria set by equipment vendors. The Welding Research Council (WRC) guidelines 107/297 provide methods for calculating nozzle loads for pressure vessels such as pulsation dampeners or scrubbers. Those methods are relatively easy to use. However, they produce very low allowable forces and moments; much lower than the actual maximum allowable nozzle loads. This may be acceptable when space is not a critical design factor (ie, off-skid piping vs on-skid piping). Similar to setting design conditions, WRC 107/297 is commonly considered the most conservative method with the highest safety factor. However, economics should be considered when choosing the method to calculate the maximum allowable nozzle loads and it is often possible to produce a safe but more economical design by applying more realistic values.

Therefore, since low allowable nozzle loads increase piping system costs because of the complex layouts and supports required to decrease the nozzle loads, vendors should provide accurate nozzle loads based on either experimental test data or detailed finite element analysis (Figure 14).
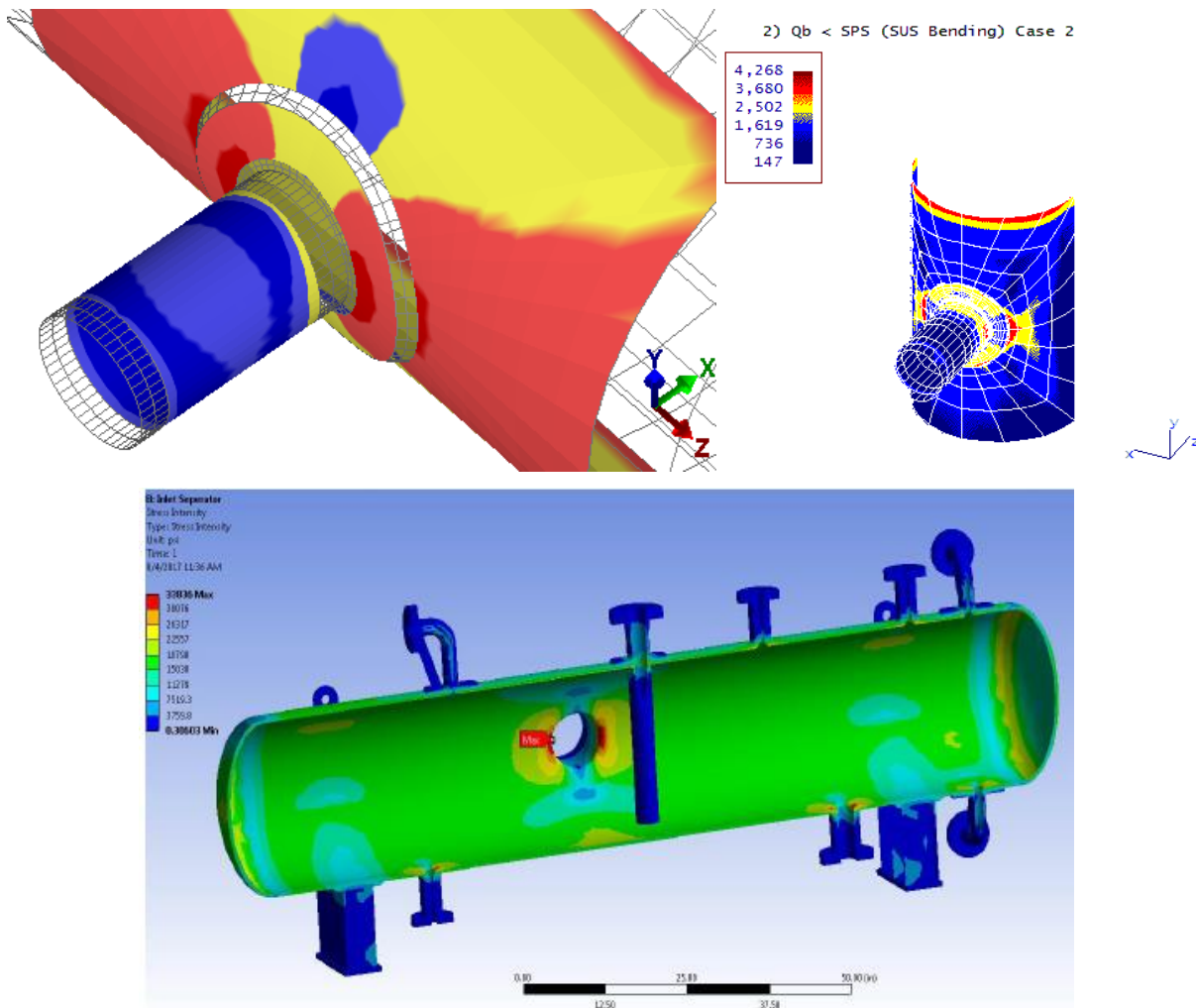

Figure 14. Nozzle load checking using finite element analysis

Engineers require piping layout, operating conditions (pressure and temperature) and nozzle movements for equipment zero point (thermal growth) to provide accurate results. Complicated equipment like centrifugal compressors, heaters and reactor requires more information from vendors, such as growth calculations, skid temperatures and the piping layout within the equipment. Reciprocating compressors require an acoustical analysis to reduce pulsation and might also require pulsation suppression devices between the compressor and piping prior to the piping stress analysis. The inlet nozzle of the suction pulsation bottle and outlet nozzle of the discharge pulsation bottle are connected to the piping. As long as the piping layout meets the requirement of the pulsation analysis, the procedure for calculating nozzle loads is not much different than for a typical piping stress analysis. Engineers should be particularly cautious to use typical methods to reduce nozzle loads such as adding springs or increasing piping flexibility by adding loops or expansion joints; because they cannot restrain the piping dynamically as mentioned earlier. Therefore, an engineering analysis should include different types of clamps with specialized functions in order to achieve dynamic stability as well as static flexibility in their

piping system. Using clamps with sliding plates or gaps controlling the direction of clamp movement are the preferred methods to control the nozzle loads in a vibrating piping system (Figure 15).

Figure 15. Various type of clamp-type restraints

Questions often arise as to whether the cylinder nozzle loads should be checked during the piping stress analysis since they are not connected to the piping directly. If piping stress engineers are tasked to check cylinder nozzle loads due to thermal expansion in the piping, they require more information from the equipment vendors, such as thermal growths for both the cylinder and the pulsation suppression devices. When two or more cylinders are connected to the same pulsation suppression devices, this information is more critical, and thermal growth of the piping has little effect on the cylinder nozzle loads. Forces induced by thermal expansion of the pulsation suppression devices are the major factor contributing to cylinder nozzle loads. In addition, regardless of the piping thermal growth, the vendor must check the thermal expansion of the pulsation suppression devices with compressor cylinder nozzle in order to avoid intolerable misalignment and excessive stresses during operation (API 618 7.9.5.2.2).

There is another type of nozzle loads, referred to as tie-in or interface nozzle loads. Due to the multidisciplinary nature of many industry projects, the boundary conditions at piping interfaces should generally be consistent between multiple parties. The agreement could be forces and moments, displacements or both. For example, a typical method of specifying a boundary condition is to use an anchor, which is zero displacement, or to meet given force and moment limits at the tie-in point. This design basis for yard or package piping generates artificially high stress in the package piping or produces unnecessary thermal flexibilities in the yard piping. Sometimes, it is impossible to lower the stress level in the package piping below the limit (Figure 16).
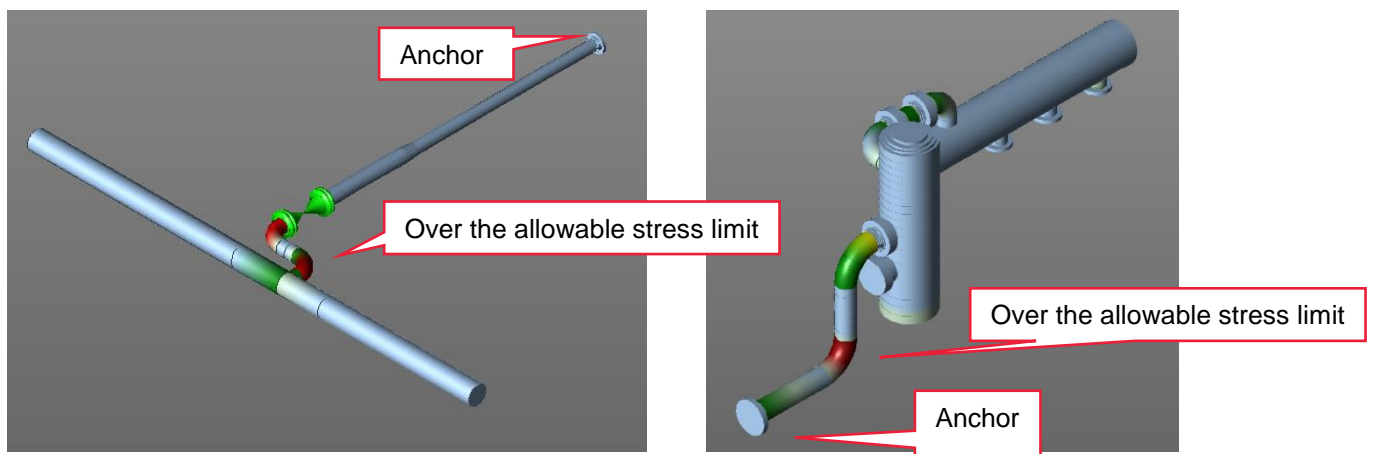

Figure 16. Zero displacement agreement at tie-in location, stress results in off-skid (left), stress results in on-skid (right)

The best approach for ensuring package piping integrity due to piping flexibility is to model the yard or off-skid piping up to a known boundary condition. The effort to add this extra piping is relatively small given the benefit in accuracy and

reliability of the results. However, the final design for yard piping is usually not completed until after the compressor package is finalized. It may be to the benefit of the equipment owner to specify that the flexibility analysis for the compressor package and yard piping be conducted by one party (Figure 17). A detailed discussion of piping flexibility analyses for package and yard piping can be found that in Eberle, *A Recommended Approach to Piping Flexibility Studies to Avoid Compressor System Integrity Risk* [7].
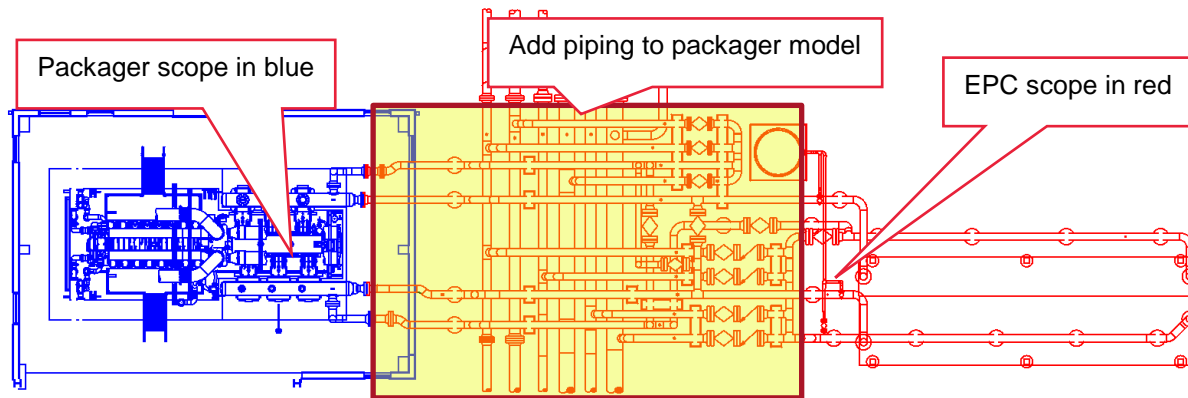

Figure 17. Example of scope break in compressor station

8. Summary

The typical approach of conducting a piping stress (flexibility) analysis does not provide reliable results and could lead to excessive vibration in the piping system. A piping stress analysis for vibration applications requires several special considerations. This paper discussed some of the major components to be considered during a detailed design analysis and scope setup:

- Code: use stress range factor as per plant start-up and shutdown, B31.3
- Vibration analysis: perform a pulsation and dynamic stress analysis along with the flexibility analysis
- Temperature: the maximum operating temperature should coincide with the pressure
- Pressure: the maximum operating pressure should coincide with the temperature.
- Support type: use vibration clamps; clamp-type, shoe and spring supports are unsuitable for vibrating piping
- Support modeling technique: various types of clamps and modeling techniques allow pipe slip in axial direction
- Engineers: must have a good working knowledge of piping dynamics and piping static analysis
- Support stiffness: use a reasonable stiffness, close to true stiffness
- Nozzle loads: use an accurate (more sophisticated) method to calculate the maximum allowable nozzle loads
- Support spacing: set the space to avoid resonance in piping system, not for the sustained load case
- Tie-in agreement: the flexibility analysis for the compressor package and yard piping should be conducted by the same party

9. References

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