Resolving a Piping Vibration Problem by Installing Viscous Damper Supports

Carlos Herrera Sierralta, Husain M. Muslim, Meshal T. Alsaiari, Daniel Fischer

***Abstract*—**The vast majority of piping vibration problems in the Oil & Gas industry are provoked by the process flow characteristics which are basically related to the fluid properties, the type of service and its different operational scenarios. In general, the corrective actions recommended for flow induced vibration in piping systems can be grouped in two major areas: those which affect the excitation mechanisms typically associated to process variables, and those which affect the response mechanism of the pipework per se. Where possible the first option is to try to solve the flow induced problem from the excitation mechanism perspective. However, in producing facilities the approach of changing process parameters might not always be convenient as it could lead to reduction of production rates or it may require the shutdown of the system. That impediment might lead to a second option, which is to modify the response of the piping system to excitation generated by the process flow. In principle, the action of shifting the natural frequency of the system well above the frequency inherent to the process always favours the elimination, or considerably reduces the level of vibration experienced by the piping system. Tightening up the clearances at the supports (ideally zero gap) and adding new static supports at the system, are typical ways of increasing the natural frequency of the piping system. However, only stiffening the piping system may not be sufficient to resolve the vibration problem, and in some cases, it might not be feasible to implement it at all, as the available piping layout could create limitations on adding supports due to thermal expansion/contraction requirements. In these cases, utilization of viscous damper supports could be recommended as these devices can allow relatively large quasi-static movement of piping while providing sufficient capabilities of dissipating the vibration. Therefore, when correctly selected and installed, viscous damper supports can provide a significant effect on the response of the piping system over a wide range of frequencies. Viscous dampers cannot be used to support sustained, static loads. This paper shows over a real case example, a methodology which allows to determine the selection of the viscous damper supports via a dynamic analysis model. By implementing this methodology, it is possible to resolve piping vibration problems by adding new viscous dampers supports to the system. The methodology applied on this paper can be used to resolve similar vibration issues.

***Keywords*—**Dynamic analysis, flow induced vibration, piping supports, turbulent flow, slug flow, viscous damper.

I. INTRODUCTION

THE importance of the piping in the industry is very remarkable as the majority of the production and refining processes involve the utilization of large quantities of pipework. Piping failures accounts for over 20% of all hydrocarbon releases [1], some of them with a potential significant impact to disrupt production, personnel safety, damage of assets/ facilities, environment, and surrounding communities.

Turbulence exists in most piping systems. In straight pipes it is generated by the turbulent boundary layer at the internal pipe wall, the severity of the turbulence phenomenon depends upon the flow regime as defined by the Reynolds number. However, for most cases the principal sources of turbulence are major discontinuities in the piping system. Typical examples of discontinuities are bends, tees, reducers, partially closed valves, or control valves. This phenomenon occurs particularly at high flow velocities and large pressure drops in short periods of time.

Turbulent flow may turn into potential high levels of broadband kinetic energy observed locally or in the surroundings of the turbulent source. For instance, higher levels of kinetic energy can be present at the immediate upstream of the turbulent flow phenomenon, and particularly at the downstream piping sections connected to the control valve.

Despite the kinetic energy is distributed across a wide frequency range, the main part of the excitation is typically concentrated at the low frequency range. The lower the frequency the higher the level of excitation that potentially can match with the lower modes of natural frequencies of the pipework, which could provoke visible vibration of the pipe and in some cases the motion of piping supports and its associated steel structure.

The fluid kinetic energy can be calculated as a product of the actual fluid density (ρ) multiplied by the fluid velocity squared (v^2), [1].

$$\text{fluid kinetic energy} = \rho \cdot v^2 \tag{1}$$

The fluid kinetic energy value can be compared with the criteria provided by Energy Institute, *Guidelines for the Avoidance of Vibration Induced Fatigue in Process Pipework* [1] which provides a simple criteria in order to categorize the Likelihood of Fatigue Failure (LOF) of a piping system due to vibration as one of the following types: *Low*, it is low LOF when the calculated fluid kinetic energy is below 5,000 kg/m.s^2, *Medium* (between 5,000 to 20,000 kg/m.s^2), or *High* (> 20,000 kg/m.s^2) [1].

Another common dynamic excitation mechanism is the generated by certain two-phase flow regimes, also known as

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slugging. A two-phase pipeline or piping system contains liquids and non-condensable gases in a common stream. There could be more than one liquid (such as oil and water) and more than one gas (such as a mixture of natural gas and air). In slug flow, the liquid waves are sufficiently high to entrap pockets of gas, forming alternating pockets of gas and liquid. This effect is called *slugging* which causes large pulses of pressure and can induce severe vibration [2].

When a liquid slug passes through and elbow, dynamic forces will be imparted on it. The generated axial impact force at a 90-degree elbow is estimated by [3].

$$\boldsymbol{Fx = Fy = \rho \cdot A \cdot v^2} \tag{2}$$

The reaction exerted by a slug force as it discharges past an elbow provokes an excitation at the piping system.

In addition to the magnitude of the force, the sequence in which the slug force impacts at each elbow of the system also influences the way how ultimately the piping vibration manifests in the system. The authors' experience in oil and gas lines shows that slug flow induced vibration, when occurs, it is observed at low or very low frequencies typically ≤ 10 Hz.

In the *slugging* phenomenon, the slug forces are generated and are acting at the changes of direction of the piping system due to change of momentum of the fluid, which can occur under certain two-phase flow patterns. The order of magnitude of such slug forces depends not only on the process stream properties such as fluid density and velocity, but also on the ratio between the fluid phases involved: liquid, and gas, converging into the same pipework' geometry.

Additionally, the effects of slug flow in a system are basically a function of time. Most of the Computerized Fluid Dynamics (CFD) programs can characterize the timing of the slug flow and calculate the sequence of the slug forces hitting the elbows in a given piping system. In the absence of sophisticated (CFD) programs, the so-called Time-History which is featured on widely used piping flexibility analysis software, such as: CAESAR II®, AUTOPIPE®, ROHR2® can be utilized in order to take into consideration the time dependent characteristics of the slug flow acting on a particular piping system, and be able to conduct a satisfactory dynamic analysis [4].

Note that it is not part of the scope of the current paper to elaborate the static equivalent method which is the default method used to comply with the sustained load cases as per the applicable design code under jurisdiction or proponent, in this particular case ASME B31.3 [6]. This paper will not elaborate the Time-History method which was detailed in the recent technical paper ASME IMECE 68915 [4]. Rather, the current paper will focus on a methodology for selecting viscous damper supports via a time-history dynamic analysis model.

In general, viscoelastic dampers reduce piping vibrations by converting kinetic energy into heat in their casing which is filled by a highly viscous medium, thus reducing the amplitude of the oscillation of the piping system. For papers on modelling and workflow please see [8] and [9]. A piston connected to the upper connection plate can move freely (within limited tolerances) in all directions within the viscous medium. In general, either the upper or lower plate of the viscous damper (Fig. 1) can be connected to the piping system, while the other plate is mounted on a fixed abutment (e.g.: steel structure member or civil foundation), but note that ultimately the viscous damper manufacturer's recommendations shall govern on how the device will be installed in the field.

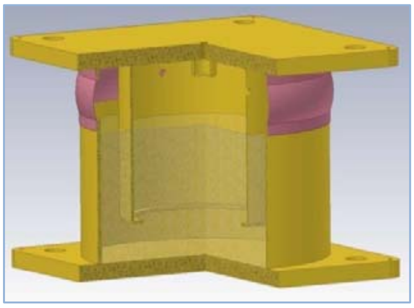

Fig. 1 Cross section through a viscous damper

It should be noted that viscous dampers are different from so-called snubbers, which are typically utilized to ensure that piping systems are protected in case of shock forces. Snubbers are typically designed to contract and expand during normal operating conditions, but at the same time they lockup under the action of a shock force, with a relatively instantaneous reaction time, in order to protect the piping system from the abrupt impact of occasional loads such as; seismic, water-hammer, as applicable. The viscous damper has a smoother reaction to the shock load, finally creating similar retention forces but with much less stress in the attached piping system.

A suitable mathematical model used to represent viscous damper behaviour is the well-known Maxwell model which consists of an ideal spring element and a dashpot (piston in a viscous medium) connected in series. This model is especially well suited for the description of viscous fluid dampers as it shows typical ideal relaxation behaviour.

The frequency dependence to the damping characteristics of any particular design of viscous damper is determined by tests performed by the viscous damper specialized manufacturer. The vibration of the piping system provokes the piston to shear which displaces the fluid of the viscous damper creating the damping effect.

For freely vibrating systems, the resulting frequency of oscillation is known as the natural frequency of the system (*f*), and it is affected by the stiffness of the structure (*k*) and the mass of the system (*m*).

The natural frequency of an undamped vibration system with one degree of freedom can be determined with [1]:

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{3}$$

and the damping ratio (***ϑ***) [7], where (c) is the damping resistance:

$$\vartheta = \frac{c}{2\sqrt{k.m}} \tag{4}$$

By substituting the stiffness, k, as obtained from (3), into (4) the following equation can be obtained:

$$c = 4\pi \cdot f \cdot m \cdot \vartheta \tag{5}$$

This paper shows the methodology based in (5), for the selection of viscous dampers to resolve a real case of danger piping vibration.

The paper is structured as follows: Section II explains the specific case of the study, provides a brief description from operational/process perspective and describes the vibration issue observed at field, and shows the relevant data of vibration readings that confirm the "Concern/Danger level" of vibration experienced in the line in question. Section III shows the methodology applied in this case of study, and the proposed steps that were conducted in order to resolve the vibration issue in this piping system. Section IV provides the detailed calculation of the damping resistance required to control the piping vibration. Section V provides further details of the evaluation/selection of the vicious damper support and its dynamic evaluation assisted by a specialized software package. Section VI described the input and output results of the dynamic evaluation before and after installing the selected viscous dampers supports.

It is worth to mention that despite the analysis described in this paper was conducted utilizing CAESAR II® and ROHR2® software packages, this does not constitute any commercial preference nor business relation among the co-authors, their employers and the referred software provider companies. The sole intention of the paper is the knowledge sharing across the industry of the described methodology, as it could be applied to solve a considerable range of vibration problems.

II. Case of Study

This paper presents a vibration issue observed in a crude oil pipe line, in an on-shore facility. The prevalent diameter of the line in question is 610 mm (24 in.) diameter, with a small section of 914 mm (36 in.) diameter.

The line is one out of three lines which control the crude oil flowrate from one production facility to another located several kilometres downstream. The material of construction of the line is carbon steel meeting specification API 5L Grade B, with wall thicknesses of 19.05 mm (0.750 in.) and 17.48 mm (0.688 in.), respectively, and flanges Class 300 as per ASME B16.5/ASME B16.47 Series A as applicable. Rest of the mechanical design conditions are as follows: Design Temperature 77 °C (170 °F), Design Pressure 23.37 Barg (339 psig), and M.D.M.T 2.22 °C (36 °F).

From process perspective the line is categorized as *Two-Phase* flow in the Piping and Instrumentation Diagram (P&ID), this leads to confirm the presence of *slugging* in the field (slug flow forces).

Due to operations flexibility, the line flowrate is needed to be increased approximately 5% above the original normal operating flow of the line. However, experience at field shows that every time the flowrate of this line is increased above the original operating then the vibration levels shifted all of the sudden from acceptable levels to danger levels. The vibration is noticed in particular at the change of directions of the line, and also at the control valve's immediate upstream and downstream sections including the control valve itself.

Based on (1) the calculated likelihood of failure of this piping system is categorized as *low* according to the Energy Institute [1], therefore the *kinetic energy* is not considered a concern in this case. However, due to the large diameter of this line and the fact of being under *Two-Phase* flow regime the pipe is subjected to a considerable *slugging* force, 4,255 kg/m.s² as calculated per (2).

The effect of the slug force is a time dependent phenomenon as it is impacting the elbows sequentially, for that reason designers should notice that a time-history dynamic analysis is essential while designing or evaluating slug flow cases.

In order to illustrate the composition of the system, Fig. 2 shows the layout of the line in question, the location of the control valve, the direction of flow of the process and the location where piping vibrations readings were taken.

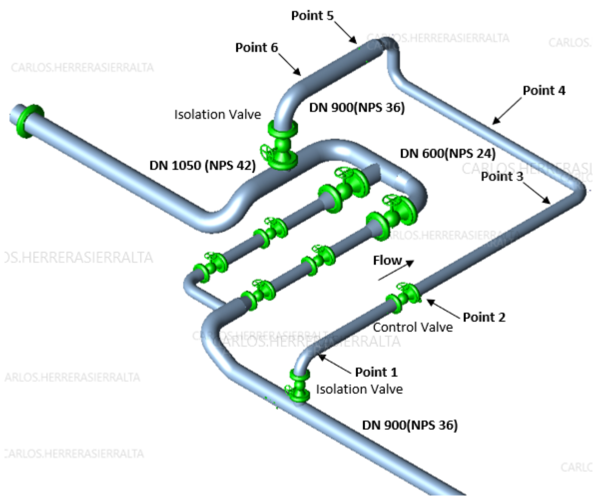

Fig. 2 General view of the piping system under study

Table I shows the set of vibration field measurements that were taken at six different locations of the line under study.

TABLE I
FIELD VIBRATION MEASUREMENTS

Location	Frequency (Hz)	Amplitude (mils pk-pk)
Point 1	*12.50*	5.25
Point 2	*3.50*	15.00
Point 3	*3.50*	70.94
Point 4	*7.30*	6.40
Point 5	*3.50*	54.09
Point 6	*4.50*	34.21

The collected data indicate that three out of six of the measurement vibration points of the line fall into the range of the so-called *Correction* zone and the boundary of *Danger* zone area as per the applicable criteria chart [5], see Fig. 3.

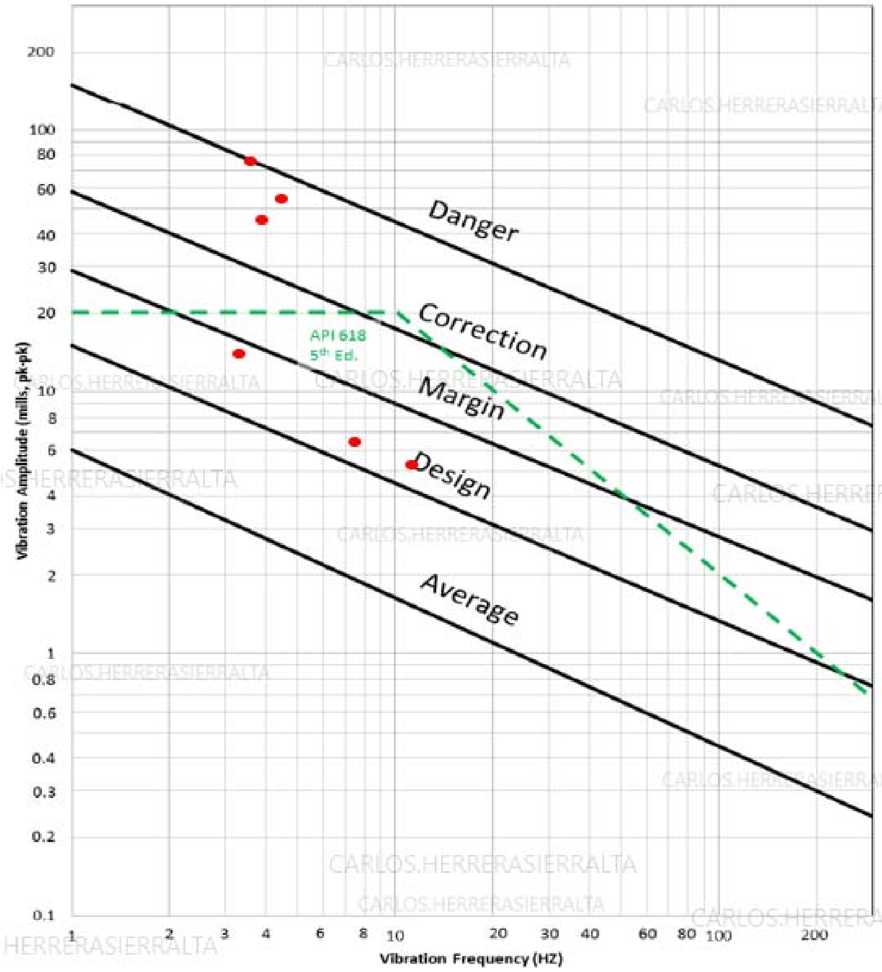

Fig. 3 Vibration measurements plotted on chart [5]

III. Methodology

The methodology implemented in this paper, which led to the validation of the dynamic model based on the real vibration operational scenario and the selection of the appropriated viscous damper supports, is described in the following steps:

Step1. Gather the required process data and perform slug force calculations as per (2) and input the resulting forces into the stress model of the piping system in its original condition, existing condition at field, and conduct static analysis to meet design code requirements, in this case ASME B31.3 design requirements [4]. It is not the objective of this paper to address this step as this is a very well-known practice conducted by piping designers.

Step2. Conduct Time-History dynamic analysis in the stress model of the piping system's original condition in order to determine the slug length that provokes the vibration amplitudes as measured in the field, conduct iterations as necessary up to obtaining the simulation of the existing field conditions. Refer to ASME IMECE 68915 [4], for a comprehensive detailed procedure of this step.

From the Time-History, two major parameters will be observed: the natural frequency at the location where the viscous damper will be installed, and the thermal expansion/ contraction displacement range amount at that particular location.

Step3. Determine the required amount of damping resistance as per (5).

Step4. Conduct the selection of the required viscous damper based on the calculated damping resistance from *step 3*.

Step5. Include the selected viscous damper as per the calculated damping resistance into the stress model.

Step6. Run Time-History dynamic analysis stress model and perform adjustments at the piping supports as necessary in order to balance and consolidate the static, dynamic results, and if possible, increase as much as possible the first mode of natural frequency of the piping system.

The objective is not only to provide the sufficient damping resistance to the system but also, if possible to be able to shift the fundamental frequency of the piping system to, ideally, the so-called *Medium Stiff* to the *Stiff* range as defined by [1], and at the same time be able to comply with the applicable design Code in terms of loads, moments, stresses and displacements.

IV. Damping Resistance Calculation

Based on the vibration data and the risk associated to the potential leak or control valve malfunctioning of the line in question it was determined in first place to install the viscous damper supports as near as possible to the existing control valve in order to mitigate the potential risk of having a fatigue cracking at the small-bore connections, which are located upstream and downstream to the control valve.

Secondly, it was taken into consideration the fact that the viscous damper supports near to the control valve will be able to mitigate the vibration at the bonnet of the control valve. Also, the installation of the viscous damper near to the control valve, will potentially be able to prevent any malfunctioning of the valve such as the rupture of instrumentation solenoids, or rupture of air tubing. These modes of failures are very common on control valve exposed to significant vibration levels.

Knowing the location of viscous damper allows to check the first mode of natural frequency at that location which is required for the *damping resistance* calculation as per (5). It also allows to check the maximum calculated displacements that are occurring at that location for the different load cases, particularly for the thermal expansion and contraction load cases. Lastly, it allows to check the load combinations for both maximum design and the range of operating cases, which is required for checking the allowable displacements limits of the viscous damper model.

Eigenvalue analysis of the structure shows a mode at roughly 3.4 Hz with big deflections very close to the valve location and a mass participation factor of around 10%, Fig. 4. This is confirmed by Time-History analysis. For this analysis the piping structure is uniformly excited with a frequency limited noise signal that acts on the entire relevant piping section. The level of the excitation is scaled to roughly match the measurement data. The pipe response is calculated with a direct-integration method, the result for the location 2 – close to the valve – can be seen in Fig. 5.

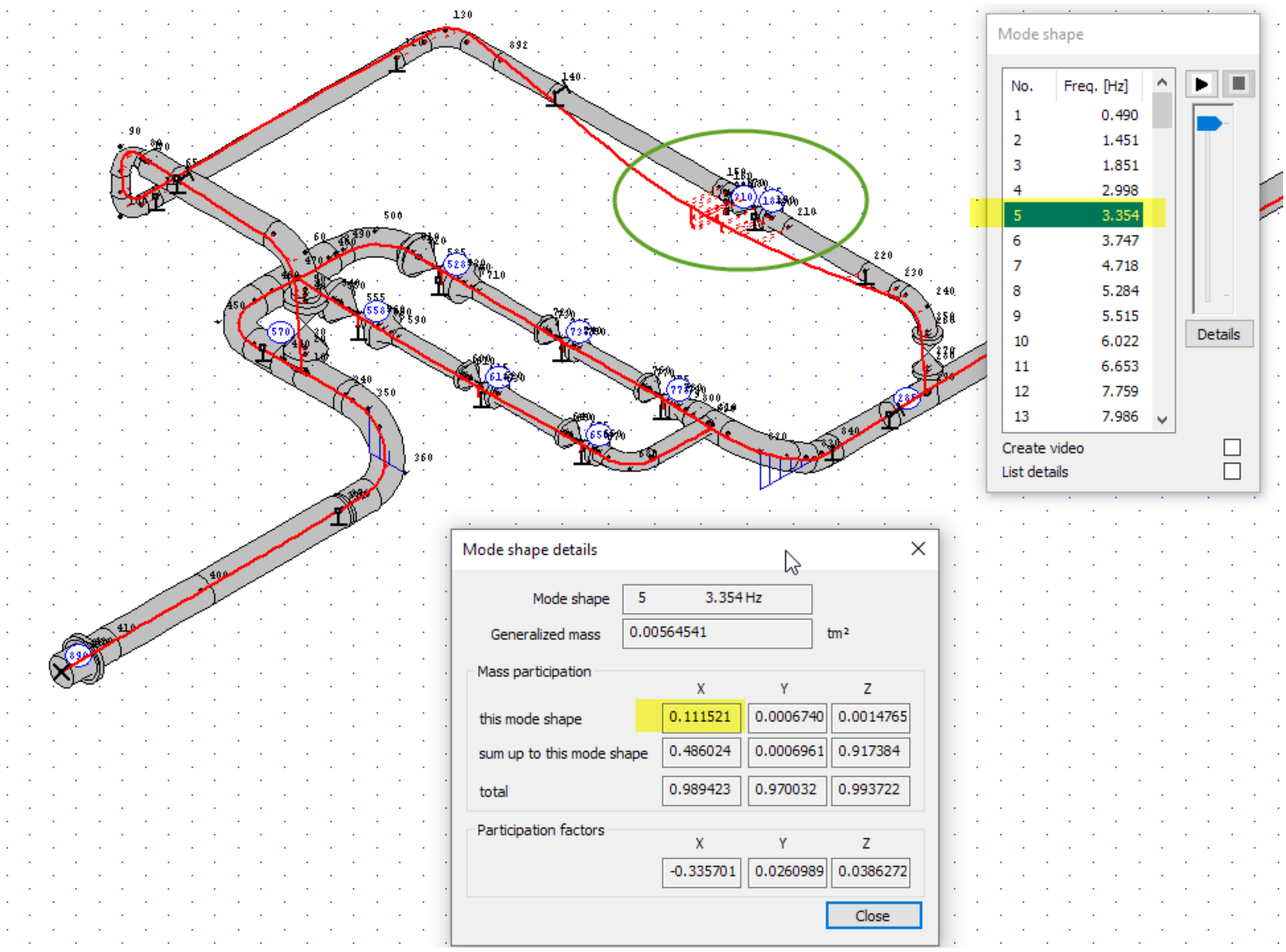

Fig. 4 Modal analysis of the piping structure

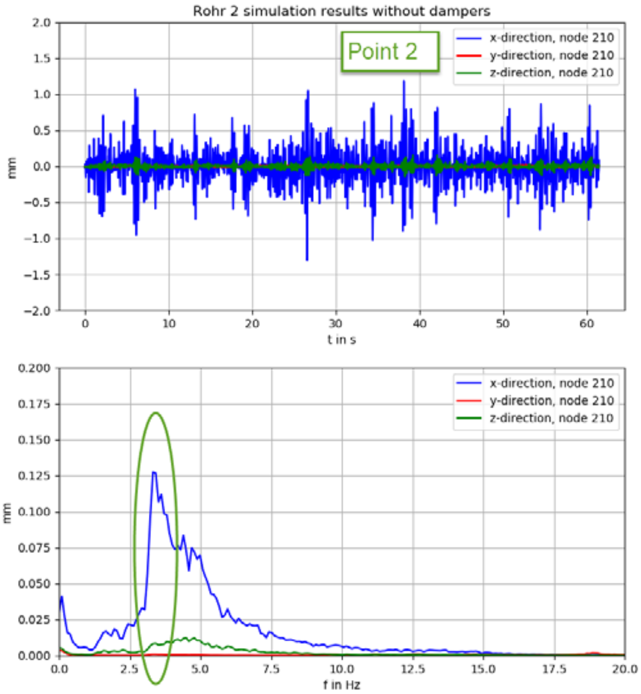

Fig. 5 Time-History and Amplitude Spectrum of the pipe response

To summarize the input data for the preliminary damper selection:

- Dominant frequency is $f = 3.5$ Hz.
- *Mode shape mass participation* factor is 0.1.
- Dominant direction is in the horizontal plane, transversal to the piping.
- Resulting modal mass of: GMAS · (*mode shape mass participation*) = 178,000 kg · (0.1) = 17,800 kg.
- The recommended *damping ratio* (ϑ) value for continuous vibration situations, such as the occurring in this case, is $\vartheta = 0.4$.

Subsequently, the necessary amount of *damping resistance* (c) that is required to be installed in the proposed location as per (5) is as follows:

$$Damping\ resistance\ (c) = 4\pi \cdot f \cdot m \cdot \vartheta = 12.57 \cdot (3.50\text{Hz}) \cdot (17{,}800\ \text{Kg}) \cdot (0.4) = 313\ \text{kNs/m}.$$

V. Viscous Damper Selection

The following boundary conditions should be considered for the viscous damper supports selection in this scenario:

- Damping resistance.
- Obtaining maximum allowable displacements (usually thermal expansion) in all the directions (x, y, z) at the location where the viscous damper supports will be installed. This is obtained via static/dynamic analysis simulation output results.
- Operating and maximum design temperatures.
- Installation temperature and minimum ambient temperature.

The calculated *damping resistance* as per (5) as obtained in Section IV is 313 kNs/m.

Maximum thermal and quasi-static displacements for all the directions (*x, y, z*) in the proposed location for the line's load cases are shown in Table II. It shows that the maximum displacement is 1.76 mm.

TABLE II
Displacement Results at the Viscous Damper Proposed Location

DX, mm	DY, mm	DZ, mm
-1.29	0.39	1.76
0.08	0.39	0.13
-0.11	0.37	0.01
-0.11	0.37	0.01
-0.12	0.02	1.76
0.19	0.02	0.14
-1.37	0.00	1.62

For the selection it was concluded that a total of two viscous damper units type *VD-630/426-15-TU* are capable to provide the required damping resistance previously obtained as per (5).

It was also observed that the selected viscous damper units can absorb a maximum allowable vertical (y) displacement of ± 74 mm and a total maximum allowable horizontal displacement (x, z) of ± 72 mm, which perfectly cover the ranges of maximum displacements as calculated listed on Table II.

The viscous damper units were selected to be installed as shown in Fig. 6 (so called tandem-arrangement).

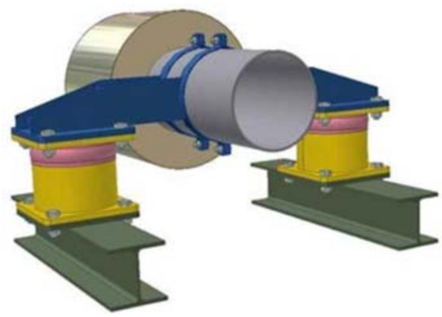

Fig. 6 General view of the selected viscous damper assembly

Damping resistance (kNs/m) versus frequency (Hz) curves for the selected viscous damper element, for both vertical and horizontal direction are shown on Fig. 7.

The calculated damping resistance as per (5) and the natural frequency obtained from the dynamic analysis simulation are confirmed to be within the capacity of the selected viscous damper element.

VI. Damping Simulation

With the viscous dampers and suitable installation locations selected Time-History analysis is repeated to calculate the expectable vibration reduction.

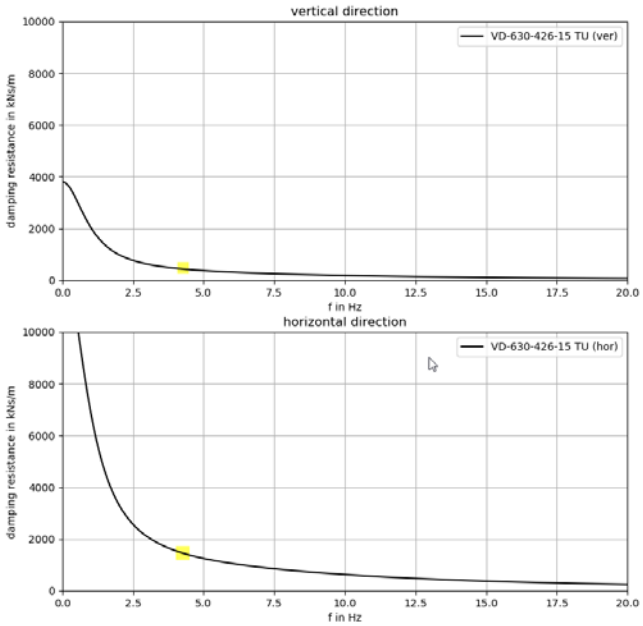

Fig. 7 Damping resistance (kNs/m) versus frequency (Hz) for the selected viscous damper element in vertical and horizontal direction

The software package utilized for the simulation was ROHR2®, which is provided with the necessary calculation feature for viscous damper. It should be noted that not all the commercialized software packages are provided with a dedicated viscous damper calculation option. In this regard, designers are advised to not use the *snubbers* option of the software as they are only designed either to react like an elastic element or lockup as a rigid stopper, rather than to provide vibration dampening resistance.

The viscous damper characteristics (i.e.: damping resistance) are already built into the software package algorithm therefore it can be directly selected from the data base listed in the viscous damper feature.

As previously discussed, the viscous damper supports were located at the control valve's upstream piping section (at 700 mm from the control valve flange end) as shown in Fig. 8.

Note that the location of the viscous dampers is at node 210 of this model, and are according to the general assembly configuration which is constituted by the couple of elements as shown in Fig. 2.

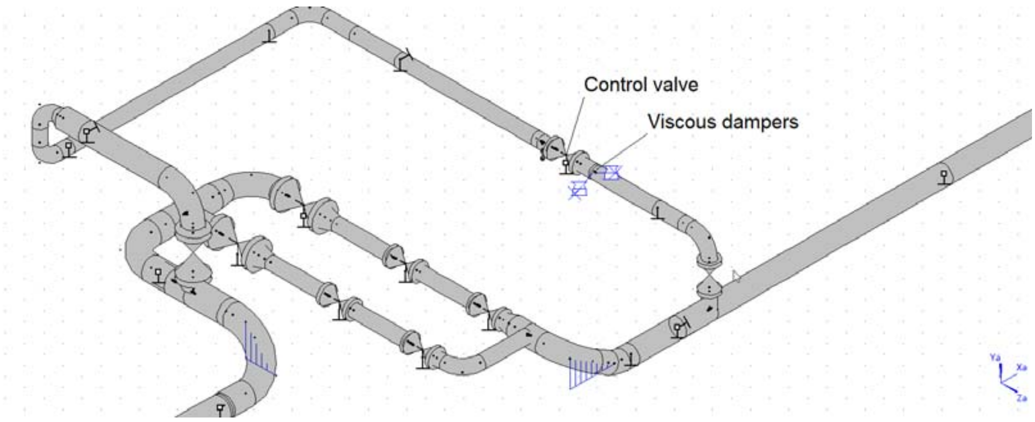

Fig. 8 General view of the selected viscous damper supports located at 700mm upstream of the control valve

The Time-History dynamic analysis model of the original condition, without viscous dampers, of the piping system in question exhibits a high level of vibration as is shown on Fig. 9. Fig. 10 shows the Time-History results from the analysis at the same location but with damper installed close to the valve. Similarly, vibration data in terms of displacement versus frequency are obtained for both, the original condition's model in Fig. 11, and the upgraded system by installation of viscous dampers in Fig. 12 with a significant reduction of the displacement levels in the low frequency transversal horizontal plane of the pipe (x direction) and a considerable reduction on the y, z directions.

The amplitude vibration results shown in Figs. 9 and 10 demonstrate a significant improvement of the vibration issue, passing from approx. 2.3 mm before the installation to 0.2 mm after viscous damper installation. Similarly, and as shown in Figs. 11 and 12, the vibration frequency is dramatically reduced from approx. 3.4 Hz to almost zero, so there is practically no resonance left, which clearly demonstrated that the addition of viscous dampers has solved the vibration issue.

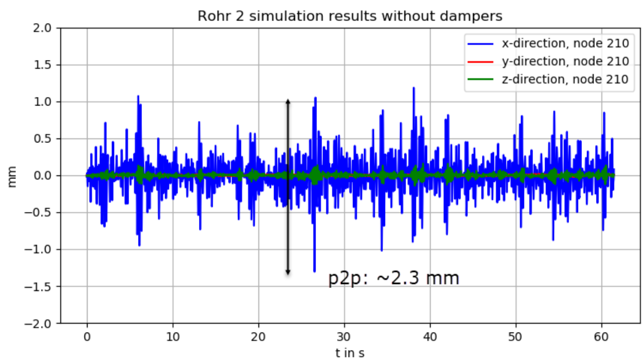

Fig. 9 Displacement (mm) versus time (s) in the original condition mode (without viscous dampers)

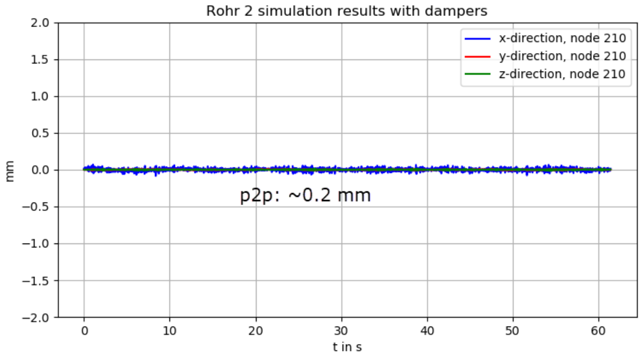

Fig. 10 Displacement (mm) versus time (s) in the modified condition (with viscous dampers)

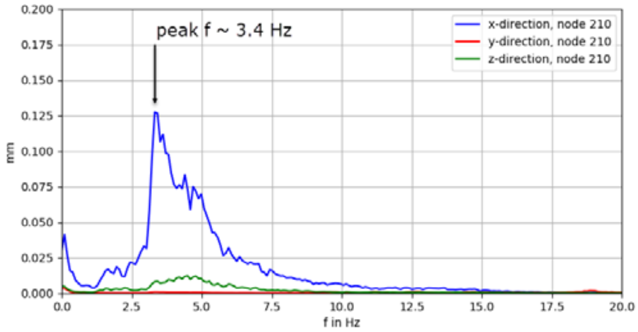

Fig. 11 Displacement (mm) versus frequency (Hz) without viscous dampers

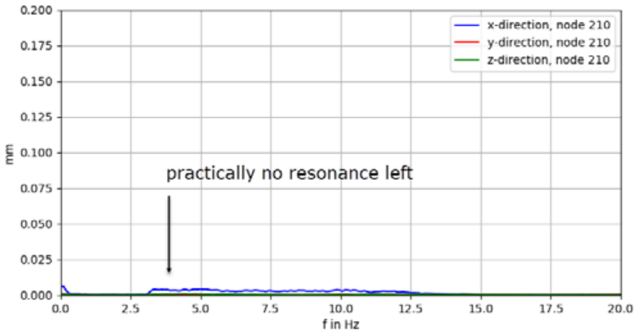

Fig. 12 Displacement (mm) versus frequency (Hz) with viscous dampers

VII. Conclusion

The development and implementation of this case study has demonstrated with a real example that the application of viscous damper supports could lead to the satisfactory resolution of piping vibration issues.

As contrary to normal static supports, the major advantage of using viscous damper supports is the capability of the dissipation of the piping movements without transferring the vibration issue to the steel structure. This is a typical problem when using normal static supports. In most cases this exacerbates the vibration problem risk as it transferred to other pieces of equipment such as process instrumentation, auxiliary structures, small bore piping, etc.

It is also worth to highlight the fact of the simplicity of the viscous damper installation at field as they can be linked to the pipes via bolted clamps which make them suitable for installation in live plants, without necessity of shutting down the production, as it was the case in this study, allowing to keep the oil crude flowrate at normal operating capacity.

We believe that the inclusion of the viscous damper algorithm feature in others vastly commercialized software packages might bring a most widely adoption of these devices across the onshore/offshore oil and gas sector, and across the industry in general.

Acknowledgment

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