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A SIMPLE ENGINEERING METHOD FOR SIMULATING FLOW-INDUCED VIBRATION IN PIPING

Peter Vasilyev[1]

[1] Principal, CKTI-Vibroseism, St. Petersburg, Russia (pvasilyev@cvs.spb.su)

ABSTRACT

Vibrations occur in nuclear power plant (NPP) piping while at work. Most often, piping vibrations are mild or moderate. However, sometimes the vibrations can be high, which may lead to fatigue problems and shorten the piping life.

There are numerous possible causes of piping vibration: rotating machines, compressors, and pumps; vortex shedding in the area of valves, tees, throttles, etc.; acoustic resonance; water hammer; external vibration transmitted through the supports; two-phase media. Among all possible causes, a class of causes can be distinguished based on the mechanism of vibration transmission to a pipe; that is flow-induced vibration. This vibration excitation mechanism is the most frequent in practice.

When the pipeline vibration raises concern, there is a need for numerical modelling and analysis of the piping vibration state. This can be useful to evaluate fatigue and predict service life. Moreover, if vibration reduction methods are required, the options for such methods must be analysed with the use of numerical models.

The problem of modelling dynamic processes in a pipeline medium (liquid or gaseous) presents significant difficulties. First, it is much more difficult to measure the pressure pulsations in the medium than the vibration of the pipeline itself. This is especially true for high-energy pipelines. Secondly, the numerical analysis of the fluid dynamics of the medium is much more labour-intensive than the analysis of the vibration of the steel part of the piping.

The proposed method for simulating the flow-induced vibration makes it possible to reproduce the vibration state of the pipeline precluding the need for hydrodynamic calculations and/or measurements of pressure pulsations. The article describes the proposed method and gives example of practical application.

BASIC CONCIDERATIONS

Piping vibration can be caused by two kinds of excitations:

a) mechanical impacts on the pipeline from equipment and supports, caused by the imbalance of moving masses, damage of bearings, etc.;

b) hydrodynamic loads on the pipeline from the medium carried by it. Pressure pulsations in the medium can, in turn, be caused by pump operation, vortex shedding with medium flowing around obstacles, general flow turbulence, two-phase flow and cavitation. See Olson (2002) for detailed overview.

Mechanical induced vibration is out of scope of this paper. Two-phase flow and cavitation have their distinctive characteristics and will not considered here either. Yet another restriction of the proposed method is the frequency content. Such method involves the use of piping beam models. Thus, it is not suitable for cases where shell modes are excited. The frequency of the first shell mode could be estimated with formula, see Kraus (1967):

$$f_i = \frac{1}{4 \cdot \pi \cdot \sqrt{3}} \sqrt{\frac{E}{\rho}} \cdot \frac{1}{\sqrt{1-\nu^2}} \cdot \frac{n \cdot (n^2-1)}{\sqrt{n^2+1}} \cdot \frac{s}{r^2} \tag{1}$$

Where E is the Young modulus, ρ is the pipe material density, ν is the Poison's ratio, n = 2 for the first shell mode, s is the wall thickness, r is the mean pipe radius. If the frequencies of the measured vibration fall within this range, a shell model of the pipe should be used, and the proposed method is not suitable for modeling the vibration state.

It is assumed that pressure pulsations occur at some obstacle (throttle, tee, valve, etc.) and propagate along the pipe at a certain velocity. This velocity for thick-walled pipes is close to the speed of sound in the medium:

$$c = \sqrt{\frac{K_w}{\rho_w}} \tag{2}$$

Where K_w is the bulk modulus, ρ_w is the medium density. The velocity could be adjusted taking into consideration of pipe wall elasticity by the Korteweg formula:

$$c' = \frac{c}{\sqrt{1 + \frac{K_w \cdot D}{E_s \cdot t}}} \tag{3}$$

Where c' is medium sonic velocity taking into consideration for pipe wall elasticity, c is the medium sonic velocity, K_w is the bulk modulus of medium elasticity, E_s is the Young's modulus of elasticity of the pipe material, D is the pipe outer diameter, t is the pipe wall thickness. Further, the velocity value can be adjusted to account for flow velocity in the pipe if it is significant.

The method assumes that damping of pressure fluctuations in the medium is not significant. However, if reliable information about attenuation is available, this can also be considered. Pressure pulsations in a medium induce dynamic forces in elbows, tees, throttles and pipe reducers. Action of these pulsations may be represented as a set of concentrated forces oriented along piping axis. For each pipe fitting direction and magnitude of resulting force-vector depend on the piping layout. For curved piping segments (elbows or bends) pressure pulsations produce the resulting force-vector as a combination of two vectors from adjusted straight pipes. Since the vector origin is located in the center of the bend curvature, it produces two in-plane bending moments. However, in the actual consideration these moments are neglected. For tee elements a resulting force-vector acts along side branch. Pressure pulsations in local hydraulic restrictions (throttle, reducer, etc.) cause forces acting along pipe axis. An example of applied forces is shown in figure 1. In most cases, these forces form pairs, for example, F1 and F2, F3 and F4. The counterpart forces are close in magnitude but differ in phase. The phase difference is determined by the time it takes for a pressure pulse to travel the distance from one elbow to another.

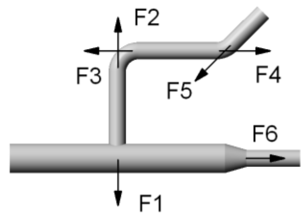

Figure 1. Dynamic forces applied to the beam piping model at tee, reducer and bends.

THE METHOD

The proposed method is based on considerations described above and includes the next steps.

Step 1. Vibration Measurements

Vibration measurement points on the piping must be selected frequently enough to capture possible local vibration modes. The measurement time step determines the width of the measured spectrum. It should be sufficiently small. For example, if you plan to explore the frequency range up to 100 Hz, then a step of 0.001 s would be recommendable. The duration of measurements determines the resolution of the spectrum - the frequency step. Typical measurement durations vary from 30 to 60 s. It is good practice to repeat measurements. This allows to assess the reproducibility of the results and select the most representative record.

To measure vibration, acceleration sensors are commonly used. The software must allow conversion of the signal from accelerations to velocities, as well as the Fourier spectrum calculation.

Step 2. Piping Model

The quality of the pipeline model determines the success of the proposed method. If the natural frequencies and shapes of the model greatly deviate from those for a real piping, it will not be possible to reproduce the vibration pattern.

The piping model must consider the weight of the medium and thermal insulation. The diameter and wall thickness must be selected taking into account actual manufacturing tolerances. The increased flexibility of the bends must be considered. The flexibility of the pipeline supports, including vessel nozzles, must be carefully calculated.

The primary check of the model quality is carried out by superimposing the natural frequencies of the model on the Fourier spectra of measured vibration velocities. The main peaks in the spectra should correspond to the natural frequencies of the model.

Step 3. Trial Excitation

A broadband signal such as white or pink noise is chosen as the basis for the trial excitation. The signal bandwidth should cover all significant peaks in the measured spectra. The same duration and time step of the signal are chosen as the ones from the measurement results.

One needs to choose a point - the source of perturbation. It can be a tee, valve, throttle, etc. The original signal will propagate from this point along the pipe back and forth. In this case, the original signal is considered to be pressure pulsations with a certain coefficient. The dynamic force acting on the pipeline element must be calculated as the pressure multiplied by the area determined by the pipe inner diameter.

Attenuation in the medium is not considered, so the pressure fluctuations are preserved along the entire length of the pipeline.

It is also necessary to consider the delay required for the pressure pulse to travel from the source of the disturbance to the point of the force application. The delay is calculated taking into account the propagation velocity (formula 3) and the distance from the source to the point of application. The final formula for dynamic force is as follows:

$$F_i(t) = k \cdot s(t - \frac{L_i}{V}) \cdot A_i \tag{4}$$

Where $F_i(t)$ is applied dynamic force, i is the force index (see figure 1), t is the time, $s(t)$ is the source signal, k is the scale factor for the transition from the source signal to pressure pulsations, L_i is the distance from the source to the point of application, V is the propagation velocity, A_i is the area determined by the inner diameter of the pipe. In practice, to consider the signal delay, it is enough to add the required number of zeros to its beginning.

Step 4. Excitation Adjustment

After applying the trial excitation at all nodes of the model, where the flow changes its velocity or direction, it is possible to calculate the vibrational state. In this case, records of velocities should be obtained at all nodes of the model corresponding to the measurement points. The result should be two sets of velocity records with the same time step and duration. The first set is the measurement result, while the second set is the calculation result. For each of these sets, the Fourier spectra, amplitude spectra, and total amplitude spectrum should be calculated according to the SRSS calculation rule.

Looking at the total amplitude spectrum a number of frequencies corresponding to the spectral peaks should be assigned. The boundaries of the spectral bands will be located between the peaks. They can be defined as the geometric mean:

$$frL_j = \sqrt{frp_{j-1} \cdot frp_j}\,; \qquad frR_j = \sqrt{frp_j \cdot frp_{j+1}} \tag{5}$$

Where frL_j is the left boundary frequency, frR_j is the right boundary frequency, frp_j is the j-th peak frequency.

RMS velocities should be calculated for each frequency band from the measurement total amplitude spectrum and from the calculated total amplitude spectrum. The ratio of these two values for each frequency band yields a coefficient for adjusting the spectrum of the trial excitation. Thus, the Fourier spectrum of the initial trial excitation should be multiplied in each frequency band by the resulting coefficient. Then the inverse Fourier transform should be performed to obtain the adjusted excitation. Note that adjusted pressure $k \cdot s(t)$ should be lower than absolute (static) pressure in the pipe.

The vibration load (dynamic forces F_i) should be obtained from adjusted excitation using the same procedure as for trial excitation.

Step 5. Quality Checking

Checking the quality of simulation of the vibrational state is carried out in two ways. In all cases, the measurement results and calculation results are compared.

The first way is to check the spatial distribution of vibration. This compares the RMS velocities for all directions and measurement points on the pipeline.

The second way is to check the frequency content of the vibration. In this case, individual amplitude spectra of velocities for different points and directions are compared.

If the control results are unsatisfactory, the pipeline model should be adjusted, or another source of pulsations should be selected. Then the whole procedure must be repeated from the beginning.

EXAMPLE

Problem Description

The steam piping connects two vessels. Medium parameters: temperature is 283 ºC, pressure is 6.4 MPa. Pipe sections are shown in Figure 2.

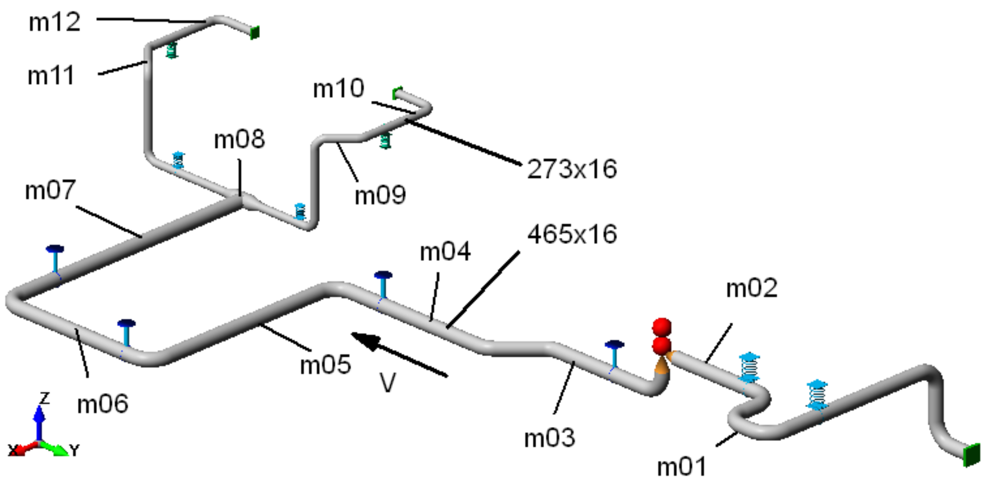

Figure 2. Piping layout and measurement points.

The support system includes 4 rigid hangers and 6 spring supports. The piping has one valve.

The piping is subjected to significant flow-induced vibration. Vibration was measured at 12 points (m01 - m12) as shown in Figure 2. At each point, vibration was measured in three directions. The RMS values of the measured velocities are shown in Table 1. Measurement directions are indicated for global axes as in Figure 2.

Table 1: Measured velocity RMS values, mm/s.

point	RMSx	RMSy	RMSz		point	RMSx	RMSy	RMSz
m01	24.1	42.8	17.5		m07	30.2	14.6	18.1
m02	29.8	10.9	18.7		m08	30.6	19.7	35.2
m03	23.2	11.2	7.9		m09	8.8	28.9	27.7
m04	33.2	18.3	12.5		m10	7.5	2.6	5.0
m05	23.5	12.3	11.3		m11	10.4	31.5	35.3
m06	12.2	19.8	6.6		m12	9.5	3.0	6.8

Measurements were made with a step of 0.001 s. The duration of the records is 30 s. A typical amplitude spectrum is shown in Figure 3.

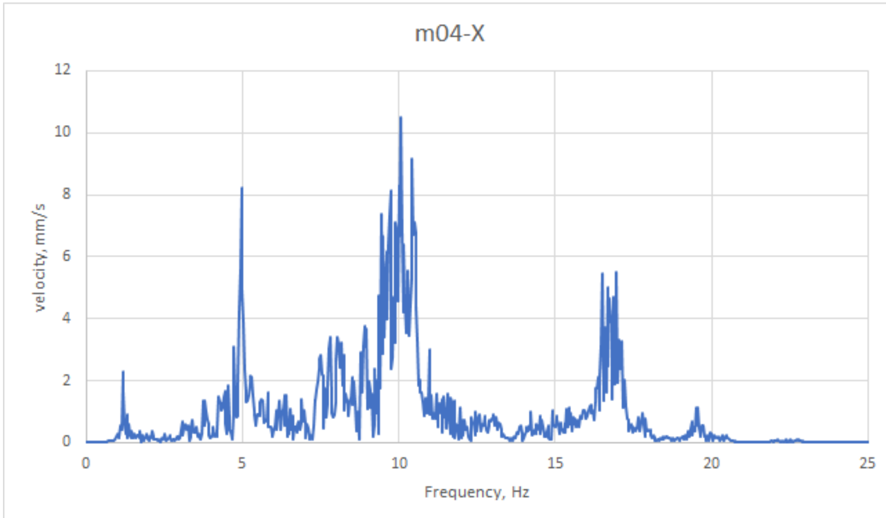

Figure 3. Typical amplitude spectrum. Measurement point: m04. Direction: X.

Piping Model

The piping model was created in the dPIPE software. The flexibility of the bends is calculated in accordance with the ASME code. The calculation of the stiffness of supports and vessel nozzles was carried out using the SOLVIA-03 software.

The first natural frequency of the piping model is 1.33 Hz. In Figure 4, the natural frequencies of the model are superimposed on one of the amplitude spectra. The model frequencies are shown as short vertical lines. Comparison of the spectrum peaks with the model frequencies allows one to reckon upon a successful simulation of the vibration.

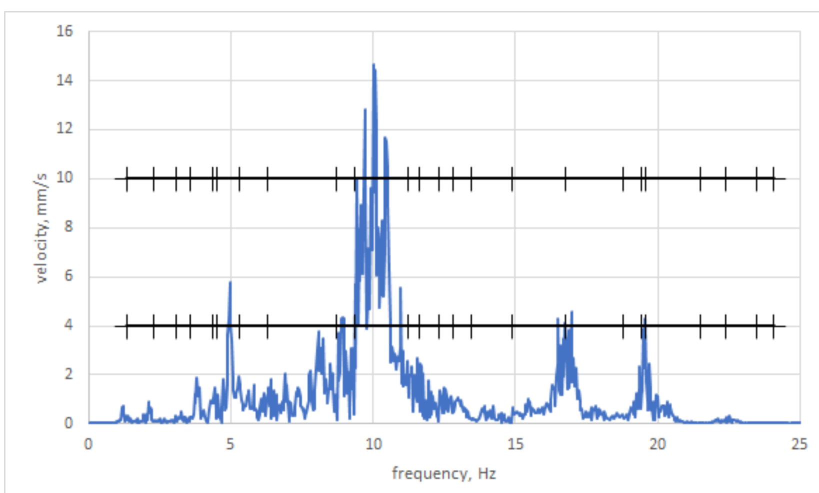

Figure 4. Amplitude spectrum compared to natural frequencies. Measurement point: m01. Direction: Y.

Trial Excitation

Broadband trial signal was generated with CVSpec program. The frequency band chosen is from 0.8 to 24 Hz. The amplitude spectrum of this signal is shown in Figure 5.

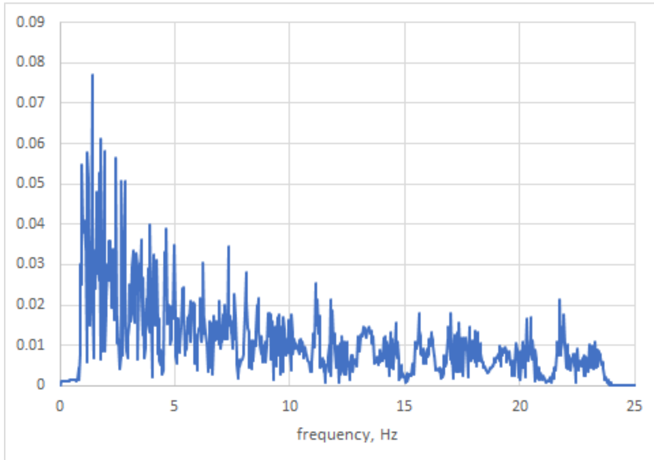

Figure 5. Amplitude spectrum of the trial signal

It was assumed that the initial pressure pulsations occur in the medium when passing through the valve. Then the pressure pulsations propagate along the piping at a certain speed. When passing bends and tees, pressure pulsations generate forces that act on the pipeline. The scheme of these forces is shown in Figure 6.

The following parameters were used to define each dynamic force. Pressure pulsation amplitude (*k* in formula 4) is 40 kPa. Each force was scaled with use the inner pipe area. Speed of sound in the steam is 495 m/s. Flow velocity is 9 m/s. Thus, the velocity of pressure pulsation along the flow is 504 m/s and the propagation velocity of the pressure pulsation against the flow is 486 m/s. The longest path length from the valve to the bend is 51.3 m. In this case, the pressure pulse delay is 0.102 s.

A trial dynamic load was applied to the model. As a result of the calculation, velocity records at the measurement points, their Fourier spectra, amplitude spectra, and total amplitude spectrum were obtained.

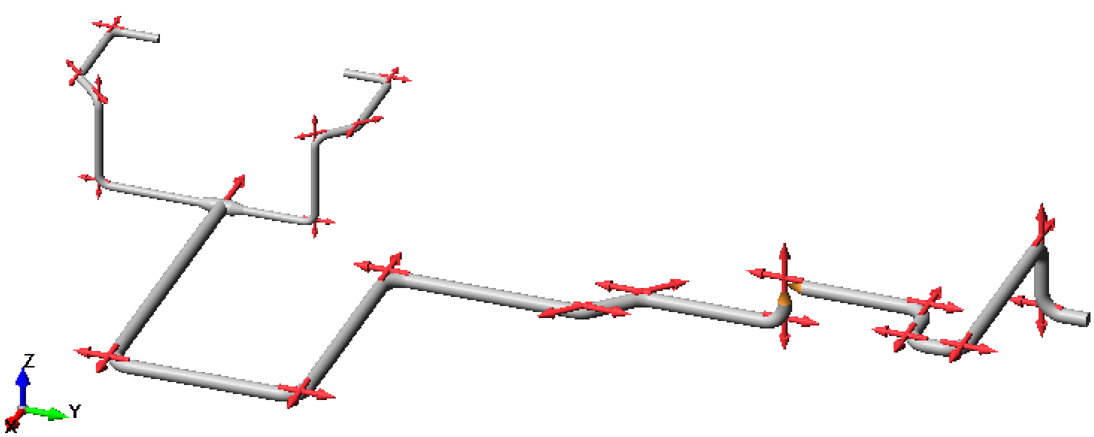

Figure 6. Dynamic forces applied to the model.

Adjusted Excitation Calculation

Figure 7 compares the total amplitude spectra obtained from the measurement results and obtained from the results of the calculation for a trial excitation.

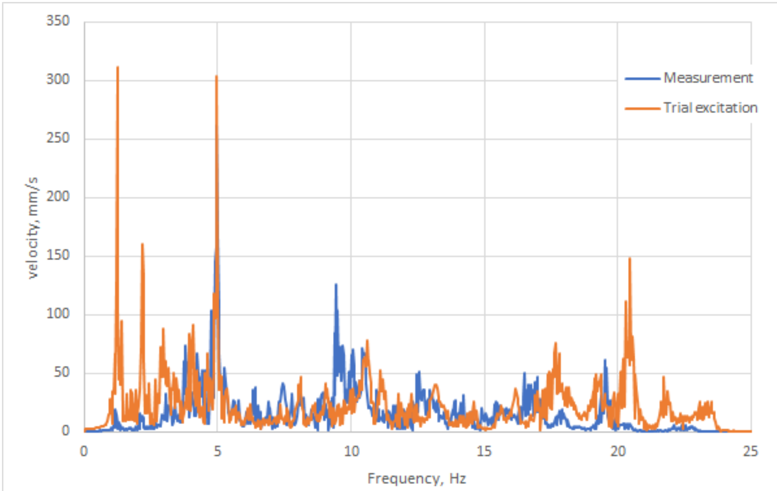

Figure 7. Measured total spectrum (blue) compared to calculated total spectrum (orange).

30 peaks were selected from the measured total spectrum. 30 frequency bands were formed using formula 5. For each frequency band RMS value was calculated from the measured total spectrum as well as from the calculated total spectrum. The ratio of these values determines the multiplier to adjust the trial excitation. The boundaries of the frequency bands and the calculated multipliers are shown in Table 2.

Table 2. Frequency bands and adjustment factors

No	Left, Hz	Right, Hz	Factor	No	Left, Hz	Right, Hz	Factor	No	Left, Hz	Right, Hz	Factor
1	0.00	1.56	0.08	11	7.61	7.96	1.46	21	14.56	15.16	1.02
2	1.56	2.54	0.12	12	7.96	8.51	0.90	22	15.16	15.96	1.39
3	2.54	3.41	0.34	13	8.51	9.16	0.90	23	15.96	16.73	1.15
4	3.41	4.01	0.96	14	9.16	9.92	3.10	24	16.73	17.43	1.07
5	4.01	4.48	0.89	15	9.92	10.70	1.08	25	17.43	18.62	0.25
6	4.48	4.85	1.09	16	10.70	11.74	0.65	26	18.62	19.43	0.36
7	4.85	5.11	1.06	17	11.74	12.77	1.41	27	19.43	19.99	1.26
8	5.11	5.78	1.24	18	12.77	13.43	0.83	28	19.99	21.48	0.09
9	5.78	6.86	1.45	19	13.43	14.07	1.12	29	21.48	23.72	0.14
10	6.86	7.61	1.92	20	14.07	14.56	1.42	30	23.72	30.00	0.28

The Fourier spectrum of the trial signal was "multiplied" by the factors from Table 2. The resulting spectrum was transformed into a new signal using the inverse Fourier transform. A comparison of the amplitude spectra of the trial and adjusted signals is shown in Figure 8.

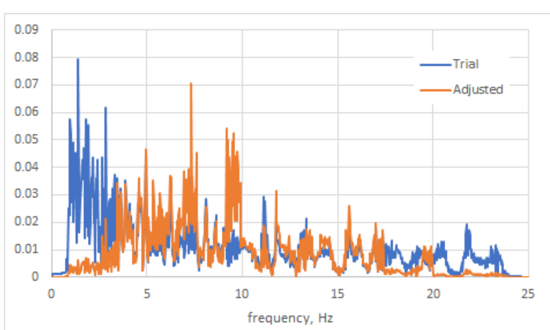

Figure 8. Amplitude spectra of the trial (blue) and adjusted (orange) signals.

After updating the source signal, the same steps were taken to form the vibration load as for the trial signal. A new adjusted dynamic load was applied to the model. As a result of the calculation, velocity records at the measurement points, their Fourier spectra, amplitude spectra, and total amplitude spectrum were obtained.

Spatial RMS distribution is shown in Figure 9. For most points, the analysis results give a good approximation to the measurement results. The maximum deviation is observed at point m09 for the Z direction and is -33.5%.

Comparison of the frequency content also indicates a good approximation of the calculation results to the real vibrational state. Figure 10 compares the amplitude spectra for point m01, direction Y.

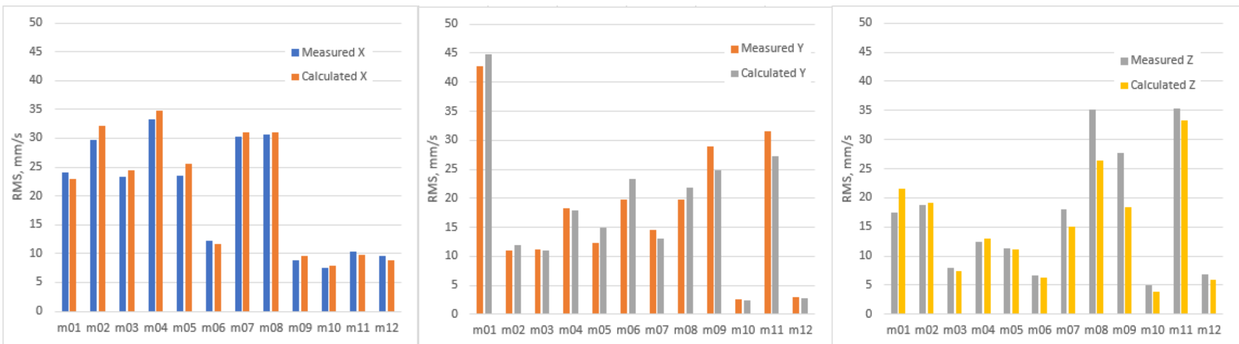

Figure 9. Velocity RMS spatial distribution. Measured and calculated values compared.

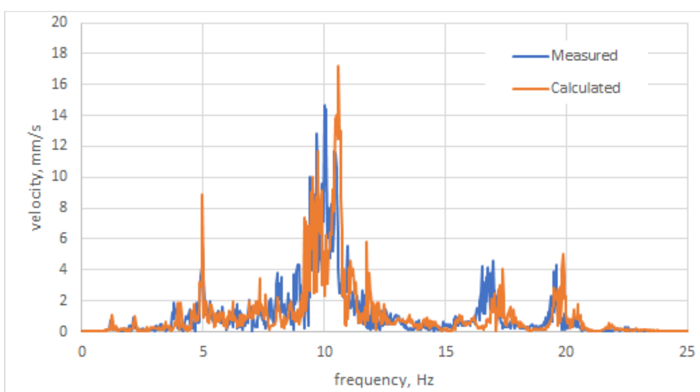

Figure 10. Measured and calculated amplitude spectra compared. Point m01, direction Y.

Further use

The resulting piping and vibration loading model can be used to calculate fatigue strength and predict service life. Another way to use the model is to develop measures to reduce vibration, for example, by installing viscous dampers. Figure 11 shows the arrangement of dampers obtained as a result of variant calculations. The aim of the optimization was to reduce RMS velocities below 15 mm/s with a minimal number of devices. Six dampers were recommended and installed. Figure 12 shows the spatial distribution of RMS velocities, comparing the initial vibration to the vibration upon damper installation. For damper modelling technique see, for example, Berkovsky et al. (2009).

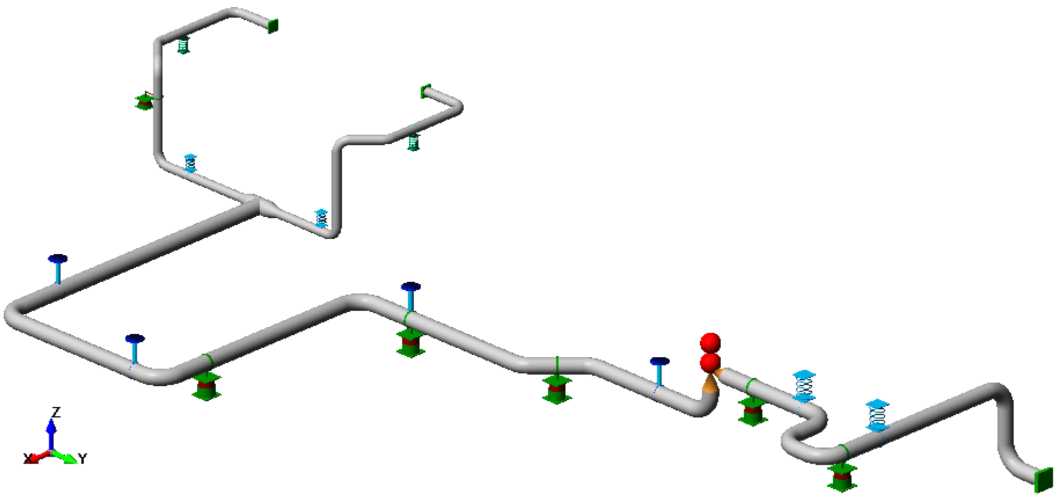

Figure 11. Arrangement of viscous dampers.

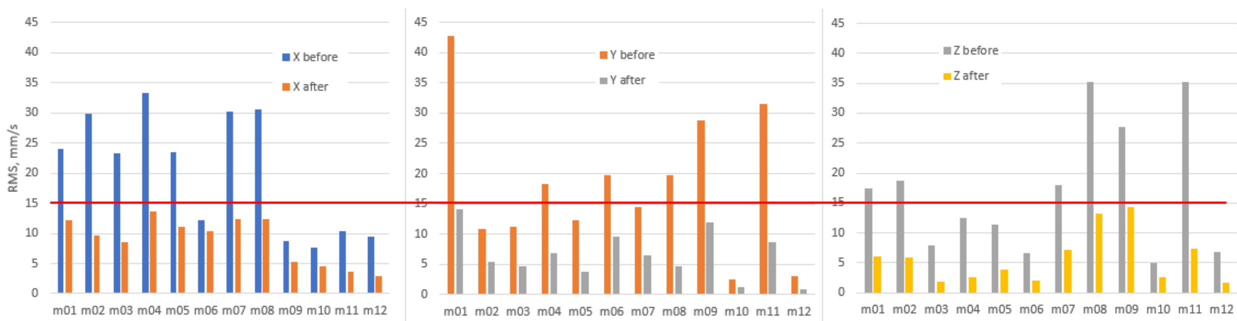

Figure 12. Velocity RMS spatial distribution. Initial vibration compared to the vibration upon damper installation.

CONCLUSION

A simple engineering method is proposed for modeling the piping vibration when the vibration is excited by the pressure pulsation in the medium. The method does not require either measurement of pressure fluctuations in the pipe or hydrodynamic analysis. However, the method is quite demanding of the piping model quality.

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