Research Article

On the Suitability of Vibration Acceptance Criteria of Process Pipework

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The risk of vibration-induced fatigue in process pipework is usually assessed through vibration measurements. For small-bore pipework, integrity personnel would measure the vibration of the pipework and refer to widely used charts to quantify the risk of vibration-induced fatigue. If the vibration levels are classified as OK, no action is required on behalf of the operators. However, if it is a CONCERN or PROBLEM vibration level, strain measurements are required to adequately quantify the risk through a fatigue life assessment. In this paper, we examine the suitability of a widely used vibration acceptance criteria through finite element models. A total of 4,800 models are used to study the suitability of this vibration acceptance criteria by monitoring both the vibration and dynamic stress. The model comprises a small-bore pipe (2″ SCH 40) that is fitted on a mainline size 5″ SCH 40 using a weldolet; the length of the mainline takes three values resulting in three models. The mainline supporting conditions will be varied using translational and rotational springs. The finite element models will be excited using a point load resembling flow-induced forces (with varying flow velocity and fluid composition). These excitations are obtained from the literature and are based on experimental studies as power spectral density functions. The results show that the studied vibration acceptance criterion is suitable in 99.73% of all the studied models with 68.27% confidence level. For the models with a shorter mainline pipe, the criterial is suitable in 76.5% of the time with 68.27% confidence level.

1. Introduction

The vibration of process pipework can lead to the development of fatigue cracks/failures which lead to hydrocarbon leaks. Statistics have shown that the root cause of about 20% of all incidents of loss of containment in the UK sector of the North Sea is vibration-induced fatigue (VIF) [1]. VIF occurs when dynamic stresses near a welded connection (or even within the parent material of the pipe) induce a crack which then propagates and causes loss of containment of the pressurized fluid.

Ideally, the risk of VIF should be quantified using strain measurements. However, in a live process plant, it is not always possible to install strain gauges (for example, on hot piping). Moreover, the installation of strain gauges can be time-consuming (requiring surface preparation, insulation removal, etc.) and usually requires the mobilization of specialist personnel. An alternative method to assess the risk of VIF is to measure the vibration of the pipework of concern. This approach has been widely adopted by the oil and gas industry. The motivation for using vibration measurements as a screening tool when assessing the risk of VIF is the feasibility and ease of implementation. A general-use accelerometer (along with a data acquisition system) can be used to record the measurements with minimal training to the personnel conducting the measurement [2]. The signal can be quickly analyzed and used as a first assessment of the risk of VIF. The recorded time history of the vibration can be postprocessed in the frequency domain and compared against vibration acceptance criteria.

Although design guidance is provided in standards for the nuclear industry [3] and reciprocating gas compressors such as ASME OM-3 [4] and API 618 [5], respectively, there are no API/ASME standards for the evaluation of the risk of VIF in in-service pipework. Large-scale experiments on in-service pipework were only reported by Müller Christoph [6] for modal and static analysis of pipes in the nuclear power sector. Recently, Kim et al. [7] proposed using a seismic isolation system on a carbon steel pipe elbow in a nuclear power plant and used a damage index to establish failure criteria for low-cycle fatigue. However, this approach would require instrumenting pipework throughout a plant which might not be feasible or practical in a process plant.

The most widely used vibration acceptance criteria (VAC) are shown in Figure 1; these criteria were adopted by the Energy Institute (EI) [8] which led to its wider acceptance in the community. The EI acceptance criteria comprise three classifications that are separated by two curves. The three classifications are OK, CONCERN, and PROBLEM. To use this VAC, the user would measure the vibration of the pipe, calculate the root mean square (RMS) of the vibration velocity, and identify the dominant peak frequency in the RMS spectrum. These two values (RMS of the velocity and dominant peak frequency) are sufficient to classify the pipe's vibration. However, the conceptual foundations for this VAC and the level of conservatism implanted in it are not well understood. Although the existing screening procedures are warranted by practice, the evaluations are open for interpretation by the user. The EI VAC was derived from peak-to-peak displacement-based vibration criteria which were first introduced by the Southern Gas Association [9]. The permissible velocity amplitudes (0-peak) were divided into units of RMS velocity by a crest factor of $\sqrt{2}$ which makes this transformation only relevant to sinusoidal vibrations (e.g., vibration due to rotating machinery). In addition, a constant amplitude fatigue limit has been set which may not be satisfactory in cases of Gaussian vibration or strong kurtosis (non-Gaussian) vibration [10, 11]. Alternatively, the displacement criterion includes the estimation of peak-to-peak displacement and five classifications [12]. These standards were proposed for average piping structures designed in compliance with sound engineering practices and that additional provision for sensitive applications or unreinforced branch connections should be given. These comments, along with numerous choices of acceptability limits and lack of usable technological foundation, put an enormous amount of pressure on the user [13].

The curves are based on a stress amplitude of 2500 psi which corresponds to 17.5 MPa. Weld class F2 of [14] has a constant amplitude fatigue limit of 35 MPa. Table 1 shows the recommended dynamic stress ranges and the corresponding classifications. Given that it is unlikely that the user would be able to ascertain the condition of the root of the weld of in-service pipework, a safety factor is applied to the constant amplitude fatigue limit which coincides with the 17.5 MPa mentioned above. The EI limits consist of ten tested classes of small-bore cantilever finite element models and one class of mainline pipes [8]. The precise number of geometry variants analyzed within each class is unclear and
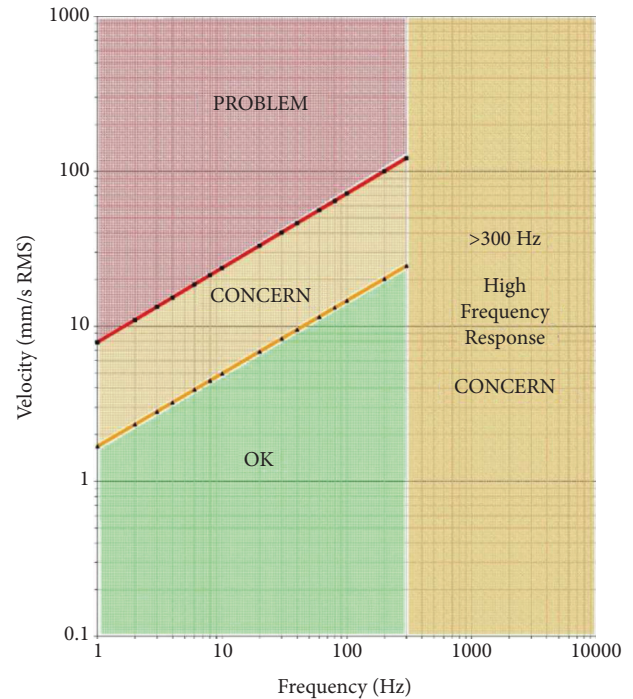

Figure 1: Vibration acceptance criteria (VAC) that were adopted by the EI [8].

Table 1: Stress limits for hydrocarbon processing pipework as adopted in [8].

Stress level	Stress range (MPa PTP)
OK	Stress <17.5
CONCERN	17.5< stress <35
PROBLEM	Stress >35

the type of finite element analysis (FEA) that was not conducted is not clarified [15]. In addition, the basis of the PROBLEM curve is unclear aside from it being roughly a factor of 4.9 greater than the CONCERN curve.

Using the curves of the EI vibration acceptance criteria for random vibrations can be troublesome for two reasons. First, the presumed crest factor of $\sqrt{2}$ was possibly used to cast the produced FEA velocity from 0-pk units to RMS units, which is unconservative for most random vibrations [11, 13]. Second, a constant amplitude fatigue limit has been developed which may not be satisfactory in cases of Gaussian vibration or high kurtosis (non-Gaussian) vibration. The EI proposed using the VAC sown in Figure 1 based on the work of the Southwest Gas Association [9]. There is, however, no agreement on pipework vibration acceptance criteria. The EI [8] curves were altered to become more conservative [16]. Operators in Norway, for example, use an aggregate of the EI [8] curves and standards established by the European Forum of Reciprocating Compressors [17]. The recommendations in [17] are more specific for machine-induced excitations and gas pulse movement—flow-induced vibrations are not specifically treated.

Because of the above variations in the assessment of pipework vibrations severity, important dynamic stress

locations may be missed which results in VIF risks not being managed properly. Besides, the permissible vibrational rates may rely on variables including pipe length, the support structure (and condition), fittings, and deflection types. Therefore, in all piping geometries, arrangements, and fittings, the criteria in [8] might not be suitable to accurately quantify the risk of VIF.

This study aims to assess the EI VAC by conducting finite element analysis (FEA) of three different setups. Each setup will comprise of a mainline pipe, and a small-bore pipe that supports a valve. Each setup will have a different length for the mainline pipe. Random vibration analysis will be conducted on the three FE models to simulate different flow and supporting conditions. Random vibration analysis will allow the focus on flow-induced vibration which was not adequately addressed in the literature and is not fully covered by the EI VAC. After simulating these models, the results will be compared against the VAC and the stress limits to ascertain the suitability of the VAC. In addition, these results will demonstrate the 1- and 3-sigma confidence levels of the VAC.

The remainder of this paper is organized as follows. In Section 2, the FE model will be introduced along with the inputs and outputs of the simulation. The variables used to create different case scenarios will be presented in Section 3. Following that, Section 4 will present the results and discussions explaining the simulation outputs. The concluded findings will be presented in Section 5.

2. FE Modeling and Analysis Setup

To assess the vibration acceptance criteria, three models were created using SolidWorks® and then imported into ANSYS Workbench®. Each model comprises of a 5″ SCH 40 carbon steel mainline that is connected to a 25 cm, 2″ SCH 40 branch pipe using a 2″ × 5″ SCH 40 small-bore connection (SBC). The pipe, weldolet, and branch were assumed to have a density of 7850 kg/m^3, Poisson's ratio of 0.3, and modulus of elasticity of 200 GPa. For the SBC, a weldolet was selected since this is a very common SBC that is used in the industry although it is usually more expensive than other types of SBCs (e.g., contoured body, welded tees). The branch pipe supports a 5 kg valve. The length of the mainline pipe is 0.3 m in Model 1, 1 m in Model 2, and 1.5 m in Model 3. The three parts (branch pipe, run pipe, and weldolet) were modeled separately and then assembled in SolidWorks®. A fillet weld was added to join the mainline with the weldolet and another fillet weld was added to join the weldolet and the branch pipe. The valve's geometry was not included in the model since the frequency of excitation is low (as will be discussed later in Section 3) and the valve's natural frequency will be very high compared to the excitation frequencies. Thus, the valve was instead modeled as a point mass. To simulate different supporting conditions, the mainline pipe was supported by rotational and translational springs (whose values will be discussed in Section 2.1). Figure 2 shows the general geometry of the pipe and the branch. The ball at the end of the branch pipe is the point mass mentioned earlier that represents the valve. The springs
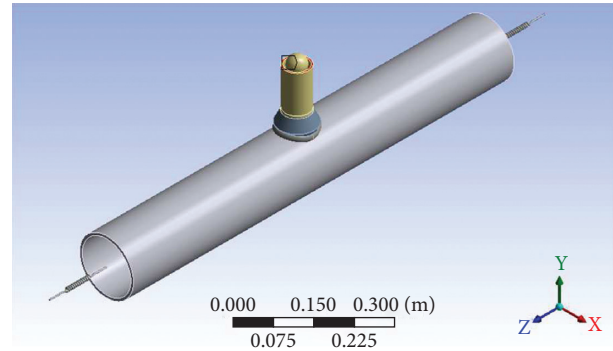

Figure 2: Setup of the generic FE model of the pipe and branch. A 5 kg point mass at the end of the branch pipe represents the valve.

shown in Figure 2 have translational and rotational stiffness and these are discussed in Section 2.1.

The FEA was conducted in ANSYS Workbench®. Solid elements were used to mesh the different parts of the model. Figure 3 shows a sample mesh that was used in this study. The element size was chosen to allow for 12 elements per wavelength (at the maximum anticipated frequency of 100 Hz) as the rule of thumb in FEA suggests [19]. After defining the boundary conditions, a modal analysis was conducted, and the first six modes were extracted and saved to be used in the random vibration analysis.

2.1. Inputs and Outputs of FE Model. Random vibration analysis was conducted using ANSYS Workbench®. The mainline pipe was excited by the power spectral density (PSD) of a force that is generated from the flow of fluid. The force was applied in the middle of the pipe in the *x*-direction (see Figure 2). For the flow-induced excitation, two parameters will be changed: the liquid/gas ratios and the flow velocity as discussed in Section 3. Four values of liquid/gas ratio will be considered along with 16 flow velocities. Furthermore, the translational and rotational springs will be varied as well to simulate different supporting conditions. Five sets of translational and rotational spring values will be considered. There are three FE setups, resulting in a total of 4800 FE models.

Using the inputs, the simulation will provide the necessary outputs which will be used to evaluate the suitability of the VAC. From each model, the vibration velocity of the valve at the end of the branch pipe will be obtained. This will be output in the form of the RMS (i.e., 1-sigma value) and 3-sigma value. These values will be checked against the VAC of Figure 1. The second output is the stress levels which will be evaluated with the vibration velocity against the VAC of Figure 1 to see how appropriate the VAC is when judging random vibrations as will be shown in Section 4.

3. Excitation and Supporting Conditions

Riverin and Pettigrew [19] experimentally modeled flow-induced forces using different flow regimes and flow speeds. They managed to characterize in-plane forces inside pipes and their results showed that the flow-induced forces can be
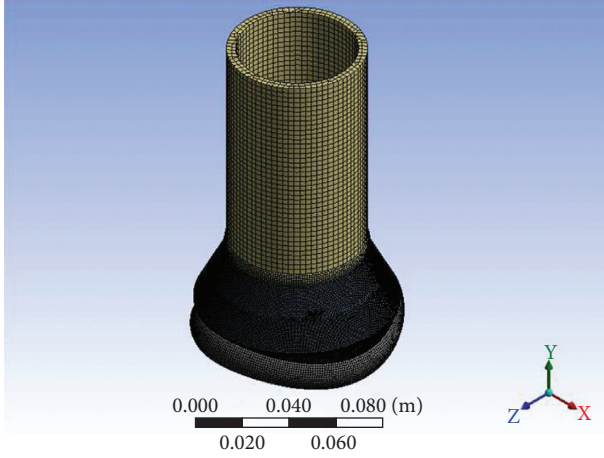
FIGURE 3: Sample mesh of the branch pipe and the welds between the mainline pipe and the weldolet and between the weldolet and the branch pipe; the mainline pipe is hidden to enable showing the fine mesh at the welds and at the weldolet.

FIGURE 4: Sample PSD curves generated for different cases of excitation.

described using a PSD, e.g., Figure 4. Their main finding is that the PSD of the force can be nondimensionalized in frequency and amplitude. Their results for the force will be adopted in this work.

Riverin and Pettigrew [19] used two variables to determine the flow conditions: superficial velocity, v, and volumetric quality (or homogeneous fraction of the void), designated as

$$\beta = \frac{Q_g}{Q_g + Q_l},$$
$$v = \frac{Q_g + Q_l}{A}, \tag{1}$$

where Q_g and Q_l are the volumetric flow rates of gas and liquid, respectively, and A is the total flow area. For this simulation, the superficial velocity will vary from 5 to 20 m/s. This range is typical for flow in pipework conveying hydrocarbons. Indeed, the 20 m/s flow velocity is considered by many operators as the limiting flow velocity in process pipework [20]. Four values for the volumetric quality will be used: $\beta = 25\%$, 50%, 75%, and 95%. Riverin and Pettigrew [19] reported their results in the form of a dimensionless frequency, $\overline{f}$, and a normalized PSD (of the force), $\overline{PSD}$:

$$\overline{PSD(\overline{f})} = \frac{PSD}{(GD)^2}, \tag{2}$$

$$\overline{f} = \frac{fD}{v}, \tag{3}$$

where f is the frequency in Hz, D is the diameter of the run pipe in m, and G is the mass flux of the mixture:

$$G = \left[\rho_g \beta + \rho_l (1 - \beta)\right] v. \tag{4}$$

Equations (2) and (3) comply with the established terminology of [21]. Experiments showed that the PSD of the
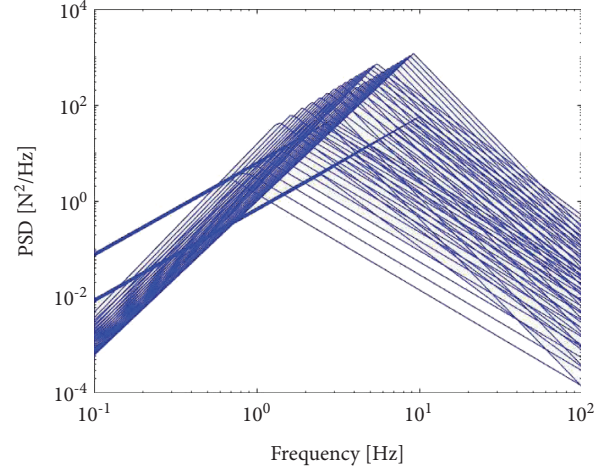
force can be described by a peak with two exponentially decaying flanks. According to [19], the nondimensional PSD is a function of the nondimensional frequency and few other parameters that were obtained experimentally. The nondimensional PSD is given as

$$\overline{PSD(\overline{f})} = \begin{cases} \left[\dfrac{\overline{PSD(\overline{f}_0)}}{(\overline{f}_0)^{m_2}}\right](\overline{f})^{m_1}, & \text{if } \overline{f} < \overline{f}_0 \\[4mm] \left[\dfrac{\overline{PSD(\overline{f}_0)}}{(\overline{f}_0)^{m_2}}\right](\overline{f})^{m_2}, & \text{if } \overline{f} \geq \overline{f}_0 \end{cases} \tag{5}$$

The values of $\overline{f}_0$, $\overline{PSD(\overline{f}_0)}$, m_1, and m_2 are dimensionless parameters that were obtained experimentally in [19] and are presented in Table 2 for completeness.

Since the input to the model is the PSD of the force, the response (in terms of velocity or stress) will be obtained as a PSD function as well. The mean square value of the response (be it velocity or stress) can be obtained as [22]

$$\overline{y^2} = \int_0^\infty |H(f)|^2 S_x(f) \, df, \tag{6}$$

where $S_x(f)$ is the PSD of the excitation, given in (2) above, and $H(f)$ is the (velocity or stress) system transfer function which will be obtained as

$$H(f) = \frac{1/k}{\left[1 - (f/f_n)^2\right] - i\left[2\zeta(f/f_n)\right]}, \tag{7}$$

where i is the unit imaginary number. In this paper, the transfer function will be obtained using FEA and the natural frequencies, f_n, will be obtained by the modal analysis that will be conducted in ANSYS Workbench®. Notice that, in equation (7), k is the equivalent stiffness of the pipe, and ζ is the damping ratio (set to 0.1% since pipework systems are usually lightly damped).

Replacing the next relation to equation (6),

$$S_x(f) = PSD(f) = \overline{PSD(f)}(GD)^2, \tag{8}$$

TABLE 2: Coefficient used in equation (6).

β (%)	$\overline{f}_0$	$\overline{PSD}(f_0)$	m_1	m_2
25	0.064	1.88E-05	1.9	−2.5
50	0.059	8.58E-04	3.2	−3.5
75	0.035	1.97E-03	3.5	−2.5
95	0.018	1.99E-03	3.5	−2.0

and applying equation (5) to evaluate the $PSD(f)$, the vibration response can be approximated. Figure 4 shows a group of PSD functions of the force applied to the pipe. As discussed earlier, the PSD function of the force has a peak and two exponentially decaying flanks.

To simulate different supporting conditions, different translational and rotational support stiffness values were used, as shown in Table 3. These values were reported in [11] and are typical values for the supporting stiffness in process pipework in operating hydrocarbon and petrochemical plants.

Table 4 summarizes all the parameters and variables used in the analysis. The frequency range was set to 0.1 Hz–100 Hz which covers flow-induced vibration problems [15] that are typically in the low-frequency range.

4. Results and Discussion

In this section, the 1-sigma and the 3-sigma results which were obtained from the simulations are presented and discussed for three models considered (0.3 m, 1 m, and 1.5 m mainline length).

4.1. Confidence of 1-Sigma. A total of 1600 cases were simulated for each model (Model 1, Model 2, and Model 3 of Section 2). The RMS and the peak frequency at the valve location were compared to the VAC of Figure 1. The highest stress in the welds were compared to the stress acceptance criteria based on [14]. For Model 1 (mainline length of 0.3 m), there was a total of 1208 cases (out of the 1600 cases) with "OK" (OK) vibration levels. Within these 1208 cases, 1048 cases have "OK" (OK) stress levels; these cases are denoted as "OO" (i.e., vibration level is OK according to the EI VAC and stress level is OK according to [8, 14]). Out of the remaining 160 cases (which have "OK" vibration classification), 72 cases have "CONCERN" (C) level of stress and 88 cases have "PROBLEM" (P) stress levels; these two cases were referred to as "OC" and "OP," respectively. Furthermore, 269 cases were found to have "CONCERN" (C) vibration classification. Out of these 269 cases, there are 87 cases with OK stress levels designated by (CO), 54 cases with C stress level designated by (CC), and 128 cases with P stress level designated as (CP). In addition, 123 cases were found to have P vibration level, 1 case was found to have a C stress level and was denoted by (PC), and 122 cases had P stress levels referred to as PP. These results are summarized in Table 5 which shows the number of cases in each category.

From Table 5, a total of 1224 cases were correctly judged by the VAC; that is, the VAC's classification was consistent with the stress classification (i.e., OO, CC, and PP). These

TABLE 3: Translational and rotational support stiffness values [11].

Translational support stiffness		Rotational support stiffness	
(N/m)	(lb/in)	(N-m/rad)	(lb-in/rad)
112.98	1×10^3	1129.85	1×10^4
5705.73	5.05×10^4	57057.34	5.05×10^5
11298.48	1×10^5	112984.83	1×10^6
570573.38	5.05×10^6	5705733.88	5.05×10^7
1129848.29	1×10^7	11298482.93	1×10^8

1224 cases represent 76.5% percent of the 1600 cases that were simulated for Model 1. The results presented in Table 5 and further displayed in Figures 5–7 are with a confidence level of 68.27%. This means that 76.5% of the time the VAC can be used to correctly judge the vibration levels with a confidence of 68.27%. Figure 8 shows a sample of the results for Model 1 where the classification of the vibration was CONCERN (as per Figure 1) whereas the stress classification was OK (as per Table 1). Figure 9 reveals more interesting results: the vibration classification was OK (as per Figure 1) whereas the stress classification was PROBLEM (as per Table 1).

4.2. Confidence of 3-Sigma. For hydrocarbon applications, 68.27% confidence level might not be sufficient given the potential severity of the human, environmental, and financial losses that may be encountered in the case of loss of containment of hydrocarbons. Hence, it is prudent to check the suitable VAC for usage with 99.73% which corresponds to the 3-sigma confidence level. Table 6 illustrates the findings with a 99.73% confidence. From this table, it can be found that 1029 cases out of the 1600 cases were judged correctly by the VAC (i.e., both the vibration and stress are of the same classification as per Section 1). These cases represent 64.31% of the 1600 cases; that is, the VAC presents a 99.73% confidence assessment 64.31% of the time.

4.3. Different Run Pipe Lengths. The same process and analysis were conducted on the 1 and 1.5 m mainline pipes to check whether the length of the run pipe affects the results or not. For the 1-meter run pipe model (Model 2), it was found that 100% of the time the VAC correctly judged the vibration levels with 68.27% confidence. Moreover, for a 99.73% confidence level, the VAC correctly judged the vibration levels 64.06% of the time. Similarly, for the 1.5 m run pipe (Model 3), the VAC correctly judged the vibration levels with 68.27% and 99.73% confidence 100% of the time. Sample results for these two models are presented in Figures 10 and 11.

TABLE 4: Varied parameters and parameter ranges for FEA pipe models.

Variable	Variable options/ranges	Assumption
Translational support stiffness	Table 3	Reported in [11]
Rotational support stiffness	Table 3	Reported in [11]
Volumetric quality (or homogeneous void fraction), β	25, 50, 75, and 95%	Based on the experimental work of [19]
Superficial velocity, V	5, 6, 7, …, 20 m/s	Typical values of velocity in pipework for hydrocarbons
Coefficients: $\overline{f_0}$, $\overline{PSD(f_0)}$, m_1, and m_2	Table 2	Based on the experimental work of [19]
f	0.1 Hz–100 Hz	Frequency range of flow-induced vibrations

TABLE 5: Classification of cases and the number of cases in each category (1-sigma results, 68.27% confidence).

Category	Description	Number of cases
OO	OK vibration, OK stress	1048
OC	OK vibration, CONCERN stress	72
OP	OK vibration, PROBLEM stress	88
CO	CONCERN vibration, OK stress	87
CC	CONCERN vibration, CONCERN stress	54
CP	CONCERN vibration, PROBLEM stress	128
PO	PROBLEM vibration, OK stress	0
PC	PROBLEM vibration, CONCERN stress	1
PP	PROBLEM vibration, PROBLEM stress	122
Total		1600

— Concern Vibration Line
— Problem Vibration Line
• Simulation Results

FIGURE 5: Simulations with OO category compared to the VAC for Model 1.

— Concern Vibration Line
— Problem Vibration Line
• Simulation Results
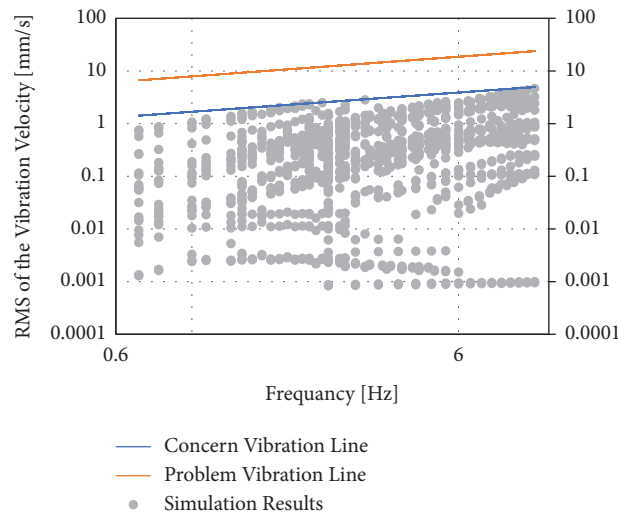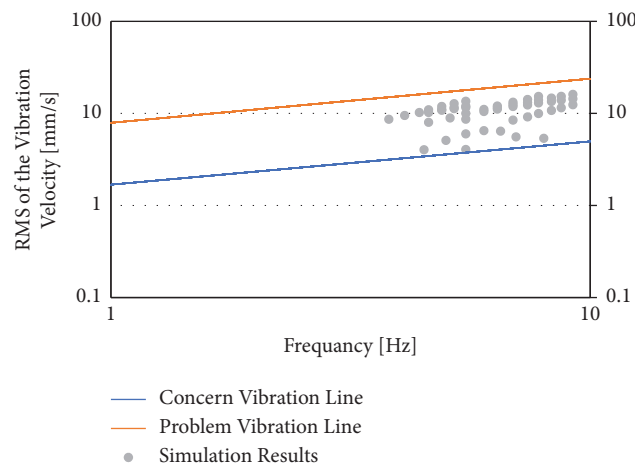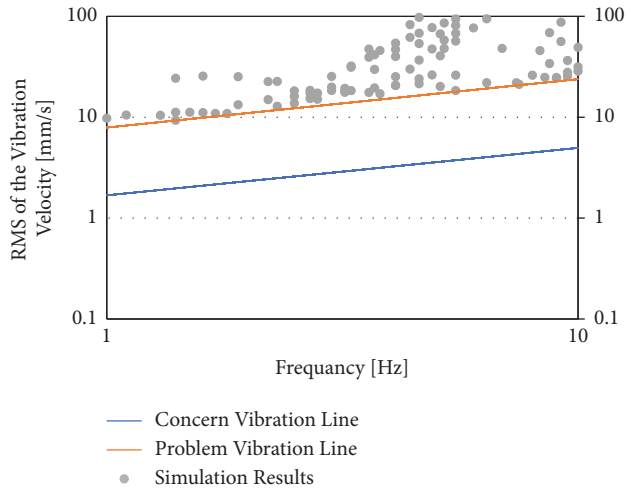
FIGURE 6: Simulations with CC category compared to the VAC for Model 1.

FIGURE 7: Simulations with PP category compared to the VAC for Model 1.

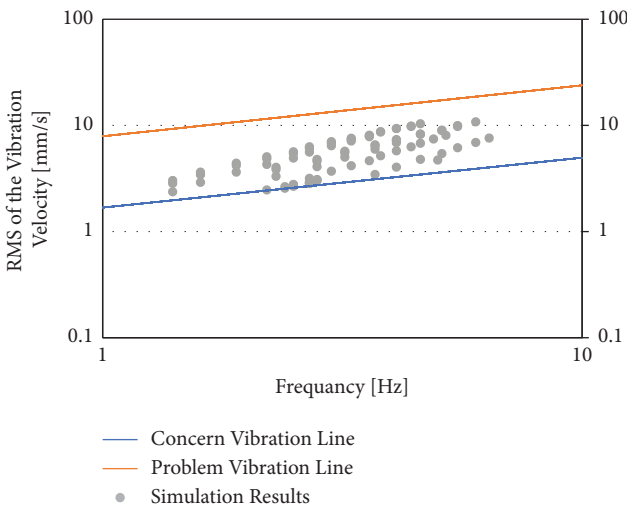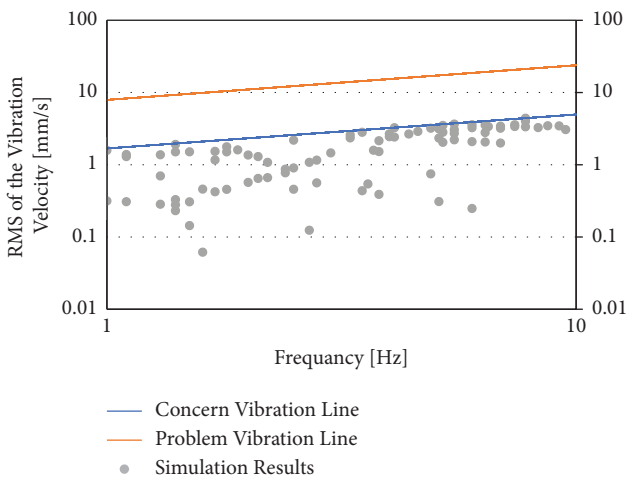TABLE 6: Classification of cases and the number of cases in each category (3-sigma results, 99.73% confidence).

Category	Description	Number of cases
OO	OK vibration, OK stress	676
OC	OK vibration, CONCERN stress	109
OP	OK vibration, PROBLEM stress	91
CO	CONCERN vibration, OK stress	178
CC	CONCERN vibration, CONCERN stress	50
CP	CONCERN vibration, PROBLEM stress	169
PO	PROBLEM vibration, OK stress	1
PC	PROBLEM vibration, CONCERN stress	23
PP	PROBLEM vibration, PROBLEM stress	303
Total		1600

FIGURE 8: Simulations with CO category compared to the VAC for Model 1.

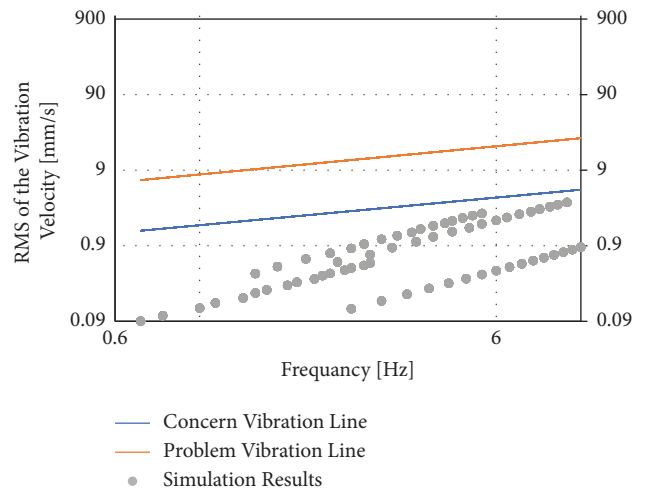FIGURE 10: Simulations with OO category compared to the VAC for Model 2.

FIGURE 9: Simulations with OP category compared to the VAC for Model 1.
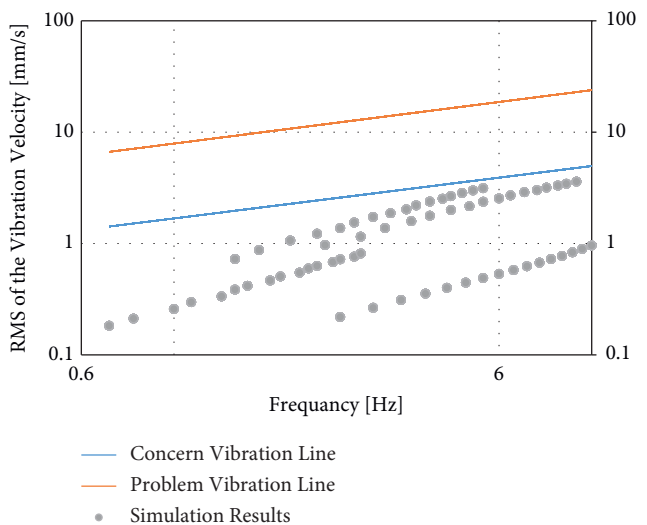

FIGURE 11: Simulations with OO category compared to the VAC for Model 3.

5. Conclusion

The objective of this paper was to assess the suitability of the vibration acceptance criteria (VAC) adopted by the Energy Institute to correctly correlate the dynamic stress levels in the small-bore connection of the pipework. The paper focused on flow-induced vibrations modeled as power spectral density (PSD) forces applied to a mainline/small-bore pipe setup with the mainline having three different lengths and varying supporting conditions. It was found that, for a short mainline pipe, the VAC can correctly judge the vibration levels with a confidence of 68.27% only 76.5% of the time. In other words, both the VAC and acceptable dynamic stress ranges as per the Energy Institute guidelines [8] are observed 76.5% of the time with a confidence level of 68.27% which corresponds to the 1-sigma results. This drops down to 64.31% when a confidence level of 99.73% is required. Based on the work done herein, the VAC seems to be more suitable for longer span pipes as demonstrated by the models that had a 1 m and 1.5 m mainlines. For both models with 1 m and 1.5 m mainline, the agreement between the VAC and the stress criteria was 100% with a confidence level of 67.27%. For a confidence level of 99.73%, the agreement between the VAC and the stress criteria dropped to 64.06% of the time for the model with 1 m mainline and 68.27% for the model with 1.5 m mainline.

Data Availability

Data used in this paper are openly available at Qatar University's Institutional Repository (http://hdl.handle.net/10576/25621).

Disclosure

Conflicts of Interest

The authors declare no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Acknowledgments

References

[1] A. McGillivray and J. Hare, *Offshore Hydrocarbon Releases 2001-2008, in HSE Books*, Health and Safety Executive, UK, 2008.

[2] M. Hamblin, "Fatigue of cantilevered pipe fittings subjected to vibration," *Fatigue and Fracture of Engineering Materials and Structures*, vol. 26, no. 8, pp. 695–707, 2003.

[3] K. Takahashi, S. Watanabe, K. Ando et al., "Low cycle fatigue behaviors of elbow pipe with local wall thinning," *Nuclear Engineering and Design*, vol. 239, no. 12, pp. 2719–2727, 2009.

[4] *Requirements for Preoperational and Initial Start-Up Vibration Testing of Nuclear Power Plant Piping Systems*, p. 25, American Society of Mechanical Engineers (ASME), New York, NY, 1983.

[5] Reciprocating Compressors for Petroleum, *Chemical and Gas Industry Services*, American Petroleum Institute, Washington, D.C, 2007.

[6] W. Müller Christoph, "Piping analysis of large scale experiments with adina and dapsy," *Computers & Structures*, vol. 26, no. 1, pp. 111–121, 1987.

[7] S.-W. Kim, S.-J. Chang, D.-U. Park, and B.-G. Jeon, "Failure criteria of a carbon steel pipe elbow for low-cycle fatigue using the damage index," *Thin-Walled Structures*, vol. 153, Article ID 106800, 2020.

[8] *Guidelines for the Avoidance of Vibration Induced Fatigue Failure in Process Pipework*, Energy Institute, London, United Kingdom, 2008.

[9] S. G. A. Pipeline and C. R. Council, *Controlling the Effects of Pulsations and Fluid Transients in Piping Systems: Seminar: Papers*, Southwest Research Institute, San Antonio Texes, United states, 1986.

[10] A. Niesłony, M. Böhm, and R. Owsiński, "Crest factor and kurtosis parameter under vibrational random loading," *International Journal of Fatigue*, vol. 147, Article ID 106179, 2021.

[11] M. F. P. Bifano, L. Breaux, A. J. Feller, and R. Brodzinski, "New fatigue screening criteria for the fitness-for-service assessment of in-service process piping vibrations," in *Proceedings of the ASME 2018 Pressure Vessels and Piping Conference*, Prague, Czech Republic, July 2018.

[12] W. W. v. Nimitz, "Reliability and performance assurance in the design of reciprocating compressor installations," in *Proceedings of the International Compressor Engineering Conference*, West Lafayette, Indiana, USA, July 1974, https://core.ac.uk/download/pdf/4955845.pdf.

[13] L. Breaux, S. McNeill, and G. Szasz, "Fitness-for-Service assessment of piping subject to random vibration," in *Proceedings of the ASME 2016 Pressure Vessels and Piping Conference*, Vancouver, British Columbia, Canada, July 2016.

[14] *BS 7608 Guide to Fatigue Design and Assessment of Steel Products*, British Standards Institute, London, UK, 2014.

[15] G. Hinz and K. Kerkhof, "System identification and reduction of vibrations of piping in different conditions," in *Proceedings of the ASME 2013 Pressure Vessels and Piping Conference*, Paris, France, July 2013.

[16] C. Harper, "Integrity evaluation of small bore connections (branch connections)," in *Proceedings of the 9th Conference of the EFRC*, Vienna, Austria, September 2014.

[17] A. Ejik, "EFRC guidelines for vibrations in reciprocating compressor systems," in *Proceedings of the 6th Conference of the European Forum for Reciprocating Compressors*, TNO Industrie en Techniek, Düsseldorf, Germany, October 2008.

[18] M. Petyt, *Introduction to Finite Element Vibration Analysis*p. 516, 2nd edition, Cambridge University Press, Cambridge, United kingdom, 2010.

[19] J.-L. Riverin and M. J. Pettigrew, "Vibration excitation forces due to two-phase flow in piping elements," *Journal of Pressure Vessel Technology*, vol. 129, no. 1, pp. 7–13, 2006.

[20] *B31.3 Process Piping*, American Society of Mechanical Engineers (ASME), New York, NY, 2021.

[21] M. J. Pettigrew and C. E. Taylor, "Two-phase flow-induced vibration: an overview (survey paper)," *Journal of Pressure Vessel Technology*, vol. 116, no. 3, pp. 233–253, 1994.

[22] W. Thomson and M. D. Dahleh, *Theory of Vibration with Applications*, Pearson, New Jercey, United States, 5th edition, 1997.