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DETERMINATION OF VIBRATION BORNE STRESS IN A PIPELINE USING FLUID STRUCTURE INTERACTION

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ABSTRACT

Excessive vibration is one of the leading causes of high cycle fatigue in process piping. Conventional way to assess the severity of vibration is to measure the stress caused by vibration with strain gages. An alternative way is discussed in this paper. The basis of this alternative way was formed by Operational Deflection Shape (ODS) analysis. In order to test the method, ODS was measured from a dual spiral system piping system which was a highly pressurized gas transporting pipeline in an offshore platform. Modal parameter such as natural frequency and mode shape were extracted from the result of ODS analysis and the modal parameter was used to correlate and verify a finite element (FE) model. Given the operating condition of the flow in the pipe, Fluid-Structure Interaction (FSI) was then performed where structural and fluid were fully coupled. From the FSI analysis, vibration-induced stress was estimated and the risk of fatigue failure of piping system can be concluded by comparing with allowable endurance limit of piping material.

Keywords: vibration-induced stress, dynamic stress, operational displacement shapes, piping vibration, fluid-structure interaction.

INTRODUCTION

Pipelines and pipe systems are of great importance in many industries. They provide transport for a wide range of substances (petrochemicals, water) and they fulfil safety functions – e.g. cooling systems in nuclear power plants.

Failure of piping systems can have disastrous effects, leading to injuries and fatalities as well as to substantial cost to industry and the environment. Besides, piping vibration problems in operating plants have resulted in costly unscheduled outages and backfits (Olson, 2002).

Vibration loading, typically mechanical or flow-induced, are the most common causes of high cycle fatigue (OECD/NEA, 2006). Besides, on a survey conducted by (Kustu & Scholl, 1980), pipe cracking was identified as the most frequently recurring problem, the most significant cause of which was determined to be piping vibration. Mechanical vibration was the cause of 22.3% of all reportable occurrences involving pipes and fittings.

The severity of vibration with respect to fatigue depends on numerous factors such as magnitude of the stress variation caused, number of anticipated occurrences of these variations during the estimated lifetime of the piping system, and different tolerance of cyclic loading in different grades of steel have (Vepsä, 2008). Because of the large number of stress cycles encountered in steady-state vibration, the allowable stress values must be determined from fatigue curves (Olson, 2002).

More detailed testing involves obtaining sufficient measurements to such as through the use of strain gauges allow pipe stresses to be accurately determined (Olson, 2002). Strain measurements are very useful for determining the effect of vibrations. A piping acceptance criterion is given in terms of maximum vibratory stress, thus strain measurements produce data directly applicable to them. In order to determine if measured vibration amplitudes of piping systems were acceptable, the dynamic stresses caused by the vibrations should be compared to the applicable endurance limit for the piping material (Wachel, 1995).

However, direct measurement of the dynamic stresses is time-consuming and complicated process. Therefore assessment of severity of vibration in pipelines is usually based on measurement of amplitude or velocity of vibration, complemented in some cases with the frequency spectrum of vibration. This is because the stress in a piping span which is vibrating at resonance is directly proportional to the maximum vibration amplitude (displacement, velocity, or acceleration) in the span.

Although there are guidelines to assess the severity of vibration, these guidelines are mainly based on operational experience of the plants and they differ from source to source. What makes estimation of the severity of vibration even more difficult is that some of the available guidelines are not stated explicitly but implicitly (Vepsä, 2008).

A summary of the guidelines or standards can be found in the work by Fomin et al. (Fomin, Kostarev, & Reinsch, 2001). According to their work, some guidelines can be used in preventing vibration related problems to occur or when assessing the severity of vibration in existing piping systems, are presented for example in the research work by Gamble and Taggart (Gamble & Tagart, 1991) and in the ASME B&PV section NB-3622.3(ASME B&PV Code-III, 2007).

The focus of this paper was estimating vibration-induced stress in a piping system using measured operational displacement shapes (ODS), a finite element (FE) model of the system and Fluid-Structure Interaction (FSI) analysis. Then followed by determining whether the piping system was safe from fatigue failure by comparing with allowable endurance limit of the piping material. The procedure was illustrated with an example of piping system in which stress caused by the vibration was computed.

Stress calculated by strain gauges might not be the maximum as the measured vibrations on a piping span during a test reflect the conditions at the time of the test. Besides, best locations of strain gauge were usually based on experience, where sometimes the locations were the not the highest stress. This approach gives only indirect information on the loading at the critical locations and generally leads to over conservative assessments. An important advantage of proposed method was complicated foundations involving weak joints on seams, various materials, and fracturing can be readily modelled, thus deciding the highest stress concentration location. Figure below show the comparison of this procedure with normal ASME Procedure.

ASME PROCEDURE	PROPOSED PROCEDURE
Strain gauges located at a specific points but away from the stress concentration region	Accelerometers located at meshed points sufficient to define the motion of the whole structure
Strain are calculated by projecting them to the stress concentrated region	Dynamic characteristic are obtained by operating deflection shape (ODS) analysis which then correlate the finite element (FE) model
Stress values are calculated from strain and by geometrical approximations. Location of the highest stress points is by intuition	Stresses are calculated using fluid-structure interaction (FSI) analysis. FSI determines the maximum stresses; both values and locations

Figure 1 Comparison of ASME procedure and proposed procedure

GOVERNING EQUATION

Interaction of fluid and structure at a mesh interface causes acoustic pressure to exert a force applied to the structure and the structural motions produce an effective "fluid load." The governing finite element matrix equations then become:

$$[M_s]\{\ddot{U}\} + [K_s]\{U\} = \{F_s\} + [R]\{P\} \tag{1}$$
$$[M_f]\{\ddot{P}\} + [K_f]\{P\} = \{F_f\} - \rho_0[R]^T\{\ddot{U}\} \tag{2}$$

$[R]$is a "coupling" matrix that represents the effective surface area associated with each node on the fluid-structure interface (FSI). The coupling matrix $[R]$ also takes into account the direction of the normal vector defined for each pair of coincident fluid and structural element faces that comprises the interface surface. The positive direction of the normal vector is defined to be outward from the fluid mesh and in towards the structure. Both the structural and fluid load quantities that are produced at the fluid-structure interface are functions of unknown nodal degrees of freedom. Placing these unknown "load" quantities on the left hand side of the equations and combining the two equations into a single equation produces the following:

$$\begin{bmatrix} M_s & 0 \\ \rho_0 R^T & M_f \end{bmatrix} \begin{Bmatrix} \ddot{U} \\ \ddot{P} \end{Bmatrix} + \begin{bmatrix} K_S & -R \\ 0 & K_f \end{bmatrix} \begin{Bmatrix} U \\ P \end{Bmatrix} = \begin{Bmatrix} F_s \\ F_f \end{Bmatrix} \tag{3}$$

The foregoing equation implies that nodes on a fluid-structure interface have both displacement and pressure degrees of freedom.

EXAMPLE OF PIPING SYSTEM

The piping system was a Dual Spiral System Pipe, which was pressurized gas transporting pipeline in an offshore platform at Malaysia. This analysis was performed based on 400mmscfd load condition, where the both Flow Control Valves (FCV) were partially opened. The loop is shown in Figure 2 together with its main dimensions. Numbered arrows refer to main components of the system. Legends for these components are listed in Table 1.

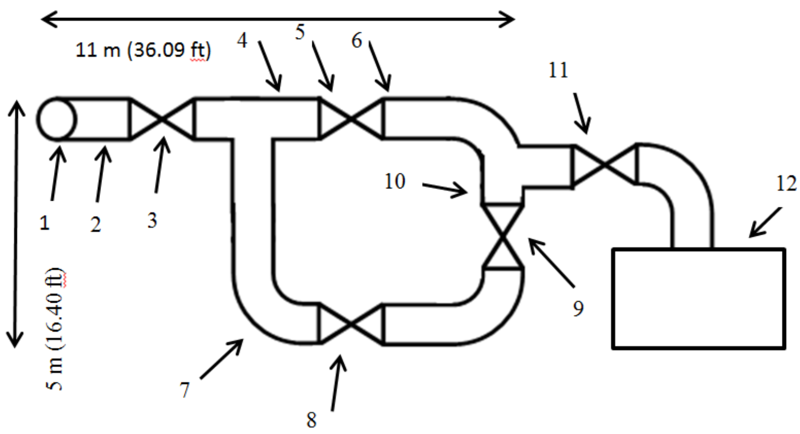


Figure 2Schematic of the Dual Spiral System Pipe

Table 1 The most influential items in the DSS pipe as regards to vibration

Item No.	Description
1	Connection from the main feed line
2	1st Pipe support
3	SDV (Shut Down Valve)
4	2nd Pipe support
5	FCV (Flow Control Valve)
6	3rd Pipe support
7	4th Pipe support
8	Butterfly Valve MV
9	FCV (Flow Control Valve)
10	5th Pipe support
11	Butterfly Valve MV
12	Storage tank

Operating Deflection Shape (ODS) analysis was carried out on-site on the DSS pipe to obtain the deflection pattern of the pipe while in operation. Measurements for the identification of the ODS were carried out in 2010. Measurement points were identified and marked. All the measurement locations were taken using tri-axial accelerometer in 3 principal directions namely X, Y and Z. Main information about these measurements was collected in Table 2. An isometric view showing the sensor locations used in the measurements is shown in Figure 3. Points were linked to obtain a wire-mesh model to represent the structure as show in Figure 2. All the collected data can be put into this model and visualize the vibration movement in the animation.

Measurement was taken by roving accelerometer, where reference point was fixed, and tri-axial accelerometer was switched to select point of record. Location of the reference was pointed by arrows in Figure 3 with the direction of the arrow indicating the direction part of the DOF. A total of 31 points of measurement or total number of DOF of 93 was taken.

Table 2 Main information about the measurement of the ODS

Reference	7Y
Type of measured motion	Translational acceleration
Total number of measured DOF	93
Total number of identified ODS	9
Frequency range of the identified ODS	3.25Hz or 22.5Hz
Average distance between two adjacent sensors	100cm (refer to the drawing)

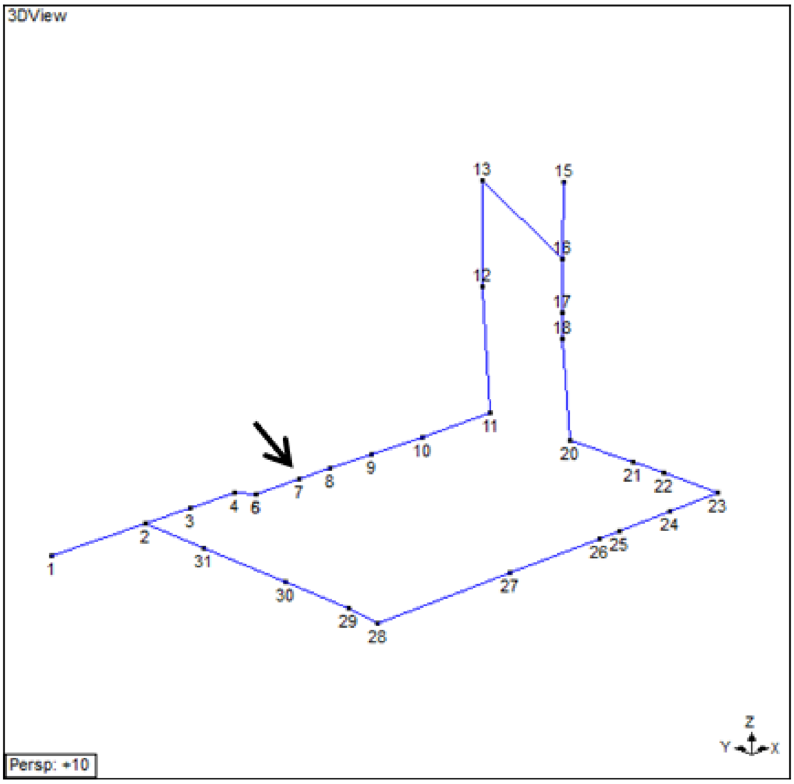


Figure 3 Measurement locations, marked with dots, connected to each other with trace lines. Location of the reference DOF are pointed by arrow, the direction of which indicates the direction part of the DOF.

RESULTS AND DISCUSSION

The flow generated random excitation to the pipe and hence there was no issue of resonance. Resonance only occurred when cyclic excitation hitting the natural frequencies region. Therefore, ODS data was capable to reveal the mode of vibration in this case(Schwarz & Richardson, 1999).

Referring to Figure 4, whenever piping vibration amplitudes at the measured frequencies were greater than the danger line, piping failures occurred. When vibration level was below the design line, very few failures occurred. Therefore, these vibrations versus frequency criteria can serve as a good starting point in evaluating piping vibrations to screen those systems that need further analysis. It can be observed from Figure 4 that the movement was dominated by two frequencies which are 3.60Hz and 4.56Hz. Hence, the 1st and 2nd natural frequencies were found.

Because the mode with the lowest frequency was most easily excited by an external force, only the modes at lower frequencies were of interest in practical piping analysis. It was very difficult for modes at very high frequencies to get excited, thus they were often ignored.

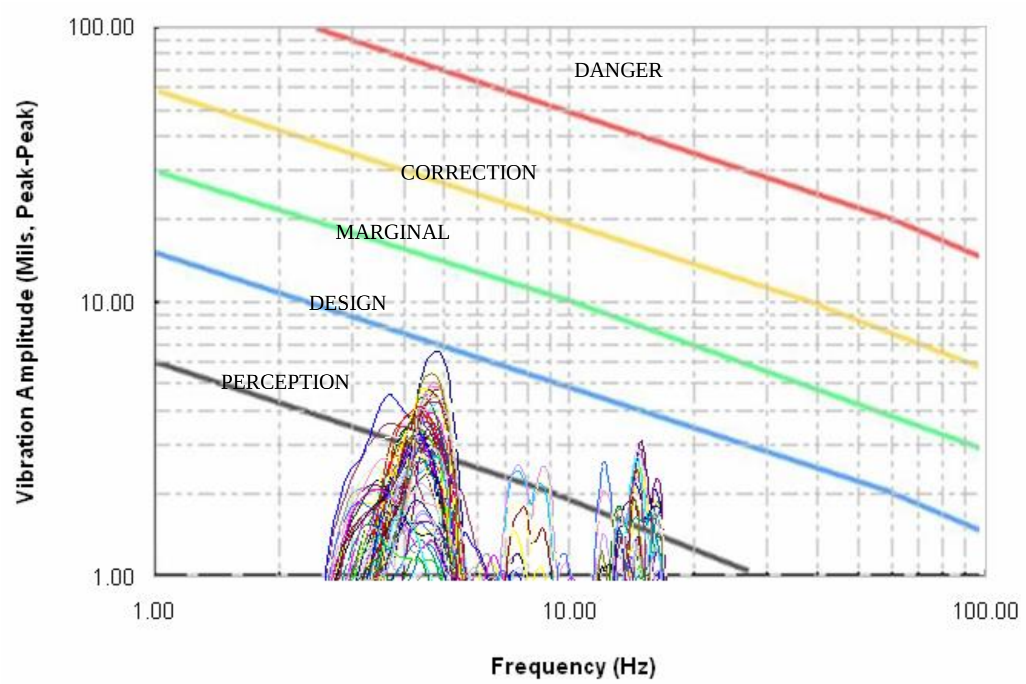
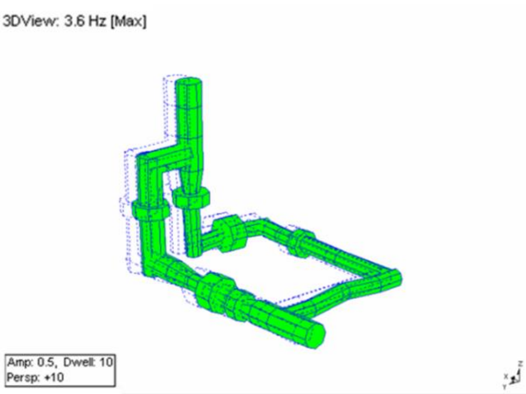


Figure 4 Overlaid ODS spectrums for DSS pipe which plot into the Allowable Piping Vibration Level versus Frequency.

Figure 5 and Figure 6 show the movement of the DSS pipe at 3.60Hz and 4.56Hz. From observation the movements were dominant in Y-direction at 3.60Hz and dominant in X-direction at 4.56Hz.



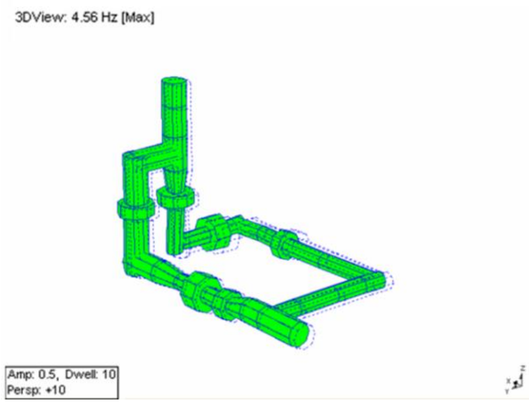
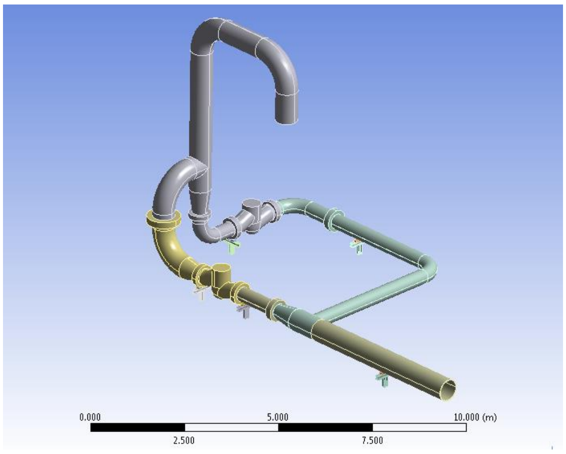


Figure 51st vibration mode at 3.60 Hz   Figure 62nd vibration mode at 4.56 Hz

FE model used in the analyses is shown in Figure 7. Mesh generation is an integral of the analysis process. The mesh quality can have a considerable impact on the computational analysis in term of the quality of the solution and time needed to obtain it. There are 50347 elements and 108190 nodes generated by the Finite Element (FE) software as show in Figure 8.In a higher order 3-D model, 10-node element with quadratic displacement behavior was well suited to modeling irregular meshes. As one boundary condition for the piping system, the connection with the main feed line was considered as rigid. On the other end of the system, the piping system was rigidly connected to the storage tank. The five pipe supports were also modeled in the FE, where the bottom parts of the supports are rigidly connected to the ground.



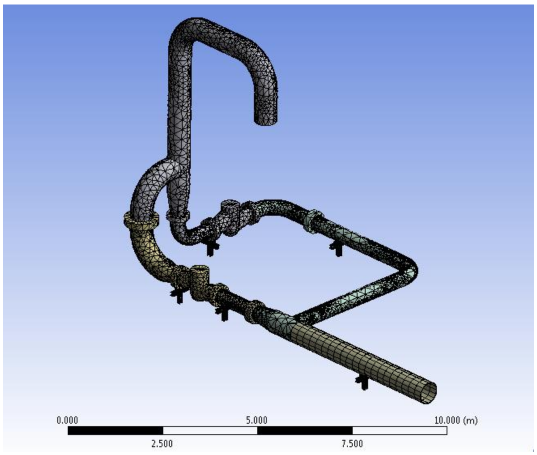
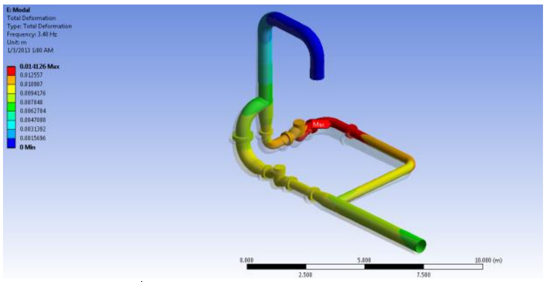


Figure 7 3D CAD Model of DSS Pipe   Figure 8 Mesh Model of DSS Pipe

Thus, the first 2 major vibration modes and frequencieswere obtained from the ODS analysis. It helped in generating a reliable model in FEA by comparing the mode shapes by visual inspection and frequency errors between FEA and ODS. A good and reliable FE model of the piping system was built based on correlation results between FEA and ODS. FE modal analysis revealed the first 2 modes of vibration at 3.48 Hz and 4.62 Hz which were dominated in X-direction and Y-direction respectively (Figure 9 and Figure 10). Therefore, the FE model was well correlated with the measurement data; further analyses such as FSI analysis can be performed.



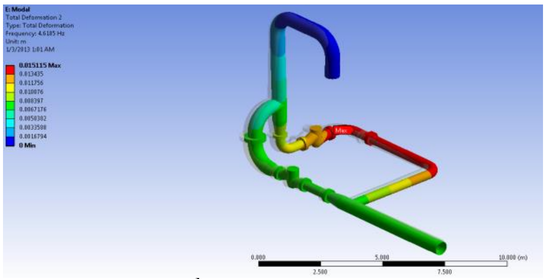


Figure 9 1st vibration mode at 3.48 Hz   Figure 10 2nd vibration mode at 4.62 Hz

Operating parameters of the fluid such as Operating Pressure (OP), Differential Pressure (DP) and flow rate were applied in ANSYS CFX, which coupled with ANSYS mechanical. From FSI analysis, dynamic stress with maximum value of 77.1 MPa and is shown by a red ball in Figure 11.

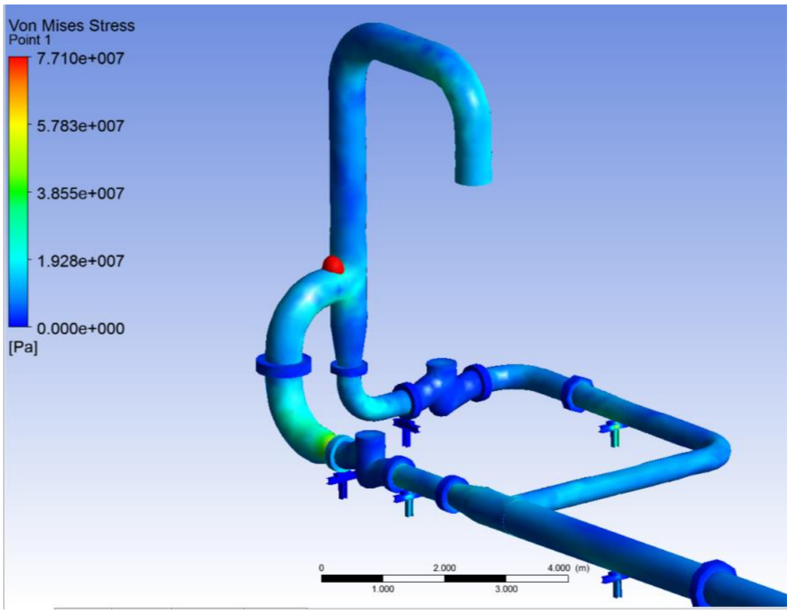


Figure 11 Maximum value and location of Dynamic Stress

Vibration stress evaluation was based on fatigue, specifically high cycle fatigue(HCF). ASME Design fatigue curves [(ASME B&PV Code-III, 2007),(Jaske, 2000)], as shown in Figure 12, were the main tool for routine evaluations.Thedesign curves were A, B, and C, three curves depending mainly on residual stress and the mean stress level. In piping using elastic analysis, only curve B and C were applicable, while curve C was generally used without going through detailed weld stress analyses(Peng & Peng, 2009). For vibration stress, we concerned the allowable endurance strength. For austenitic stainless steels and high alloy steels, the curve *C* endurance strength of 93.8 MPa was used without going through detailed weld stress analysis.From Figure 11, the maximum dynamic stress was 77.1 MPa, which was below the allowable endurance stress limit of 93.8 MPa, hence the pipeline can be said in safe condition.

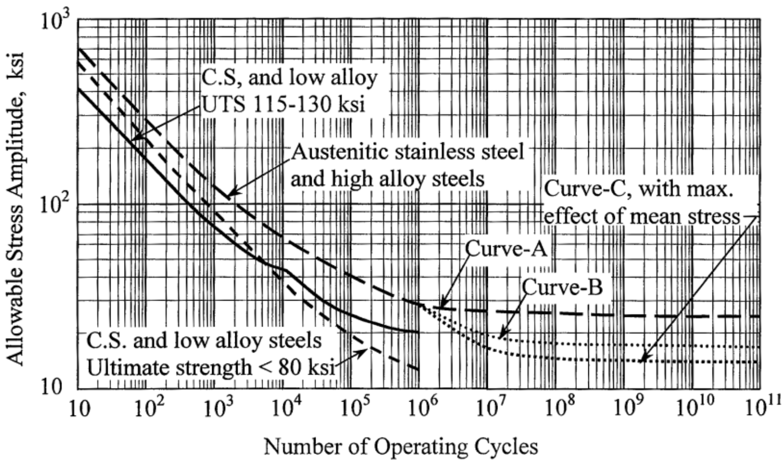


Figure 12 Typical ASME Design Fatigue Curves [(ASME B&PV Code-III, 2007),(Jaske, 2000)]

CONCLUSIONS

By applying the proposed method, vibration-induced stress of pipeline can be estimated without application of strain gauge.The maximum vibration-induced stress was77.1 MPa, which was below the allowable endurance stress limit of 93.8 MPa. Hence, the studied piping system was safe from fatigue failure.

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