Analysis of pipeline fatigue performance under random vibration load

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Abstract. As one of the most important equipments in power plant, the mechanical properties of steam pipe were analysed in this paper. Random vibration theory and Miner linear cumulative damage theory as well as frequency-domain fatigue analysis method were adopted, modal analysis and fatigue performance analysis of the in-service steam pipe were carried out with ANSYS, calculated first ten natural frequencies and cumulative damage under random excitation use Steinberg three-interval method. Results show that no fatigue failure will occur under this excitation load, the random vibration frequency-domain fatigue analysis method can effectively evaluate the fatigue performance of steam pipe, and provide a reference for the pipeline random vibration fatigue analysis.

1. Introduction
In recent years, the safety requirements of power plants are getting higher, and the pipeline vibration control is among those important ones. Long-term vibration will cause fatigue failure of pipeline and support, affect service life of pipe, and threaten the safety of thermal power unit. Therefore, analysis and pipeline vibration control are of guiding significance to the unit safety and pipeline design.

Although random vibration theory is widely used, frequency domain method is rarely used to evaluate the performance of power plant pipeline. This paper introduced the method of random vibration fatigue analysis based on frequency domain method, steam pipeline modal analysis has been carried out, obtained its first ten natural frequencies and modes by ANSYS, assessed PSD response under random vibration and fatigue damage. The results are valuable for pipeline vibration fatigue study.

2. Random vibration analysis theory
Vibration is caused by excitation force, and the vibration of system is a response to that force, the exciting force of pipeline mainly comes from two parts including the pipeline system itself and the outside world. Pipeline vibration is usually classified as pipe resonance, forced vibration, valve self-excited vibration, two-phase medium unsteady flow caused vibration, shock vibration caused by water hammer and random vibration, etc. Most of the fluid in steam pipe of power plant belongs to unsteady flow, and the fluid disturbance is one of the main causes of pipeline vibration [1-3], these are excitation forces from the system itself. In addition, the vibration of steel structure and mechanical equipment, and loads like wind load and seismic load are also the factors causing the vibration of the pipeline, these are excitation forces from the outside world.

The vibration of pipeline can directly cause damage of pipeline and structural attachments, in serious cases, it may lead to fatigue failure and cause unnecessary economic losses. With the
development of new energy, it has become a trend for thermal power units to participate in the deep peak shaving of power grid, which will generate new and more complex pipeline vibration problems. At present, time-domain and frequency-domain analysis methods are the dominating ways to assess random vibration fatigue problem of linear structures [4-6]. In practical engineering, random vibration load is very complicated, time-domain analysis needs to be combined with cycle counting, data calculation is large, while frequency-domain method that based on power spectral density is simple and requires little computation, therefore, the latter is mostly adopted in random vibration analysis [7-10].

2.1. Structural random vibration theory
In order to ensure the dynamic performance of an important structure subjected under dynamic load in engineering, the random vibration analysis of the structure is generally required. Although the response of random vibration can only be studied and described by probability statistics, analysis of random vibration based on finite element method still needs to be combined with structural modal analysis, obtain frequency response function by calculating the response of structure under excitation. The dynamic equation of the structure can be expressed as
\[
[M]\ddot{x} + [C]\dot{x} + [K]x = \{F\}
\]
(1)
Where \([M]\), \([C]\), \([K]\) are the mass matrix, damping matrix and stiffness matrix of the system respectively, \(F\) is the excitation force and \(x\) is the system's response to excitation, the steady state is \(x = A\sin(\omega t - \theta)\).

Power Spectral Density (PSD) function can be used to describe random vibration response of structure, the frequency response function of system is obtained through frequency response analysis, and the system response PSD is obtained by matrix operation between excitation power spectrum and system frequency response function, combined with fatigue damage criterion, the fatigue properties of structure were analysed.

2.2. Frequency-domain fatigue damage
Time-domain analysis is more visual and intuitive, but it is difficult to obtain the ideal random load samples of structure in service life and the data volume is huge, while it is much easier to describe load in frequency-domain way. Time-domain and frequency-domain analysis of random process can be connected by Fourier transform to realize time-frequency conversion. According to Parsaval's equation, the total energy in time domain and frequency domain is equivalent. For the continuous time signal \(x(t)\) and the spectrum function \(X(\omega)\), there is
\[
X(\omega) = \int_{-\infty}^{+\infty} x(t)e^{-j\omega t} dt
\]
(2)
after discretization, there is
\[
X(k) = \sum_{n=0}^{N-1} x(n)e^{-j\frac{2\pi}{N}nk}
\]
(3)
where \(0 \leq k \leq N-1\).

Fatigue failure of structures often occurs in local dangerous and weak locations, for steady-state random processes, the structural stress response power spectral density \(\sigma_{PSD}(\omega)\) is associated with system's frequency response function \(H(\omega)\) and excitation power spectral density \(F_{PSD}(\omega)\).
\[
\sigma_{PSD}(\omega) = H^2(\omega)F_{PSD}(\omega)
\]
(4)
In actual situation, random loads on structural mostly conforms to stable gaussian distribution, and its statistical characteristics do not change with time \(t\), that is, the probability density function of random variable at any time is the same as probability distribution function. Statistical characteristics
and peak expected rate can be obtained through spectral moment of PSD, n-order spectral moment of PSD is

\[ m_n = \int_{-\infty}^{+\infty} f^n G(f) df \tag{5} \]

The Miner [11, 12] linear cumulative damage theory is commonly used to evaluate fatigue life of structures by describing the probability density function of stress amplitude according to spectral moment parameters and combining with cumulative damage theory. The cumulative damage of the structure under multi-stage stress is

\[ D = \sum D_i = \sum \frac{n_i}{N_i} \tag{6} \]

Where \( n_i \) represents the number of cycles when stress level is \( S_i \), and \( N_i \) represents fatigue life at this stress level, while number of cycles within stress range \( (S_i, S_i + \Delta S_i) \) in time \( T \) is

\[ n_i = v T p(S_i) \Delta S_i \tag{7} \]

Where \( v \) represents the number of stress cycles per unit time, and \( p(S) \) is stress amplitude probability density function. Therefore, the cumulative damage in time \( T \) under continuous stress state is

\[ D = v T \int_{0}^{\infty} \frac{p(S)}{N(S)} dS \tag{8} \]

In actual structure, there are many factors such as weld and notch, which lead to the increase of local stress, stress concentration of notch should be considered in calculation, calculated spectral density should be modified with fatigue notch coefficient \( K_f \). The basic flow of fatigue analysis in the frequency-domain of random vibration is shown in Figure 1.

![Figure 1. Random vibration fatigue analysis process.](image)

### Table 1. Pipeline parameter.

| Size     | Material | Circumferential weld coefficient | Calculated temperature /°C | Calculated pressure /MPa |
|----------|----------|---------------------------------|----------------------------|--------------------------|
| \( \phi_{368.3} \times 40 \) | A355P91  | 0.9                             | 541                        | 17.50                    |
| \( \phi_{373.05} \times 30 \) | A355P91  | 0.9                             | 541                        | 17.50                    |
| \( \phi_{610} \times 90 \)  | 12Cr1MoV | 0.9                             | 541                        | 17.50                    |

### 3. Fatigue analysis

#### 3.1. Modal analysis

A 300MW power-generating heating unit in a power plant has been operating for about 40,000 hours, main steam pipe and its supports and hangers are in good condition after macroscopic inspection. The pipeline is selected as study object to analyse its natural frequency and mode, it provides reference for vibration control of pipeline, inspection and adjustment of supports and hangers, and fatigue life
calculation. The geometric dimensions of material parameters, calculated temperature and pressure of the pipeline are shown in Table 1, geometric model of the pipeline is shown in Figure 2.

The stress distribution of the main steam pipe is shown in Figure 3 by static calculation of the pipe. The terminals of the pipe are restrained by displacement, and other numbered nodes are equipped with spring hangers or constant force hangers. As we can see, maximum value of first principal stress is 128.241 MPa at the elbow of first branch pipe at the lower end of the pipe, which is lower than the basic allowable stress of the pipe.

![Figure 2. Main steam pipe model.](image)

![Figure 3. Pipeline stress nephogram.](image)

The first ten natural frequencies and first four modes of the calculated pipelines are shown in Table 2 and Figure 4 respectively, apparently the first few orders of natural frequency are relatively low, in order to avoid resonance, natural frequency should be avoided during operation or a damper should be installed to reduce vibration by changing the stiffness of the pipeline. It can be seen from the
observation of the vibration mode that amplitude of the pipeline is large at the middle and lower elbow, 
so the inspection of pipeline and the strength adjustment of the pipeline weld should be considered in 
particular.

Table 2. First 10 natural frequencies.

| Order | Frequency /Hz | Order | Frequency /Hz |
|-------|---------------|-------|---------------|
| 1     | 0.4046        | 6     | 2.0883        |
| 2     | 0.9685        | 7     | 3.0039        |
| 3     | 1.1747        | 8     | 3.2107        |
| 4     | 1.3480        | 9     | 3.5361        |
| 5     | 1.7463        | 10    | 3.6522        |

Figure 4. The first four modes of the pipeline.

3.2. Spectrum analysis
The frequency response analysis was completed and excitation load spectrum was applied on structure, 
as shown in Table 3.

Table 3. Velocity load spectrum.

| Frequency/Hz | 75   | 78   | 97   | 100  |
|--------------|------|------|------|------|
| Velocity PSD/(m²/s) | 0.37 | 0.52 | 1.3  | 1.0  |
Figure 5. Displacement response spectrum.

Figure 6. Velocity response spectrum.

Figure 7. Acceleration response spectrum.
The PSD of pipeline response to a random vibration is calculated, maximum stress is appeared near point 909 and its response spectrums are shown in Figure 5 to Figure 7. By calculating the stress response PSD data, maximum stress of the response was 5.245MPa. Vibration time is $4 \times 10^4$ hours, according to the P-S-N curve of pipeline material, use Steinberg [13] three-interval method to calculate fatigue strength, for allowable fatigue life of all is near $\infty$ (where $n_1$ is $9.27 \times 10^6$ and $N_1$ is $\infty$), substitute it into damage Formula (8) and get $D=0.153<1$, and it is proved that the pipeline meets fatigue requirements under excitation spectrum and no fatigue failure occurs, which accord with DL/T616 standard requirements. Comparison with other methods, like time-domain analysis method, is very valuable and need to be done in the further researches.

The calculated results show that the measured vibration will not cause fatigue failure of the pipeline, however, long-term vibration may lead to the failure of the pipeline attachment, the deviation and failure of support and hanger, etc. Therefore, it is still suggested to take effective measures to reduce vibration so as to improve the operating safety factor in the actual situation. If conditions allow, appropriate experiments can be used to improve the safety level.

4. Conclusions
(1) In this paper, main steps and procedures of structural random vibration fatigue analysis by frequency-domain method are introduced. Results show that the frequency-domain method is simple, accurate and applicable, and it has broad application value in the analysis of structural random vibration fatigue in engineering.
(2) The first ten natural frequencies of the main steam pipe in a power plant are obtained by means of ANSYS modal analysis, the natural frequency of the pipeline is low, and the tenth order is 3.6522Hz. By comparing vibration modes, it can be seen that the pipe has a large amplitude at middle and lower elbow. This method can also be used as a reference for vibration reduction of pipelines, the inspection and adjustment of supports and hangers, and the design of structures.
(3) It is calculated that no fatigue failure will occur under the excitation load. If conditions permit, appropriate experiments can be carried out, and the method of using multiple groups of measured data combined with time-domain simulation can be closer to actual load spectrum of the pipeline, and the calculation of transient dynamic characteristics of structure can further improve the safety level of fatigue analysis.

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