

Research of Portal Crane Vibration Modal Analysis

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Keywords: Portal crane, Steel material, Vibration, Modal analysis, Safety assessment, Fault diagnosis.

Abstract. Portal crane is an important port production machinery what frequently work in moist environment leads to greater structural damage. The weight of the metal structure is usually 60% to 90% of the whole weight. Steel Q235B material in the early is widely used in the portal crane design and manufacture. For more than a certain number of service life period of portal crane, it is necessary to do safety assessment to reflect their state of health. The traditional safety assessment cannot reflect the structure of the health portal crane vibration state, therefore, vibration testing was added to the portal crane fault diagnosis technology in the present study, which use ambient vibration modal analysis to test a portal crane, and acquired abnormal vibration signals, it is improve the safety assessment system, achieved the good test results.

Introduction

With the continuous development of economic globalization and logistics industry, the role of the port in the modern flow of goods increasingly important, which level of development has become a measure of the main mark level of social development of the country. Port handling machinery is developing towards maximization, continuous, high-speed and automatic, which composition and structure becomes more complex. Portal crane is a major port handling machinery what operations condition is good or bad to impact on productivity and economic efficiency of the port.

Health testing [1] and diagnosis portal crane become an important aspect of crane safety assessment. Current techniques for safety assessment portal crane have visual inspection technique, ultrasonic testing technology [2], magnetic particle inspection techniques, stress and strain detection technology [3]. These techniques can reflect the health of crane metal structure, because the portal crane body of work from frequent braking, speed and load fluctuations, it has a very complex vibration characteristics, and the detection technology can not reflect portal cranes Vibration structural health state [4]. Therefore, it is necessary to add vibration testing technology to the crane fault diagnosis [5]. Structural damage occurs due to various reasons, such as overload, shock, cracks, corrosion, fatigue, manufacturing defects [6, 7], which would led to the physical characteristics of stiffness of the structure, mass, damping changes, and this change is accompanied by the structure dynamic characteristics change [8]. Portal crane can used the vibration test structure for dynamic measurements, to obtain dynamic data, and these dynamic characteristic data as the evaluation of portal crane structure based on health status [9].

Vibration Modal Analysis

Modal analysis [10] is an inverse problem in structural dynamic analysis method, it with the traditional method of direct problem, for example, the finite element analysis, what is established on the basis of the experiment or test, adopts the method of experiments combined with theory to deal with vibration in engineering. Modal analysis is definition of the linear time-invariant system vibration differential equations of physical coordinate transformation for the modal coordinates, that

make the equations coupled and get separate equations of modal coordinates and parameters. Finally gain the modal parameters of system. Transformation of coordinate transformation matrix of the modal matrix, its each column is modal vibration mode. Due to the processing method using the modal truncation, what has greatly reduced number of equations, thus greatly save the computer time, reduced the machine capacity and the computational cost. It is bring great benefits for vibration analysis of the large complex structure. Modal analysis of the goal is to identify the modal parameters of the system that provide the basis for vibration characteristics of structure analysis, fault diagnosis [11], forecasting and dynamic characteristics of the structure optimal design.

Doing the structure model analysis, most of the structure has characteristics of multi degree of freedom system [12]. A typical multi degree of freedom motion equation of linear time-invariant systems in physical coordinate system as follows:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{f(t)\} \quad (1)$$

$$[C] = \alpha[M] + \beta[K] \quad (2)$$

Among them, α and β is the proportionality constant, $[C]$, $[M]$ and $[K]$ are damping matrix, mass matrix and stiffness matrix of structures. Comply damping with the Eq. 2 is proportional damping. Usually the matrix of $[M]$ and $[K]$ are real coefficient matrix, $[M]$ is positive definite matrix, $[K]$ is positive definite for constrained system with no rigid motion, but positive semi-definite for unconstrained system with rigid motion. $[C]$ is symmetric matrix when the damping is proportional damping. $\{f(t)\}$ is the exciting force vector, $\{x\}$, $\{\dot{x}\}$ and $\{\ddot{x}\}$ are the displacement, velocity and acceleration response vector of the structure.

Eq. 1 laplace formula was

$$([M]s^2 + [C]s + [K])\{X(s)\} = F(s) \quad (3)$$

Among them, $F(s)$ and $X(s)$ are laplace transform $\{f(t)\}$ and $\{x(t)\}$.

Then it solves Eq. 1 to obtain the natural frequencies matrix and mode shapes matrix.

$$[\Omega] = \text{diag}[\Omega_1, \Omega_2, \dots, \Omega_N], [\Phi] = [\{\phi_1\}, \{\phi_2\}, \dots, \{\phi_N\}] \quad (4)$$

$[\Phi]$ is modal matrix consist of each order structural vibration vector $\{\phi_i\} (i=1, 2, \dots, N)$. In the modal coordinates system, $[\Phi]$ as the base vector matrix space coordinate system, suppose

$$\{x\} = [\Phi]\{q\} \quad (5)$$

$\{q\}$ is modal coordinate vector.

Suppose $\{f\} = \{F\}e^{j\omega t}$, $\{q\} = \{Q\}e^{j\omega t}$, $\{x\} = \{X\}e^{j\omega t}$, frequency response prediction formula is:

$$\{X\} = [H]\{f\} \quad (6)$$

$$[H] = [\Phi][Y_r][\Phi]^T \quad (7)$$

$$[Y_r] = \text{diag}[Y_1, Y_2, \dots, Y_N] \quad (8)$$

$$Y_r = (k_r - \omega^2 m_r + j\omega c_r)^{-1} \quad (9)$$

Conversion from the physical model to the modal model, it means a process of the force balance equation becomes energy balance equation in a physical.

Any row of frequency response function matrix is

$$[H_{i1} \quad H_{i2} \quad \cdots \quad H_{iN}] = \sum_{r=1}^N \frac{\varphi_{ir}}{k_r - \omega^2 m_r + j\omega c_r} [\varphi_{1r} \quad \varphi_{2r} \quad \cdots \quad \varphi_{Nr}]$$

Obviously, any line of $[H]$ that contains all the modal parameters, and the ratio of this row and r-order modal frequency response function values, namely the r-order mode shape.

Modal analysis is divided into two categories, the one operate equipment to test structure dynamic response under known excitation and according to the theory of structural dynamics to identify modal parameters of the structure, this method is called experimental modal analysis. Another use the finite element software to establish structure finite element model, then obtained the modal parameters, what is called theory modal analysis. According to using special excitation equipment or not, portal crane experimental modal analysis can be divided into two categories: the traditional experimental modal analysis and environmental incentive experimental modal analysis. Traditional experimental modal analysis is the use of special excitation equipment, test structure response under the excitation device, and use the excitation and response to solve the structure of the frequency response function. Then determine the structure modal parameters. According to different points and the number of measuring points of structural response, the traditional modal analysis is divided into a single point of single point output, a single point of incentives more output and more incentive more output. Environmental incentive mode analysis refers to do not use special excitation equipment, that test load response of portal crane structure, only using the output response identifies modal parameters. Due to environmental incentive mode analysis is the parameter identification method by only using the output response, so was named work modal analysis. Because it is not operate special excitation equipment as incentive and use normal working load, then reduce the complexity of the test system. At present, work modal analysis become mainstream method in the portal crane.

Material

Select a model of MQ10-30 portal crane as the test object, this crane rated load is 10t, amplitude is 15m to 30m, lifting height of rail surface is 25m, under the rail surface is 15m, work level is A6, work year is 25 years. The main metal structure of this portal crane is steel Q235B, what is a kind of carbon structural steel materials. It is have suitable carbon content, and have excellent comprehensive performance, thus its strength, plasticity and welding workability can get better satisfied. This kind of material in the early is widely used in the portal crane design and manufacture.

In order to obtain the mechanical properties of the steel Q235B, what was selected to material performance test, and results follows, tensile strength of tensile properties is 385MPa, yield strength is 235Mpa, Young's modulus is 208GPa.

Analysis

The use of environmental incentives modal analysis system is TMR-211 and acceleration sensor is AS-1GB. In four key position to put four acceleration sensor, which correction coefficients were 0.007644m/s^2 , 0.007632m/s^2 , 0.007864m/s^2 , 0.007674m/s^2 . In the center of the turntable, facing the cab direction is positive, the four-point position on the right side in front of the box girder, left side in front of the box girder, right side of the rear box girder, left side of the rear box girder were named measure points 1,2,3,4. Test conditions had two, one jib starting position is westward parallel tracks, the minimum amplitude state and unloaded, portal crane operation is running at maximum luffing to rise spreader system move twice jib began to turn north 90° , walking carts, jib to a minimum at the start, westward turn 90° . Another jib starting position is westward parallel tracks,

minimum amplitude and load 11t, portal crane lifting load, increased brake load twice the maximum luffing to decrease brake three times, the jib swing northward 45° , close to the minimum width, load dropping brake one, westward swing arm 45° , then load floor.

Under the working condition of the above two tests of four measuring point vibration response results are shown in Table 1, the vibration signal of the sequential environment as shown in Fig. 1-8.

The Table 1 shows that vibration attenuation characteristic of the turntable in the vertical direction is good. The main frequency of point 1,2,3,4 under first working condition were 28.9Hz, 11.3Hz, 5.7Hz and 34.7Hz. Their under second working conditions were 28.9Hz, 12.0Hz, 5.7 Hz and 35.4Hz.

Table 1. Each measuring point vibration response under the condition of test results.

| measure point | characteristic parameter | first working condition | second working condition |
|---------------|---------------------------|-------------------------|--------------------------|
| 1 | acceleration amplitude[g] | 0.61 | 0.54 |
| | signal frequency[Hz] | 28.9 | 28.9 |
| 2 | acceleration amplitude[g] | 0.17 | 0.05 |
| | signal frequency[Hz] | 11.3 | 12.0 |
| 3 | acceleration amplitude[g] | 0.08 | 0.02 |
| | signal frequency[Hz] | 5.7 | 5.7 |
| 4 | acceleration amplitude[g] | 0.03 | 0.03 |
| | signal frequency[Hz] | 34.7 | 35.4 |

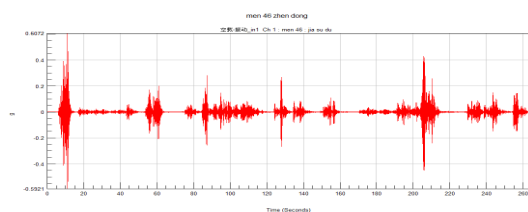


Fig. 1 Vibration data signal of point 1 under first work condition.

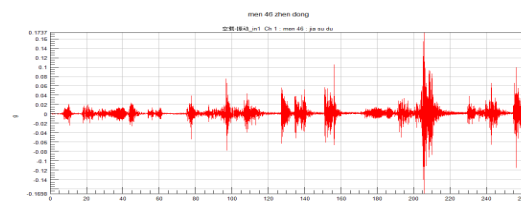


Fig. 2 Vibration data signal of point 2 under first work condition.

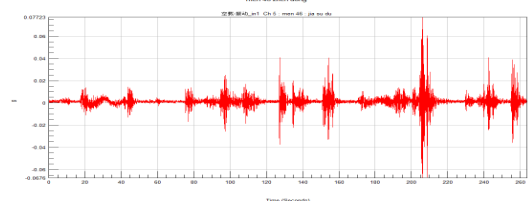


Fig. 3 Vibration data signal of point 3 under first work condition.

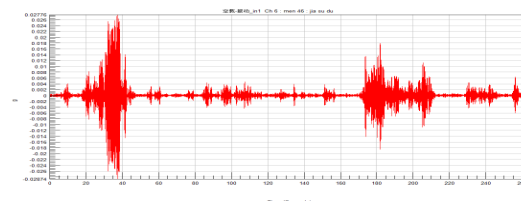


Fig. 4 Vibration data signal of point 4 under first work condition.

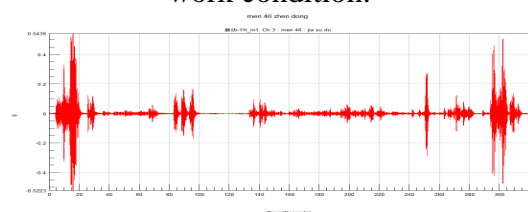


Fig. 5 Vibration data signal of point 1 under second work condition.

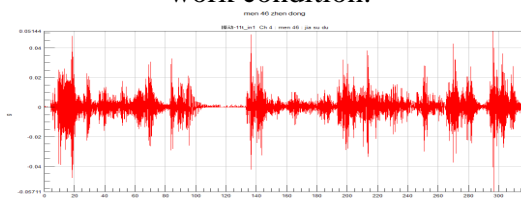


Fig. 6 Vibration data signal of point 2 under second work condition.

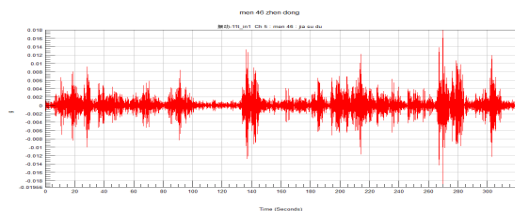


Fig. 7 Vibration data signal of point 3 under second work condition.

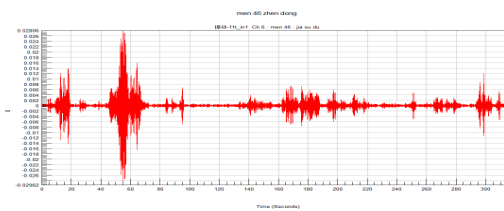


Fig. 8 Vibration data signal of point 4 under second work condition.

The acceleration amplitude of point 1,2,3,4 under first working condition were 0.61g, 0.17g, 0.08g, and 0.03g; Their under second working conditions were 0.54g, 0.05g, 0.02g and 0.03g.

The measured vibration data under all condition acceleration amplitude of point 1 not less than 0.2g that over the correlative standards about portal crane. The portal crane was found in a large four-link structure connecting pins were badly worn so that the whole space activities, what cause the whole machine structure vibration amplitude exceeds bid badly. The crane structure surface have different degrees of corrosion, such as gantry frame leg balance beam, gantry frame circle weld and turntable box girder had crack. It is caused vibration amplitude is bigger under unload work condition by material degradation and fatigue damage.

From Fig. 1 to 8, the portal crane is matched with the corresponding working condition, turntable vibration magnitude under unload conditions is greater than rated load by acceleration and vibration damping characteristics. The front of turntable vibration magnitude is greater than the back in 11t load.

Conclusion

- 1) Vibration signal was match actual operation by used the environmental incentive experimental modal analysis.
- 2) There was a measuring point of the vibration amplitude seriously overweight under unload conditions by analyzing the collected vibration data, And through further testing to identify the crane had wear and tear, defects cracks, these interaction of defects lead to excessive vibration, what indicated the vibration characteristics can reflect the health status of the portal crane.
- 3) The vibration modal analysis techniques as a supplement to conventional portal crane safety assessment, that can improve the evaluation system, and found that conventional technology cannot be found fault. From the macro to the security status of the crane early warning.

Acknowledgements

This work was financially supported by Technology Projects of Guangdong Provincial Quality and Technical Supervision (2015CT02、2015CT09), Guangzhou Innovation Platform and Sharing Special Projects (201509010008), Science and Technology Planning Project of Guangdong Province (2014a040401046).

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