

Modal Analysis of Temporarily Installed Suspended Access Equipment Based on ANSYS

Xijian Zheng^{1,a}, Zhen Lu^{1,b}, Xiangyun Zhao^{1,c}, Xingliang Gao^{1,d}

¹School of transportation & Mechanical Engineering, Shenyang Jianzhu University, Shenyang 110168, China

^axijianzheng@sina.com, ^bluzhen1983@126.com,
^czhaoxiangyun6@126.com, ^dxinglianggao@163.com

Keywords: Temporarily Installed Suspended Access Equipment, structure modal analysis theory, FEM, natural frequency

Abstract. The paper chooses ZLP800 temporarily installed suspended access equipment (TISAE) as a object of study, it is based on theory of structural model analysis. The finite element model has been established by the finite element method (FEM). The model is analyzed by the modal analysis module of ANSYS, the first four natural frequencies and corresponding modal shapes are extracted in this paper, the cause of every modal shapes are analyzed in the paper. In order to get a more rational constraint mode, the natural frequency under different constraint condition is calculated. Then the external excitation factors, such as the vibration frequency of the electrical machine in a hoister, swing of suspended platform and the friction disk in a hoister are calculated respectively, finding the external excitation factors which would cause a sympathetic vibration, and giving the suggestions for averting the sympathetic vibration, providing some academic reference to improve and perfect the structure design of the TISAE products.

Introduction

TISAE plays an important role in modern high-rise building construction, its working conditions are complex and volatile under actual construction, it usually bears dynamic load, some vibrations will come into being from its structural system, which would not only caused fatigue breakdown of structure, but also made it possible to cause sympathetic vibration when these vibration frequencies are equal to the outside driving frequency. In order to avoid the happening of sympathetic vibration, make the theory and method on design of TISAE more rational, and enhance its structural reliability as far as possible, especially in fatigue reliability, it becomes very important to analyze the structure dynamic characteristics of TISAE.

Structure modal analysis theory

In the course of mechanism design, in order to avert a sympathetic vibration, the problem of elasticity vibration is researched by us, the concrete mechanism is considered as a multi- freedom of motion vibration system, which contains multiple natural frequency, and shows multiple sympathetic vibration areas in the impedance test, the basic vibration characteristic of the structure in free vibration is called modal of the structure. The structure modal is conditioned by the its own characteristics of the structure and material, and is not related to the external loads.

A linear system with N freedom of motion [1], its movement differential equation is written as:

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

[M] is the mass matrix of the TISAE system, [C] is the damping matrix of the TISAE system, [K] is the rigidity matrix of the TISAE system. $\{x\}$ is displacement response vector, $\{F(t)\}$ is impel force vector of TISAE structure. [M], [K] are real coefficient symmetrical matrix, and [C] is nonsymmetrical matrix, therefore the Eq. 1 is a set of coupling equation, when there is a large freedom of motion in the system, it will become difficult to solve the equation, it is the main problem solving by the modal analysis, that whether the coupling equation above could transform into uncoupling independent equations set.

If there is not action of external force, namely $\{F(t)\} = \{0\}$, and the damping [C] is omitted, the non-damping free vibration equation is expressed as:

$$[M]\{\ddot{x}\} + [K]\{x\} = \{0\} \quad (2)$$

The corresponding equation is written as:

$$([K] - \omega^2[M])\{x\} = \{0\} \quad (3)$$

where, ω is natural frequency of the TISAE system, Hz. The natural frequency and modal shapes could be got from the Eq. 3. ω is a main dynamic characteristic parameter which is obtained from the modal analysis.

Modal analysis of TISAE

Make use of the finite element analysis software——ANSYS, the finite element model of the TISAE is established in the paper, the model contains 15895 nodes, the number of freedom of motion corresponding is very large, so it is very difficult to solving the whole natural frequencies and modal shapes [2]. According to the vibration theory, it is the low order natural frequency and the corresponding modal shape that make the major actions, but the high order natural frequency and the corresponding modal shape make less influence to the structural vibration, and a certain damping existed in the actual structure makes a quick attenuation to the corresponding modal shape of high order natural frequency, in other words, the dynamic characteristic of the system is decided by the low order frequency. In the paper, it can be discovered that the modal shape which has a strong impact on the structure focuses on the first four orders, therefore the first four natural frequencies and corresponding modal shapes are extracted in this paper, the first four natural frequencies are shown in table 1, and the first four modal shapes is shown in Fig. 1- Fig. 4.

Table 1 Natural frequency of the TISAE

order	Frequency(Hz)
1	0.26018
2	5.3315
3	8.9569
4	9.3014

The Solutions of Modal Analysis

The first four orders natural frequencies of the complete TISAE are obtained, as shown in table 1. What is clear from the datas is that the whole structure may have a great vibration when excitation frequency generated by external factors is close to the results of modal analysis. Under certain conditions it is likely to cause damage to the equipment. To avoid this pitfall and reduce the vibration caused by these factors, we can modify the structural design to improve the vibration frequency of the whole machine, thus improving operating comfort.

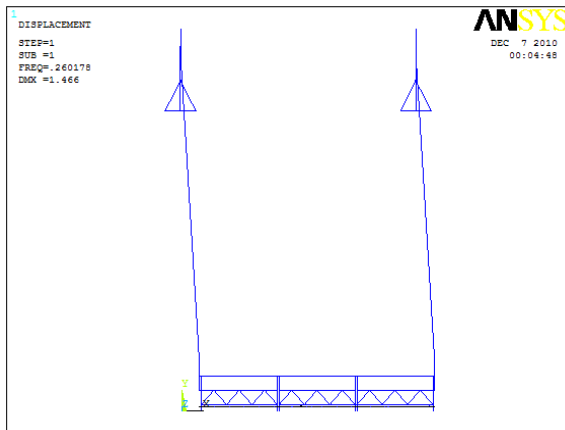


Fig.1. The first modal shape of the mechanism

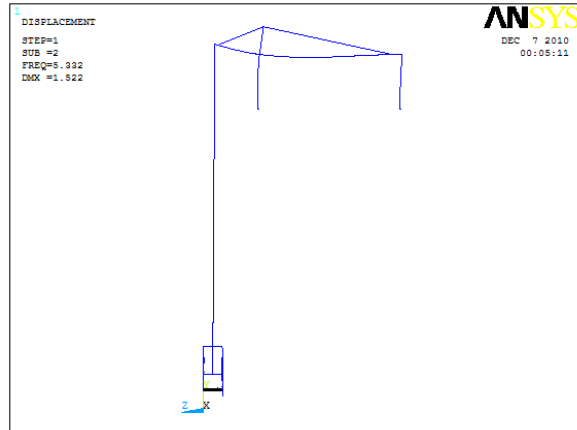


Fig. 2. The second modal shape of the mechanism

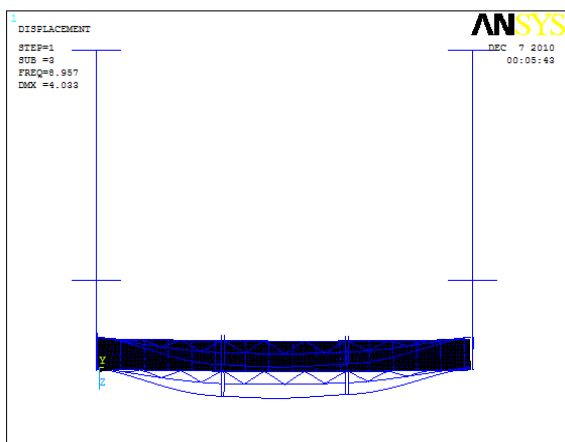


Fig. 3. The third modal shape of the mechanism

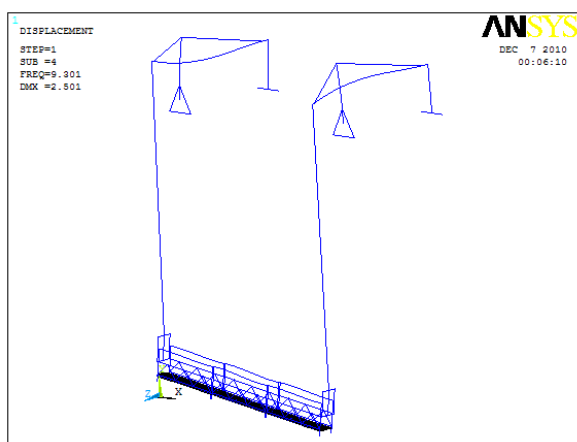


Fig. 4. The fourth modal shape of the mechanism

The lateral horizontal vibration of the complete TISAE, which is caused by the horizontal wind load, the movement of operators within suspension platform and so on, can be seen from the first order vibration mode figure. The second one is vertical vibration of the whole machine in the vertical direction, which results from the suspension platform hoisting from ground suddenly, a sudden braking and unloading during the rising in the air and so on. The third one is the horizontal vibration of the fence before and after suspension platform, which is aroused by the horizontal wind load. The fourth one is the upper and lower vibration of cantilever beam of suspension mechanism and inconsistent direction of vibration, which may be due to load bias within the suspension platform.

Table 2 The natural frequencies of the system in different constraints

order	The natural frequency in constraint 1 (Hz)	The natural frequency in constraint 2 (Hz)
1	0.26018	0.26015
2	5.3315	5.1715
3	8.9569	6.6675
4	9.3014	6.9647

Different constraints on the natural frequencies of the system

Constraint 1: all DOF (degrees of freedom) of suspension mechanism UX, UY, UZ, ROTX, ROTY, ROTZ, and line displacement of suspension platform UZ.

Constraint 2: three line displacements of suspension mechanism UX, UY, UZ, and line displacement of suspension platform UZ.

As can be seen from Table 2, the main conclusions are as follows:

1) Under the constraint 1, the value of the first four orders natural frequencies of the complete SAE is greater compared with the constraint 2. The greater value of the low natural frequency is the more difficult for the incentive stir-up. When the system is a low vibration one, vibration damping time may be reduced, which is conducive to stable operation of the whole system.

2) In actual working conditions, we can set reasonable constraints on the structure. The change of constraints has a significant impact on the natural frequencies and mode shapes of construction platform system. Therefore, from the actual situation, it is more reasonable to adopt the constraint 1.

Calculation of the external excitation operating frequency

The vibration of SAE is caused by many factors, such as the motor vibration of the hoist machine, the vibration of the transmission, the deflection vibration of the suspension platform, a sudden starting and braking vibration and so on. These factors will cause structural vibration, and even resonance, which could severely damage the structure, and even lead to casualties. Therefore, we must try to avoid the sympathetic vibration [3]. We would focus on the calculation of the three main frequencies, as follows:

Calculation of the vibration frequency of the motor. According to vibration theory, low natural frequency has a greater impact on dynamic response of the system [4]. By contrast, high order resonance has less damage to the system, namely, dynamic response of the system is less affected by high natural frequency. In the paper, the motor of the hoisting mechanism of ZLP800 SAE is the electromagnetic brake three-phase asynchronous motor, of which the model is YEJ100LA-4. Motor poles are four. Rated speed is 1420r/min. The relation of its operating frequency and the speed

is $n = \frac{60f}{p}$, where, n is Motor speed, f is operating frequency of the motor, p is Motor poles. It can

be seen from the above equation, it will reach the maximum operating frequency, which is about 47Hz, and far greater than low natural frequency of the system when the motor reaches rated speed. Therefore, it is unlikely to structural resonance caused by the motor vibration.

Calculation of the deflection vibration frequency of suspension platform. It may cause the platform around the suspension point pendulum-type swing, and lead to a deflection vibration of the whole structure when the suspension platform withstanding wind loads, the movement of staff or load bias within the suspension platform and so on[5]. In the paper, the length of hoisting cable of the established model is set to 10m. The deflection vibration frequency of the suspension platform which can be calculated with Eq. 4 is written as

$$f = \frac{1}{2\pi} \sqrt{\frac{g}{l}} = \frac{1}{2\pi} \sqrt{\frac{9.8}{10}} = 0.158 \text{ Hz} \quad (4)$$

And it can be seen from (4) that the greater the length of hoisting cable, the smaller the deflection vibration frequency.

Calculation of the operating frequency of the rope sheave of the hoist. The hoisting mechanism of SAE is built based on the principle of friction gear, and driving force transfers torque to the wire rope around the wheel through the first level of worm drive and the second level of gear drive. Accessing to design account book of SAE, we can see hoisting cable around the sheave pitch diameter $D = 210.6\text{mm}$, hoisting speed $v = 8.03\text{m/min}$, the speed of the sheave calculated with Eq. 5 is written as

$$\omega = \frac{v}{\pi D} = \frac{8.03}{0.2106\pi} = 12.137 \text{ r/min} \quad (5)$$

Then the operating frequency which could be calculated is $f=0.202\text{Hz}$.

Conclusion

In this paper, the modal analysis is carried out to the mechanism system of the TISAE, the conclusion are as follows:

The first four orders natural frequencies of the complete SAE are obtained by the modal analysis to the structure, the vibration frequency of the motor, the deflection vibration frequency of suspension platform, and the operating frequency of the rope sheave of the hoist are calculated, it could be known that the working frequency of the rope sheave is approximate to the first order natural frequency, in some specific conditions, it is impossible that the working frequency of the rope sheave is equal to the first order natural frequency, and then causing the structure sympathetic vibration. Consequently, it should be as far as possible to increase the rigidity when designing the structure of the TISAE, and then increase the low orders natural frequency, and enhance its anti-interference ability.

The first and second order modal shapes of TISAE separately react the horizontal vibration and the vertical vibration of the structure, according to the theory of vibration shapes supraposition, once the master vibration shapes is aroused, the life of the suspended platform would be badly influenced, and the installation and operation will become very inconvenient. Therefore, the obstruction force in the direction of master modal should be reduced as far as possible, the measurement of the vibration extent is influenced by the suddenly hoisting from the ground, suddenly brake in the air, and suddenly uninstall etc.

Acknowledgements

The research in this paper has been supported by National "Eleventh Five-Year" Key Scientific and Technological Plan, 2008BAJ09B02-2. Shenyang science and technology projects, F11-030-2-00.

References

- [1] Cao Shuqian. The Modal Analysis theory, test and application of vibration structure. Tianjin University Press, 2001.
- [2] Zheng Yuqiao, Huang Jianlong, Zhang Shuzhen, Finite Element Modal Analysis of Structure of Gantry Crane. The Mine Machine. 17 (2010) 68-70.
- [3] Liu Zhuli, He Peipai, Wang Shuangxi, FEA Analysis of Gantry Crane. Water Conservancy & Electric Power Machinery. 8 (2006) 23-26.
- [4] Sha Daliang, Modal Analysis and Comparison of Flat-top Tower Cranes. Construction and Mechanization. 8 (2008) 31-33.
- [5] Yang Huiyong, Yin Chenbo, Yao Keheng. The Static and Modal Analysis of Tower Crane's Jib Based on ANSYS. Construction and Mechanization. 4 (2010) 53-56.