Chapter 16
Modal Analysis of a Two Mass Feeder

Qingrui Meng, Chuwen Guo, Jie Quan and Zhipeng Hu

Abstract Conveying velocity of a two mass feeder is adjusted by using a frequency converter according to practical requirements. But the trough mass and the exciter are often damaged because of resonance. To resolve this problem, modal characteristics of a two mass feeder are analyzed in this paper. At first, the FEM model of the trough mass is built and analyzed by using ANSYS; correctness of the simulation results is proved by modal tests. Then modal characteristics of the two mass feeders are analyzed, the natural frequencies and the mode shapes are obtained. The results show that the fifth and seventh frequencies are closer to the working frequency and may cause large deformation of the trough mass and the exciter, which are the main reasons for the damage of the two mass feeder. Analysis results of this paper provide theoretical basis for correct use and further improvement of the two mass feeders.

Keywords Two mass feeder · Modal characteristics · ANSYS · Modal tests

Q. Meng (✉)
School of Mechanics and Civil Engineering, China University of Mining and Technology, Xuzhou, China
e-mail: 516512636@qq.com

C. Guo
School of Electric Power Engineering, China University of Mining and Technology, Xuzhou, China
e-mail: spiderman22@126.com

J. Quan · Z. Hu
School of Mechanical and Electrical Engineering, China University of Mining and Technology, Xuzhou, China
e-mail: cumtquanj@126.com

Z. Hu
e-mail: 962989346@qq.com

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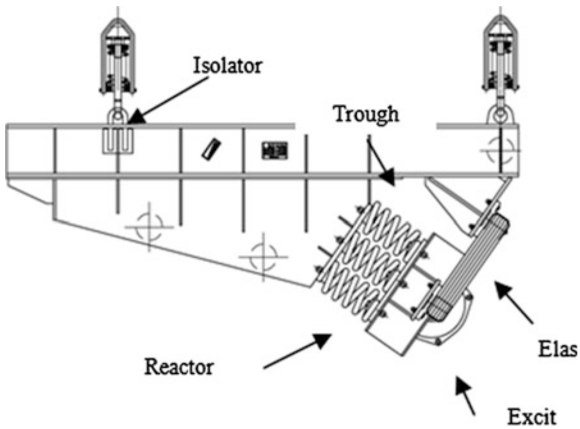
16.1 Introduction

Inertial vibratory feeders are widely used in many industry fields for bulk solids handling. However, because of complexity of its mechanical structure and limited capability, there exist various impending problems such as severe noise, low efficiency, fracture of the steel plate, and so forth. Many research works have been carried out to solve these problems [1, 2]. But the problems still exist and are disadvantage factors in many cases.

A better choice is to use a novel inertial vibratory feeder—two mass feeder (refer to Fig. 16.1), which comprises trough mass, exciter, reactor springs, and isolator springs. The unbalance rotating masses are installed in exciter. A two mass feeder adopts the principle of near-resonant inertial vibration, its exciting force and power is highly reduced compared with non-resonant vibrating machines. With the advantages of high conveying velocity, low noise, small power consumption, and long service life, it has a wide application prospect. Many research works on its characteristics have been carried out. Koizumi et al. presented vibration characteristics with full consideration of transmissive force and the optimum design procedure of a vibration feeder [3]. Gerstel and Scheublin developed a computer simulation to facilitate direct and fast calculation of the forward speed of the particles on the vibrating deck in relation to the design parameters [4]. On the basis of calculating the system response, the required unbalance and the transmitted force of a two mass vibratory feeder, Pramanik discussed a method for selection of the feeder's spring stiffness [5].

Conveying velocity of a two mass feeder is often adjusted by using a frequency converter according to practical requirements. But weld cracking or fracture of the steel plate are often caused by resonance. To resolve these problems, in this work, a system model is established using a finite element method (FEM). By adopting commercial software package ANSYS, modal characteristics of the trough mass of the two mass feeders are analyzed and verified by experiments.

Fig. 16.1 Two mass feeder



16.2 Modal Analysis of the Trough Mass

For the sake of avoiding producing too much finite elements, which will increase computing time, reduce grid quality, and analyze precision, certain simplifications of analysis model of the trough mass have been made, including bolt holes, chamfers, fillets, and so forth. The analysis model built in ANSYS as shown in Fig. 16.2.

Material and related parameters of the trough mass are shown in Table 16.1 Mesh generation of the trough mass is shown in Fig. 16.3.

The rigid mode should be removed from solutions of the free mode, thus the first 10 free modes can be calculated. Since the working frequency is 16.25 Hz, we just study the first 5 mode shapes obtained from the FEM analysis as shown in Fig. 16.4.

From Fig. 16.4 it can be seen that the first mode presents a twist motion of the trough mass with respect to the z-axis, and the deformation is quite large. It indicates that the trough mass will become asymmetric and the track motion will be seriously affected when resonance occurs. The second mode presents rotational motion of both the junction panels in opposite direction with respect to x-axis, while the third mode presents rotational motion of the junction panels in the same direction with respect to x-axis. The fourth mode indicates bending motion of both the side panels along x-axis; it has the same effect as the first mode on the trough mass when resonance occurs. The fifth mode indicates bending motion along z-axis.

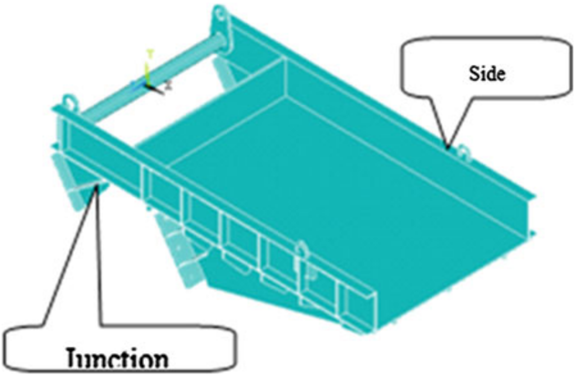


Fig. 16.2 Trough mass

Table 16.1 Related parameters

Item	Value
Material	Q235
Elastic modulus(GPa)	206
Poisson ratio	0.3
Density(Kg/m^3)	7,850
Element type	Solid95

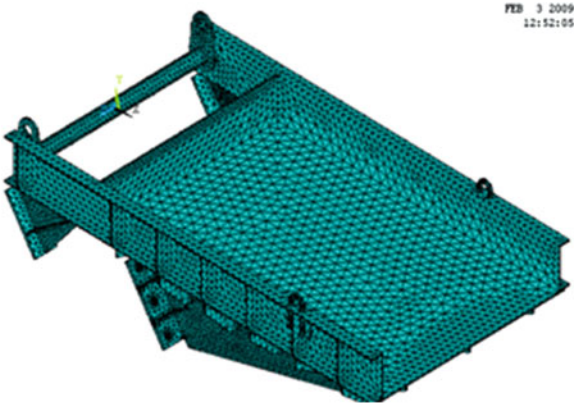


Fig. 16.3 Mesh generation of trough mass

From Table 16.2 it can be seen that, among the first 5 orders of natural frequency, each has a large difference with the adjacent ones. It indicates that frequency of external interference can only be close to one of them and the superposition of vibration of the trough mass can be avoided.

The first modal frequency (23.205 Hz) is greater than the working frequency (16.25 Hz) about 43 %, which is higher than 20 %. It proves the rationality of the structural design of the trough mass.

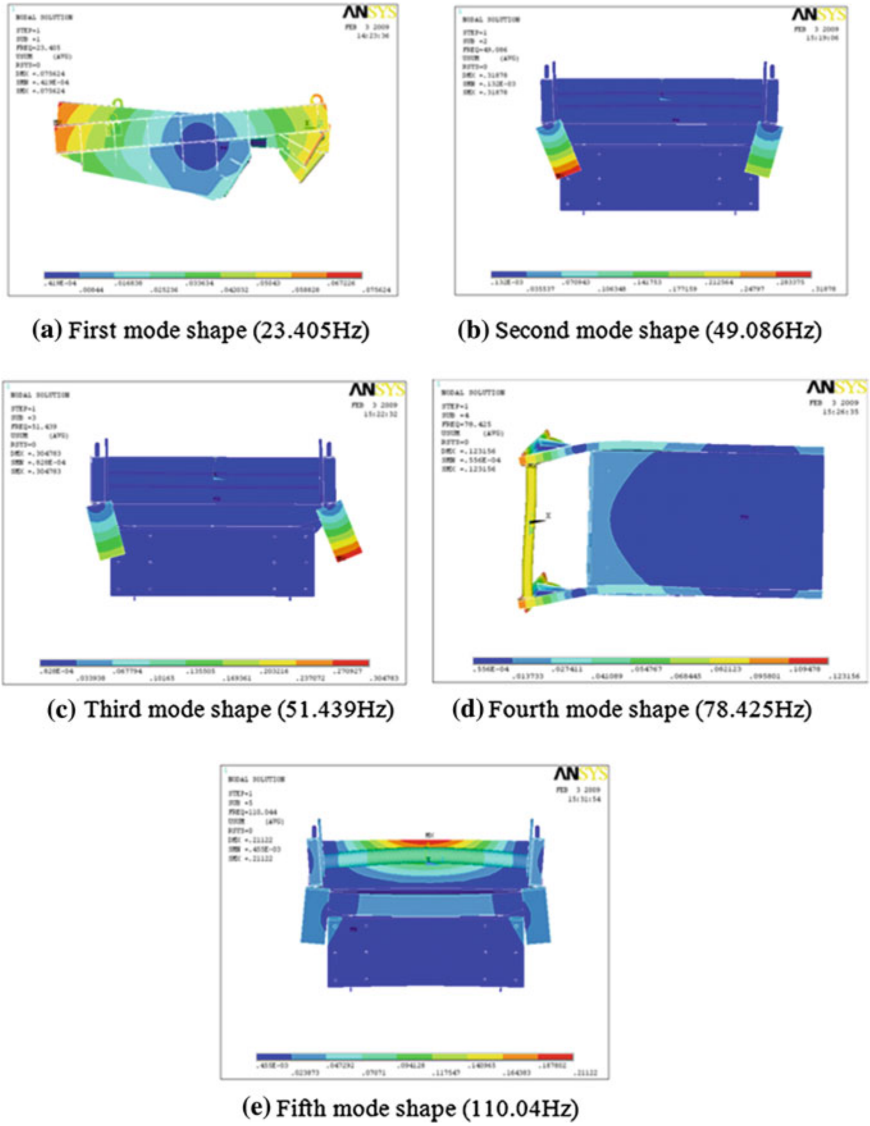
16.3 Modal tests

In order to verify correctness of the simulation results, a trough mass is experimentally measured. The testing instruments used in this test include a PCB excitation hammer, an accelerometer of type KD1005, a charge amplifier of type uT41C3, a signal analyzer of type uT3208F, and a PC. The diagram of the testing system is shown in Fig. 16.5.

The state of the trough mass is of utmost importance to experimental results. Both a free state and a fixed state are used at present. By comparison, the intrinsic attributes of the trough mass can be experimentally obtained by using a free state. So a free state is adopted in this work.

The following requirements are considered in selection of the location and the quantity of the measuring points and the measuring direction.

(1) The exciting point should be away from the node of any of the mode shapes, so as to ensure that the acquired signals have a higher signal-to-noise ratio. (2) The arrangement of the measuring points should represent the geometry characteristic of the structure. (3) Deformation features of each mode can be distinguished clearly. (4) The measuring points should include all of the key structural points.



(a) First mode shape (23.405Hz)

(b) Second mode shape (49.086Hz)

(c) Third mode shape (51.439Hz)

(d) Fourth mode shape (78.425Hz)

(e) Fifth mode shape (110.04Hz)

Fig. 16.4 Simulated mode shapes of the trough mass

The arrangement of the measuring points is shown in Fig. 16.6. In order to compare with the FEM analysis results, only the first 10 orders of natural frequency of experimental results are listed, as shown in Table 16.2.

From Table 16.2 it can be seen that relative errors of 8 orders of natural frequency are less than 10 %, and the maximal relative error is 20.9 %. The first 5 orders of measured mode shapes agree with the FEM analysis results quite well.

Table 16.2 Comparison of natural frequency

Order	1	2	3	4	5
Theoretical value/Hz	23.405	49.086	51.439	78.425	110.04
Measured value/Hz	27.5	50.0	65.0	83.0	103.0
Relative error	14.8 %	1.8 %	20.9 %	5.5 %	−6.8 %
Order	6	7	8	9	10
Theoretical value/Hz	117.72	124.82	144.95	151.78	155.63
Measured value/Hz	113.3	129.0	144.3	147.5	151.3
Relative error	−3.9 %	3.2 %	0.4 %	2.7 %	−2.9 %

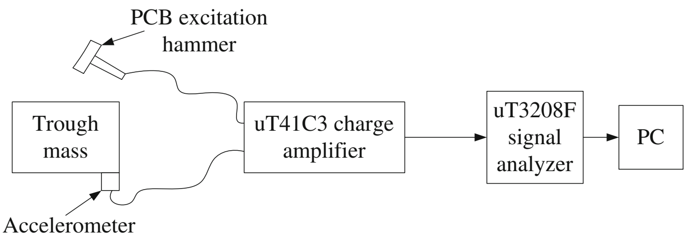
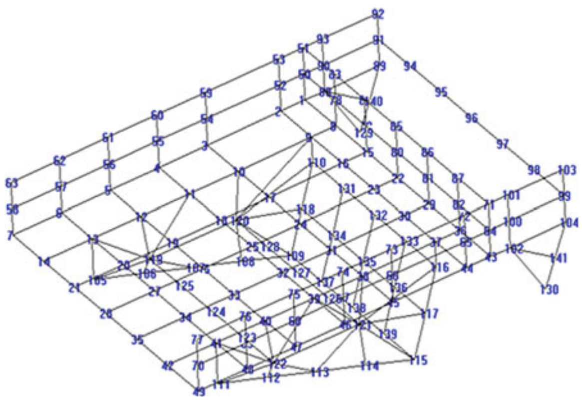


Fig. 16.5 Diagram of the testing system

Fig. 16.6 Arrangement of measuring points



In a word, the agreement between the FEM analysis and measured results is excellent and the experimental results prove the rationality of the FEM model.

16.4 Constrained Modal Analysis of a Two Mass Feeder

Besides the previous simplification of the trough mass, the following simplifications of analysis model of the two mass feeders are made.

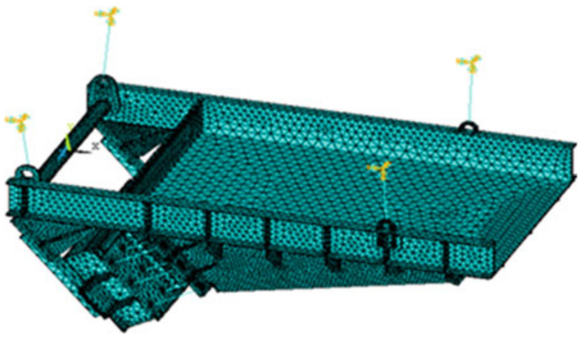


Fig. 16.7 Mesh generation of the two mass feeders

Table 16.3 Natural frequencies of the two mass feeders

Orders	1	2	3	4	5
Natural frequency	1.316	4.938	5.054	6.879	10.062
Orders	6	7	8	9	
Natural frequency	19.153	22.467	28.270	32.331	

1. The vibration motor is replaced by three mass units. The left and the right mass units represent the eccentric blocks located at both sides of the vibration motor respectively and the middle one represents the vibration motor. The middle mass unit works as major node and the other two as secondary ones, all of them are located on the exciter rigidly.
2. The reactor springs and the isolator springs are replaced by unit COMBIN14, which can effectively simulate stiffness and damping of the springs.
3. Unit SOLID95 is used in mesh generation (refer to Fig. 16.7), where upper ends of the isolator springs are all fixed.

The rigid mode should be removed from solutions of the constrained mode, thus the first 9 constrained modes can be calculated. The first 9 orders of natural frequency are shown in Table 16.3. The first 9 mode shapes are also obtained and shown in Fig. 16.8.

The first mode shows a counterclockwise rotational motion with respect to an axis parallel to the y-axis, the isolator springs have a significant deformation, while the deformations of the trough mass and the exciter are rather small.

The second mode shows a rotational motion with respect to an axis parallel to the x-axis, both the reactor springs and the isolator springs have significant deformation, but the deformations of the trough mass and the exciter are rather small.

The third mode presents a moving motion along the y-axis; the exciter has a parallel movement relative to the trough mass along the direction of the reactor springs. The reactor springs have significant deformation. The deformations of the trough mass and the exciter are rather small.

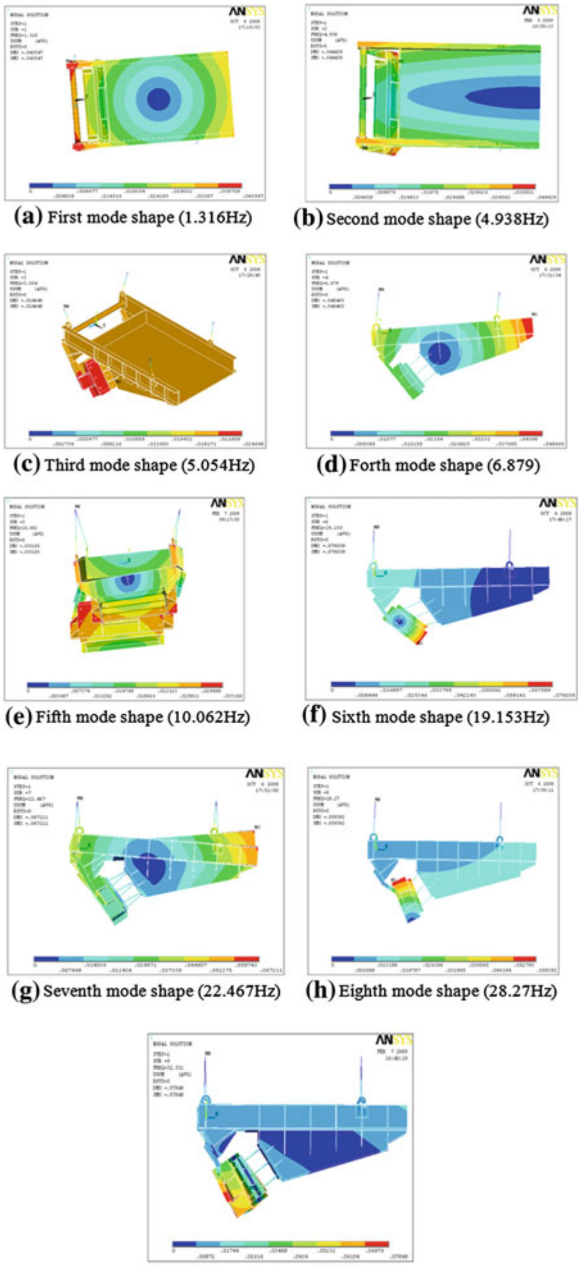


Fig. 16.8 Simulated mode shapes of the two mass feeders

(a) First mode shape (1.316Hz)
(b) Second mode shape (4.938Hz)
(c) Third mode shape (5.054Hz)
(d) Forth mode shape (6.879)
(e) Fifth mode shape (10.062Hz)
(f) Sixth mode shape (19.153Hz)
(g) Seventh mode shape (22.467Hz)
(h) Eighth mode shape (28.27Hz)
(i) Ninth mode shape (32.331Hz)

The fourth mode presents a rotational motion with respect to an axis parallel to the z-axis, the isolator springs have a significant deformation, and the deformation of the trough mass and the exciter is very small.

The fifth mode presents a twist motion of the trough mass. The elastic panels have a rotational and a bending motion with respect to the x-axis. The deformation of the exciter is very small.

The sixth mode presents a bending motion of the elastic panels.

The seventh mode presents a bending motion of the elastic panels and a twist motion of the trough mass with respect to the x-axis.

The eighth mode presents a bending motion of the elastic panels and a clockwise rotational motion of the exciter with respect to an axis parallel to the z-axis.

The ninth mode presents a bending motion of the elastic panels and a twist motion of the exciter. The deformation of the trough mass is very small.

As to the first 4 orders of natural frequency, the difference between the two adjacent ones is rather small, it indicates that the exciting frequency may be close to several of them simultaneously and the vibration superposition can easily be induced.

16.5 Conclusions

FEM model is built and modal characteristics of the trough mass are analyzed by using ANSYS. Correctness of the simulation results are proved by modal tests. On this basis, FEM model of the two mass feeders is built and its modal characteristics are analyzed. Natural frequencies and mode shapes of the two mass feeders are obtained and the reasons for the damage are found.

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