Numerical Simulation and Experimental Validation of Failure Caused by Vibration of a Fan

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Abstract. This paper presents the root cause analysis of an unexpected fracture occurred on the blades of a motor fan used in a natural gas reciprocating compressor unit. A finite element model was established to investigate the natural frequencies and modal shapes of the fan, and a modal test was performed to verify the numerical results. It was indicated that the numerical results agreed well with experimental data. The third order natural frequency was close to the six times excitation frequency, and the corresponding modal shape was the combination of bending and torsional vibration, which consequently contributed to low-order resonance and fracture failure of the fan. The torsional moment obtained by a torsional vibration analysis of the compressor shaft system was exerted on the numerical model of the fan to evaluate the dynamic stress response of the fan. The results showed that the stress concentration regions on the numerical model were consistent with the location of fractures on the fan. Based on the numerical simulation and experimental validation, some recommendations were given to improve the reliability of the motor fan.

Key words: motor fan; modal analysis; finite element analysis; modal test

1. Introduction

The reciprocating compressor units driven by the motor are widely used in petrochemical fields. The oscillatory torques on the compressor crankshaft will induce torsional vibration of the compressor shaft system which mainly consists of the crankshaft, coupling, motor shaft and the components mounted to the shafts. With the development of large-scale and high-speed natural gas reciprocating compressor, the torsional vibration problems in compressor shaft system become more and more serious. As an auxiliary equipment on the motor shaft, the motor fan with a number of blades has high probability of fracture failure due to oscillatory torques and centrifugal forces[1]. As a method to study the dynamic characteristics of the structure, modal analysis plays an important role in the analysis and solution of engineering vibration and fault diagnosis[2]. It is very important to study the vibration characteristics of the fan to ensure that it is not abnormal vibration in the working speed range. Yang Wen studied the response characteristics of fan by using numerical simulation method[3]. Srinivasan summed up the impact on the vibration problems of the gas turbine blades[4].
Poursaeidi analyzed the effect of natural frequency on the failure of the compressor blades[5]. Cheng studied the vibration and stress caused by centrifugal and aerodynamic loads of axial fan blades by using the finite element analysis and experimental measurements[6].

In this paper, the root cause of fracture failure of a motor fan in a reciprocating compressor unit were investigated. Vibration modes of the fan were studied by numerical simulation and experimental validation. The Campbell diagram was used to study the possibility of the low-order resonance of the fan. On the base of modal analysis, a torsional vibration analysis of the shaft system of the reciprocating compressor unit was conducted to obtain the working torques of the fan, and the harmonic response analysis of the fan was performed to predict the dynamic stress of the fan under the acting of working torques. Based on the investigation of vibration modes and stress response, the root cause of fracture failure were identified, and some recommendations were proposed to solve the failure problem of the motor fan and improve the reliability of the reciprocating compressor unit.

2. Numerical simulation

2.1. Geometrical model
The motor fan discussed in this paper was used in a natural gas reciprocating compressor unit. The compressor was a four-throw and balanced-opposed reciprocating compressor, and was driven by a motor, as shown in figure 1. The fan is located at the end of motor shaft. The rated speed of the compressor is 995 r∙min\(^{-1}\) and the first order excitation frequency is 16.6 Hz. The material of the fan is aluminum alloy, and the outer diameter of the fan is 910 mm. A total of 12 blades are evenly distributed, with the thickness of 3 mm. Geometrical model of the fan is shown in figure 2. In the regular inspection, it was found that fracture failure occurred on the blades and base plates of the fan, as shown in figure 3. The cracks initiated from the root of blades and propagated to the middle of blades. After the replacement of new fans in the natural gas reciprocating compressor unit, the fan still have failure fracture problems during the short-term operation, which is in line with the characteristics of torsional resonance damage. In order to identify the root cause of the fan failure problem, the vibration modal analysis of the fan was carried out by finite element simulation and experimental validation.

![Figure 1. Shaft system of the reciprocating compressor unit.](image-url)
2.2. Numerical modelling

2.2.1 Meshing. After simplifying the geometrical model, it is necessary to mesh the model. For some non-key research areas, the simulation results are not sensitive to the density of the mesh. So it can be relatively sparse grid to reduce the amount of calculation and do not affect the final result. And for the focus of the study areas or narrow irregular areas, the mesh is need to be fine, because the number and quality of the grid will directly affect the simulation results. Finally, the tetrahedral element is used for meshing. For the shaft plate of the fan, the grid size of 10 mm was set. For the fan and support plate parts, grid size is set to 2.5 mm due to the thin structure. Through the grid-independent verification, the effect of the calculation frequencies is less than 1% after further increasing the number of grid. A total of 456003 nodes and 187984 elements were generated in the numerical model of the fan.

2.2.2 Boundary condition. The various components of the fan are connected by welding. In the pre-treatment of finite element analysis, the contact area of each component is treated with spot weld, so that the load transfer between the various parts of the model is closer to the actual situation. Considering the constraints of the actual operation of the fan, when the fan is subjected to finite element modal analysis, the shaft of the fan and the keyway side wall are fixed. The damping of the fan is small, and its influence on the natural frequencies of the fan is negligible, so it is not to be considered. The finite element model of the fan is shown in figure 4.
3. Experimental modal analysis

3.1. Modal test

In order to verify the reliability of the finite element analysis results, it is necessary to carry out the modal test of the fan. The experimental modal analysis is based on the input and output of the system, that is, the excitation and response data collected, to make modal parameter identification[7]. The experimental modal analysis method is divided into multiple input single output method, single input multiple output method and multiple input multiple output method. In this paper, single point excitation and multiple point response method (SIMO) is selected. The test system of the fan is shown in figure 5. The entire test system consists of four parts, which are excitation equipment, sensors, acquisition equipment and analysis software. When excitation equipment stimulate the vibration, the sensors pick up the vibration signal into the computer analysis software by acquisition equipment. Finally, the natural frequency, vibration mode, damping and other modal parameters of the fan can be obtained.

In the experimental modal analysis, there are two kinds of structural support types, which are the free suspension and the original boundary conditions of the specimen. This experiment focuses on the resonance frequency and vibration reduction technique of the analysis structure. And the boundary condition of the actual constraint was simulated. According to the selected modal test method, the 086D50 type hammer of PCB company, the 356B18 three-way acceleration sensor, the Dasp-V10 intelligent data acquisition and signal analysis system provided by The Beijing Oriental Vibration and Noise Technology Research Institute were used as test equipments. The experimental modal test of the fan is shown in figure 6.

![Figure 4. Finite element model of the fan.](image)

![Figure 5. Modal test system.](image)

![Figure 6. Modal test.](image)
3.2. Excitation point and response point selection
The selection of excitation and response point is very important in the modal test. The fan is a continuous vibration system. The inertia, elasticity, damping and motion of the system depend on the spatial coordinates. So it is impossible to obtain the continuous distributed response measurement and the infinite number of characteristic solutions. Therefore, in the vibration analysis and parameter identification, infinite degree of freedom of the continuous vibration system is limited to a limited degree of freedom of the vibration system. And a total of 56 vibration points of the fan on both sides of the upper and lower sides of the distribution were selected. Vibration point distribution is shown in figure 7. The principle of selecting excitation point is to make the exciting force easy to transfer to the various parts of the structure and to avoid the vibration nodes and the weak structure\[8\], the test excitation point is to be determined by the reference value of the simulation results. And through trying to knock, the different excitation of the time domain waveform and spectrum are compared. After several trials, the 17th point was selected as the excitation point, and the direction was the vertical direction.

![Figure 7. Modal test model.](image1)

![Figure 8. Result of ERA method.](image2)

3.3. Modal parameter identification
The main task of modal parameter identification is to determine the modal parameters of the vibration system from the measured data, including the modal natural frequencies, modal damping ratio, modal mass, modal stiffness and mode\[9\]. The current parameter identification is mainly divided into frequency domain method and time domain method. Aiming at the this modal test, the morphological parameters of the frequency domain method can be overcame by using the feature system algorithm (ERA method), which is useful for the small damping and modal coupling system, because that can identify dense modes and heavy roots. The result of ERA method is shown in figure 8.

4. Results

4.1. Natural frequencies and vibration mode
The first eight orders of vibration modes of the fan were obtained by numerical and experimental modal analyses. The natural frequencies of the fan are shown in table 1. Limited to the length of the article, the finite element analysis and test results of the most important first three order modes are listed, the contrasts of the first three modes are shown in Figure 9, 10 and 11 respectively.
It can be seen from the table 1 that the calculated results are corresponded to the experimental data, and the natural frequencies calculated by the finite element model agree well with the measured frequencies. The absolute error of the corresponding frequency is below 10%. From the analysis, the modes reflect the structural distortion very well. The first order and the second order are the overall bending mode, while the third order are the composite mode of the bending and torsional, indicating that the established finite element model reflects the actual structure well. The reliability of the finite element modal analysis is verified.
Table 1. Natural frequencies of calculated and measured.

| Order | Calculated(Hz) | Measured(Hz) | Difference(%) |
|-------|----------------|--------------|---------------|
| 1     | 55.70          | 59.07        | 5.71          |
| 2     | 55.72          | 60.41        | 7.76          |
| 3     | 103.79         | 94.60        | -9.71         |
| 4     | 243.67         | 250.00       | 2.53          |
| 5     | 244.09         | 250.08       | 2.39          |
| 6     | 245.05         | 252.15       | 2.81          |
| 7     | 255.66         | 258.44       | 1.07          |
| 8     | 256.10         | 258.58       | 0.95          |

Figure 9. The first-order vibration mode.

Figure 10. The second-order vibration mode.

Figure 11. The third-order vibration mode.
4.2. *Campbell interference diagram*

From the finite element simulation and the experimental modal analysis results, combined with the interference pattern of the fan vibration mode shown in figure 12, the first order natural frequency of the fan is close to the speed of 3 times under the constraint condition, therefore the first order bending mode of the fan may be excited in the actual case, causing the failure of the fan. Despite that the first order mode is the bending mode, there are bending vibrations in the compressor shaft system and the fan will have a certain bending load. Furthermore, the vortex formed by the air flow in the fan will also cause the force to the fan. At the same time, the third order natural frequency of the fan is close to the speed of 6 times, and the bending and torsional modes are shown in figure 11.

![Figure 12. Campbell interference diagram.](image)

5. *Dynamic stress analysis*

5.1. *Torsional vibration analysis of the compressor shaft system*

To obtain the working torques of the fan, the torsional vibration analysis of shaft system of reciprocating compressor units were conducted. The rotating components of the compressor shaft system mainly include the compressor crank, flexible coupling, motor shaft and motor external fan. The reciprocating components including the piston, piston rod and crosshead also have effects on the torsional characteristics of the compressor shaft system. The mass-elastic model of the compressor shaft system was established, as shown in figure 13. The rotating components with large inertia were simplified as lumped mass elements, and rotating components with small inertia were simplified as elastic elements. The equivalent inertia of reciprocating components were simplified by the principle of equal kinetic energy in a cycle. And the shaft system of the reciprocating compressor unit was simplified as 29 lumped masses elements connected with each other by elastic elements. The inertia and stiffness of each section of the compressor shaft system are provided by the manufacturers of the compressor, coupling and motor.

Without considering damping, The torsional vibration analysis was conducted to obtain torsional natural frequencies (TNFs) and mode shapes of the compressor shaft system. The results showed that the first two order TNFs of the compressor shaft system were 92.19 Hz and 191.00 Hz. The first order TNF is appropriately positioned between the fifth and sixth order of run speed, and the torsional resonance of the shaft system will not occur.

The forced response analysis calculates the dynamic torsional response of the compressor shaft system. A modal damping factor of 2% was utilized. The input torques were applied to the mass-elastic model, and the deformation, stress and torques at all sections of the mass-elastic model were obtained. The working torques of the fan were extracted, and the Fourier decomposition was performed to obtain the torque harmonics at each order of run speed, as shown in figure 14.
5.2. **Harmonic response analysis of the fan**

Since the first natural frequency of the fan was in the range of +10% of the sixth order of excitation frequency 99.5 Hz, namely 89.55-109.45 Hz, the low-order resonance was likely to occur. The harmonic response analysis of the fan was carried out to study dynamic stress response of the fan. The sixth order torque harmonic was applied to outer surface of the hub, with the use of the mode superposition method. The damping ratio obtained from the modal test was used for the harmonic response analysis.

After harmonic response analysis of the fan, the stress distribution at the third order frequency was obtained, as shown in figure 15. It can be seen from the figure that the stress concentration areas coincide with the fan's crack locations, which further confirms the possibility of torsional resonance. Based on the above analysis, it was indicated that the excessive dynamic stress at the root of blades caused by torsional resonance attributed to the fatigue fracture of the fan blades.
6. Summary and recommendations
This paper presents the root cause failure of the fan blades of a natural gas reciprocating compressor unit. The results show that the first order natural frequency of fan is close to the speed of 3 times, the natural frequency of third order is close to 6 times, which is easy to cause low-order resonance, and the third order mode of bending and torsional is the main reason. The torque data obtained by a torsional vibration analysis of the compressor shaft system was exerted on the numerical model of the fan to evaluate the dynamic stress response of the fan. The results of the dynamic response analysis of the fan shows that the fan stress concentration regions coincide with the fan crack locations.
It is recommended to adjust natural frequencies of the fan by increasing bending and torsional stiffness or decreasing the weight of the fan to avoid the low-order resonance. At design stage of the compressor, it is necessary for compressor manufacturers to conduct modal analysis, torsional vibration analysis and dynamic response analysis of auxiliary components mounted to the shafts like the motor fan. The torsional natural frequencies of the auxiliary components should be away from compressor excitation frequency to avoid torsional resonance induced by working torques and insure the reliability of the reciprocating compressor unit.

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