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Analysis for Fatigue Failure Causes on a Large-scale Reciprocating Compressor Vibration by Torsional Vibration

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Abstract

The crankshaft of a large-scale reciprocating compressor frequently cracks because of the fatigue failure and the 1st and 2nd crank bearings often are scuffed due to torsional vibration. Such problems are handled by connecting the 2nd and 3rd shaft with two plate flanges in the form of interference fit assembling. Using numerical simulation, models are built for the two crankshafts with modification. Modal analysis and dynamic response calculation of crankshaft system under different modification projects are performed. The results show that the frequency ratio r between sevenfold rotating speed of the initial crankshaft and the first order torsional vibration natural frequency is 1.016, which makes the initial crankshaft to resonate. With the enlarged rotating inertia-mass on modified crankshaft, the frequency ratio r increases to 1.064, and the resonance is eliminated. The rotating vibration displacement of the initial crankshaft at the 1st crank pin is far larger than that of the 6th, which is the primary cause of crank bearing scuffing. The inertia-mass of initial crankshaft results to additional stress with resonance, which is the primary cause of crank cracking for fatigue failure.

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Keywords: reciprocating compressor; failure causes; dynamic analysis of crankshaft; crankshaft torsional vibration

1. Introduction

Reciprocating compressor plays an important role in petroleum and chemical industry. With a sharp increase in flow process, reciprocating compressor is being developed to the large-scale and multi-cranks. The fatigue failure cause and time are different on compressors[1-3].

A crankshaft of a six-crank reciprocating compressor frequently cracks between the 1st and 2nd crank during production. At the same time, there are fissures at the fillet corner between the 6th crank and the main shaft. If the crankshaft is taken the split structure—making two flanges to be connected with heating assembling, these problems can be solved. So the service life of a six-crank reciprocating compressor will be prolonged.

Reciprocating compressor vibration has been the bottleneck of the development for large-scale reciprocating compressors. For many years, some scholars have been engaged in the research work on the FEA dynamics of compressor crankshafts[4-7], but less work has been emphasized on the integrated effect of bearing constraints, reciprocating inertia mass and motor on the torsional vibration of crankshaft system.

In the paper, the dynamic analysis of crankshafts has been developed by finite element software. With the analysis result, the failure reasons are found, and the modification design is proved reasonable.

2. Finite element model of compressor shaft system

The shaft system of reciprocating compressor contains reciprocating mass, crankshaft, rotor of electrical machine, and rigid coupling etc. External loading contains driving force of electrical machine, reciprocating inertial loading and gas pressure on pistons. Main constrains are radius displacement constrains on bears. The compressor has 6 cranks in balance type, on which air cylinders is arranged in sequence 213465.

Shown in Fig.1 is the finite element model (FEM) of initial crankshaft system. According to the structural features and the loads acted on shaft, 3D 10-Node Tetrahedral Structural Solid is adopted in the FEM. Reciprocating

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mass is applied on the system by defining the material density ρ i of every crankshaft pin. The radial displacement constraints on 8 bearings are expressed as ① to ⑧ in the figure. Tangent and radial loads on the 6 crankshaft pins are marked as I~VI, among them the load resulting in torsional vibration of system is the torque acted on crankshaft. The elastic module E, Poisson's ratio μ and the damping ratio ξ of material are 210000Mpa, 0.3 and 0.00015 respectively. The torque acted on the first crank pin of compressor is shown in Fig. 2. The modified shaft system is shown as Fig. 3. The structure shadowed in Fig.3 is rigid.

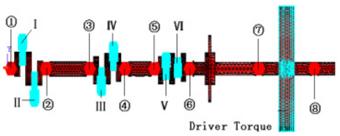
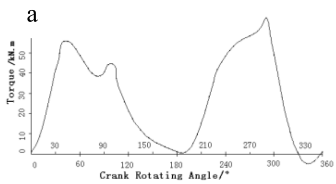


Fig.1 Finite element model of crankshaft



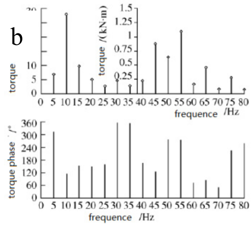


Fig.2 (a) Time history variables of torque (b) Harmonic analysis of torque

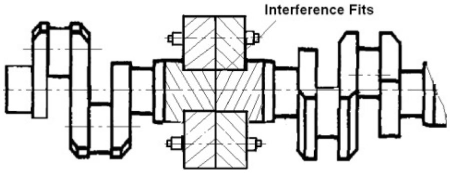


Fig.3 The modified crankshaft

3. Torsional vibration analysis of crankshaft system

The torsional resonance vibration appears when rotating speed (or its integer times) of crankshaft is equal to or close to the 1st torsional natural frequency of the system. Through dynamic analysis, torsional natural frequency, modal shape, time history variables of amplitude and stress at every key point can be obtained. With such parameters the effect of torsional vibration on structural strength on the fatigue failure can be evaluated.

3.1. Modal analysis of crankshaft system

The torsional natural frequency and modal shape of the crankshaft system are gotten here. According to the relationship between frequency and integer times of rotating speed of compressor, the status of resonance vibration of crankshaft system can be identified.

The front three orders of torsional natural frequency before and after modified are picked up (listed in Tab. I).From Tab.1 and Fig.2(b), it can be seen that harmonic frequency of 35Hz approaches the 1st order of torsional natural frequency. The ratio $\boldsymbol{r}$ of sevenfold rotating speed harmonic frequency $\boldsymbol{\omega}$ to the 1st order of torsional natural frequency $\boldsymbol{\omega}_{1,}$ before and after the structure is modified, is 1.016 and 1.064 respectively. So before modified , the compressor rotating speed locates in the resonance region.

Table 1 The Torsional vibration natural frequency of crankshaft

Crankshaft Set	**Initial crankshaft** Frequency Hz	**Modified crankshaft** Frequency Hz
1	34.433	32.895
2	75.546	75.754
3	126.20	129.77