Transient Dynamic Analysis of Crankshaft Torsional Vibration for Reciprocating Piston Compressor

Shijie Wang[1]; Jie Lu[1]; Nan Zhao[1]; Zengjin Xu[2] and **Yuan Li[3]**

[1]School of Mechanical Engineering, Shenyang University of Technology, Shenyang, China
[2]R & D center, Shenyang Blower Group, Shenyang, China
[3]School of Science, Shenyang University of Technology, Shenyang, China

ABSTRACT

The crankshaft of 6M50 type reciprocating piston compressor frequently cracked because of torsional vibration. To perform full analysis of crankshaft vibration of the compressor and solve the cracking problems completely, the finite element analysis software is used, and dynamic simulation of crankshaft torsional vibration is developed and analyzed. Finite element modeling process of compressor crankshafts is discussed, and the results of modal analysis for different modification plans are compared. Transient dynamic analysis of the system is performed. The basic cracking reason for crankshafts with different major diameters is explained theoretically by comparison of calculated and practical results.

Sp. Issue 2013/ # Paper 6
Corresponding author: Shijie Wang
Mailing address: No.111, Shenliao West Road, Shenyang, Liaoning, P. R. China
E-mail: wangshijie@sut.edu.cn
Tel.: +86-24-25496889
Fax: +86-24-25496271

KEYWORDS

Piston compressor; Crankshaft torsional vibration; Modal analysis; Transient dynamic analysis; Finite element method

Introduction

Reciprocating compressor plays an important role in petroleum and chemical industry. With the expansion of flow scale, the compressor is being developed toward the tendency of the large-scale and multi-crank, and the dynamic analysis of crankshaft system is becoming important.

By experiments, it is found that if the number of cranks on the compressor is less than 4, the torsional vibration would not occur; if the number increase to 4~6, the phenomenon would occur (Liu, 2006). In accordance with the rule of critical rotating speed in Standard GB/T20322-2006 (Chinese standard), if the rotating speed of compressor is in the scope of 10% natural frequency of crankshaft or 1/10 rotating speed of compressor in the scope of 5%, the analysis of torsional vibration of shaft system must be required (Kang, *et al*., 1996; Mourelatos, 2001). The reliability analysis of dynamic system is also studied (Wang, *et al.*, 2011; Huang and An, 2009).

In order to research a series of problems occurred on crankshaft of compressor and solve them for enhancing its dynamic behavior, the various reasons of crankshaft crack is illustrated by model and transient dynamic analysis on torsional vibration of 6M50 reciprocating piston compressor, with ANSYS finite element software.

Analysis Methods

6M50 reciprocating piston compressor is a large-scale nitrogen-hydrogen compressor, which came in use in 2000. The compressor has 6 cranks in balance type, on which air cylinders is arranged in sequence 213465. Its main technical parameter is showed as follows.

* Rotation speed: 300r/min
* Stoke: 450 mm
* Maximum force: ≤380kN
* First input pressure: 0.006MPa
* Sixth output pressure: 31.4MPa
* Volume flow: 340m^3/min
* Diameter of crank shaft: 280/300mm
* Shaft power: 4593kW
* Flywheel Moment of electrical machine: 65850kg·m^2
* Coupling: Rigid friction connect

In the original design of the compressor, the diameter of crank shaft was 280mm. After some months, the diameter was increased to 300mm. The abnormal vibration did not occur in the first form and bearings scuffing, but the abnormal vibration occurred in the later. And the bearing scuffing occurred frequently at the 1st and 2nd crank pin. After being used for 202 hours and 346 hours, two cranks were twisted off respectively, and the destroyed location was all in the fillet between the 2nd crank and shaft.

Because productive need, two broken crank shafts were connected together between the 2nd crank pin and the 3rd one by a couple of flange. As a result of interference fit at some crank, the natural frequency of the system changed, and abnormal vibration and bearings scuffing disappeared. After the compressor had operated safely for 4 years, the crack happened at the filet between the 6th crank and shaft. Before the crack of shaft whose diameter is 300mm is not solved, the shaft whose diameter 280 mm, was installed again. After a few months, the crack happened again at the same location. Above mentioned configuration and crack locations are shown in fig.1,a,b,c, and d.

The crankshaft system consists of the reciprocating mass, the crankshaft, the rotor of motor and the rigid coupling. Loads on the system include the driving moment from the motor, the reciprocating inertia force and the gas pressure. The constraint comes from the radial displacement of bearings.

The finite element model of crankshaft is simplified as follows:

(i) according to the principle of energy conservation, reciprocating mass is transformed to the friction damp of the moving component on every crank pin (Ostman *et al*, 2008; Xu *et al*, 2009; Wang *et al*., 2012);

(ii) the fraction force instead of the friction damp of moving parts, the gas pressure and reciprocating inertial force affect together on the system;

(iii) the construction damp is a constant value ξ;

(iv) on the condition of the shaft resonance, the damp of the oil film of main bearings compared with compressor torque is ignored.

The kinematic differential equation of the torsional vibration system of the compressor shaft system is as follows.

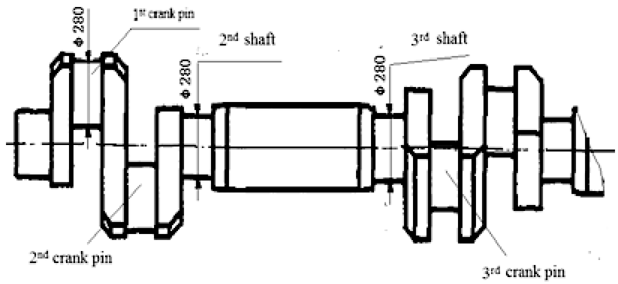
$$[M]\{\ddot{X}\}+[C]\{\dot{X}\}+[K]\{X\}=\{F\} \qquad (1)$$

where $\{X\},\{\dot{x}\},\{\ddot{X}\}$ represent the vectors of the angular displacement, the angular velocity and the angular acceleration of every node in the shaft system respectively. $[M]$, $[C]$, $[K]$ represent the torsional mass matrix, the torsional damping matrix and the torsional rigidity matrix of the whole structure of the compressor shaft system respectively. {F} is the column vector of the node load (Wang, *et al.*, 2012).

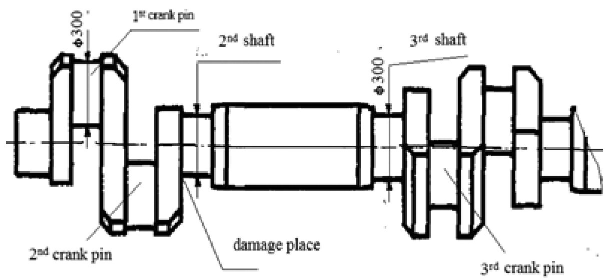
In the paper, the finite element model of compressor shaft is 3D solid model. The time-displacement result *UX* of the crank pins is outputted on node coordination system. *UX* is the displacement (or torsional displacement) in the X -axle direction on the node coordination system. The node torsional angle-displacement θ on the crank pins can be calculated by the function $\theta=\frac{UX}{R}$, here R is the distance from the node to the shaft, and the unit of θ is radian.

During setting up the finite element model (FEM) of the crankshaft system, the assumption should be done. The model of the initial crankshaft system is shown in fig.2. According to the structural features and the loads acted on shaft, 3D 10-Node tetrahedral structural solid element is adopted in the FEM. The reciprocating 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 2 in Figure 2. The tangent and radial loads on the 6 crankshaft pins are marked as I~VI, among them the load resulting in the torsional vibration of the system is the torque acted on the crankshaft.

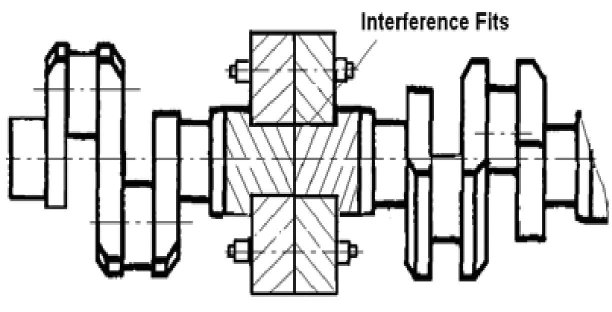
The torque acted on the first crank pin of the compressor is shown in Fig.3, in which the relationship between the torque and the crank rotating angle is expressed with Fig. 3(a) and the Fourier analysis result of the torque can be seen in Fig. 3(b). It is shown in Fig.3 that the phases and amplitudes of harmonic waves are different, and the 7th harmonic frequency leads to the shaft resonant. The elastic module E, the Poisson's ratio μ and the damping ratio ξ of material are 210000Mpa, 0.3 and 0.00015 respectively. The number of the compressor columns is 1, 2, 3, 4, 5 and 6 from the left to the right.



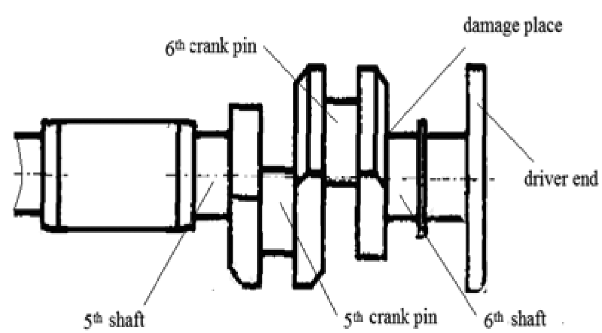
(a) No. 1 Configuration



(b) No.2 Configuration



(c) No.3 Configuration



(d) Crack Location

Fig.1 (a,b,c): Three Configurations of the Shaft and the Crack Location

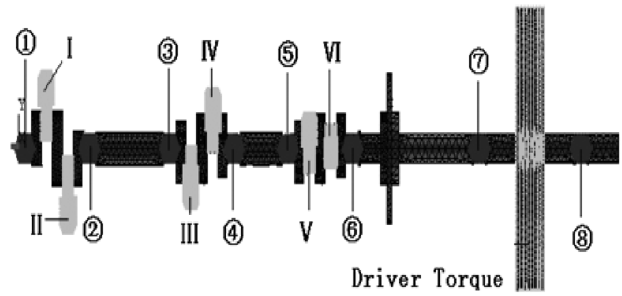
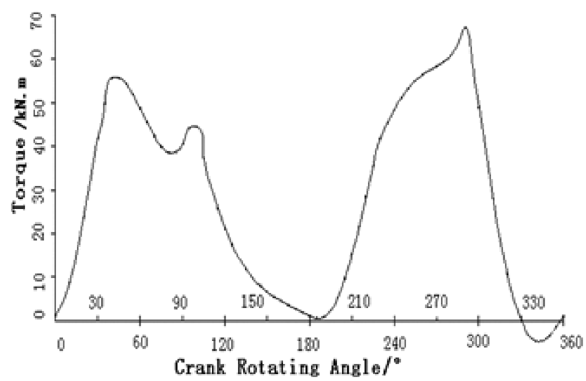
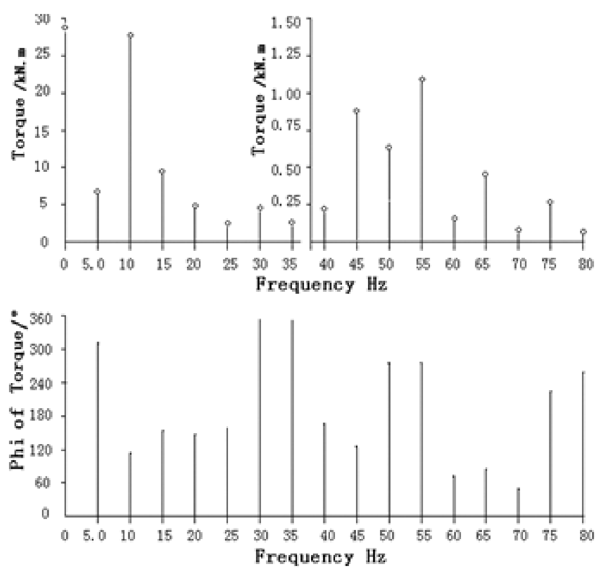


Fig.2: Finite element model of the crankshafts



(a) Time History Variables of the Torque at the crank



(b) Fourier Analysis of Torque

Fig.3 (a&b): The Torque of Crankshafts at the 1st Crank Pin

Results

Torsional vibration is a kind of special dynamic response for compressors systems, where the torsional resonance vibration appears when the rotating speed (or its integer multiple) of its crankshaft is equal to or close to the 1st torsional natural frequency of the system. Through the transient dynamic analysis, the torsional natural frequency, the modal shape and the time history variables of the amplitude and stress at every key point can be obtained. With such parameters, the effect of torsional vibration on the structural strength can be evaluated. Here the modal and dynamic response analysis is performed with mode-superposition method. The analysis contains modal and transient dynamic analysis.

(1) Modal Analysis of the Crankshaft System

By means of the modal analysis, the torsional natural frequency and modal shape of the crankshaft system can be gotten. According to the relationship between torsional natural frequency and integer multiple of the rotating speed of the compressor, the status of resonance vibration of the crankshaft system can be identified. The modal analysis of the crankshaft is the foundation of the transient dynamic analysis by mode- superposition method.

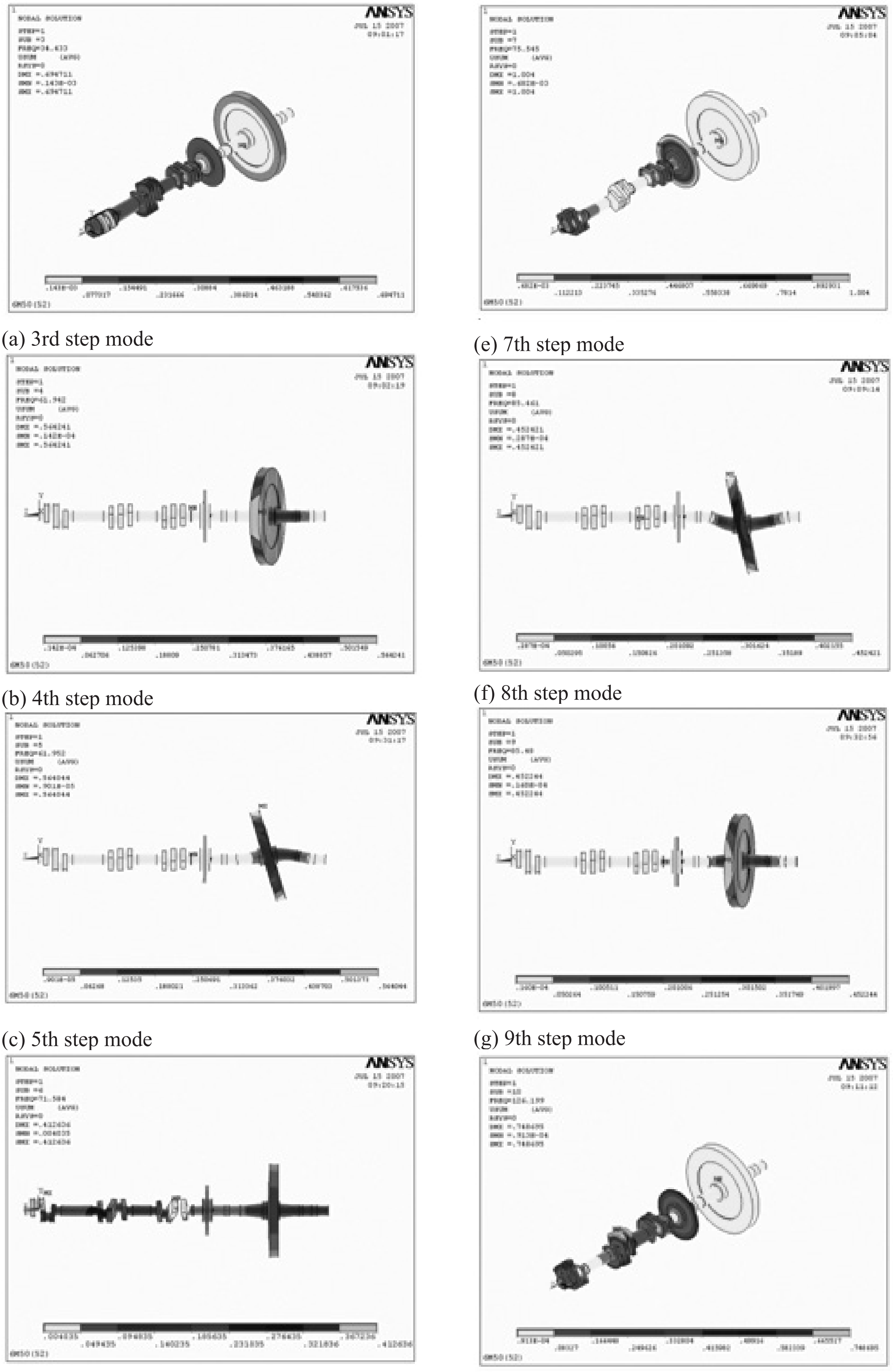
The natural frequency and the modal of a certain 6M50 are shown as table 1. The crankshaft diameter is 280mm on configuration No.1; on configuration No.2 it is added to 300mm; and on configuration No.3, two broken crank shafts are connected together between the 2nd crank and the 3rd one by interference fit with the 300mm diameter. In figure 4, it is shown the 3rd~10th step model of vibration mode on configuration No.2.

From the above results, it is shown that there are multiform modes including the bending vibration, the torsional vibration, the transverse vibration and the combination vibration with the bending and the torsion on the compressor crankshaft system, which special loadings can effect to occur the resonance. On the compressor operation in reality, the crankshafts must have imbalance mass, but the rotating speed of the compressor is far away from the natural frequency of the bending vibration. So the resonance condition would not be satisfied. Because of the imbalance of the torsional moment on the compressor, the system suffers the torsional mode of vibration. And the crankshaft system suffers easily the torsional vibration on the certain conditions. As a result, the main analysis content is the natural frequency and the vibration mode.

By research of every step mode, it is gotten that the 1st and 2nd step mode are rigid displacement in the circumferential and axle direction; the 3rd, 7th and 10th step mode are torsional vibration. In Figure 4, USUM means the total node displacement. In Figure 4(a), (e) and (h), there is a node between the shaft of electrical machine and the crankshaft in the 1st torsional mode, and the amplitude maximum locates on the 1st crank pin; there are two nodes, separately between the 2nd crank and the 5th crank, and between the shaft of electrical machine and the crankshaft; there are three nodes, separately between the 2nd crank and the 3th crank, and between the 6th crank and the coupling, and between the coupling and the shaft of electrical machine.

On the same working condition, the crankshaft system maybe occur many steps torsional modes. For the high step torsional vibration, on the high harmonic frequency of loading stimulating high step torsional vibration, loading energy is weak; on the other hand, many nodes of the high step modes cause the crankshaft vibration difficultly. So the research focuses on the loading causing the 1st step torsional vibration. In Table 1 and Figure 3 (b), it is known that sevenfold harmonic frequency approaches to the 1st order of torsional natural frequency 35 Hz. The ratio r of sevenfold harmonic frequency ω to the 1st order torsional frequency of three configurations, is 1.036, 1.016 and 1.038, respectively.

According to the criterion of GB/T20322-2006, if the value of r lies in the scope of 1 ± 0.05, the resonance vibration will appear. It can be concluded that resonance vibration exists in the three kinds of configurations in practical application with different vibration level. So the analysis on torsional vibration is needed.



(a) 3rd step mode

(b) 4th step mode

(c) 5th step mode

(d) 6th step mode

(e) 7th step mode

(f) 8th step mode

(g) 9th step mode

(h) 10th step mode

Fig.4: 3rd~10th Step Vibration Mode on Configuration 2

Table 1: The Natural Frequencey and Modal of 6M50 Reciprocating Piston Compressor

Step	Frequency (Hz)			Vibration mode
Shaft Configuration	No. 1	No. 3	No. 3	
1, 2	0.0000	0.0000	0.0000	Rigid displacement
3	33.793	34.433	33.731	1st step torsional vibration
4	61.937	61.945	54.323	Bending vibration
5	61.954	61.966	54.341	Bending vibration
6	67.374	71.588	68.287	Transverse vibration
7	73.575	75.546	74.935	2nd step torsional vibration
8	85.459	85.456	75.470	Bending vibration
9	85.472	85.470	75.482	Bending vibration
10	121.87	126.20	124.58	3rd step torsional vibration

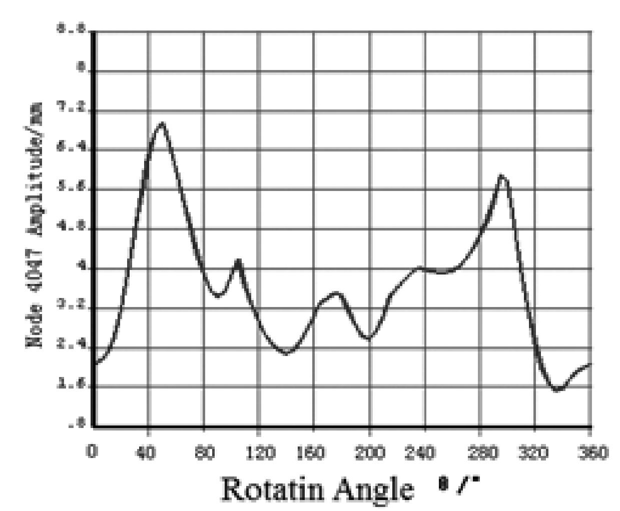
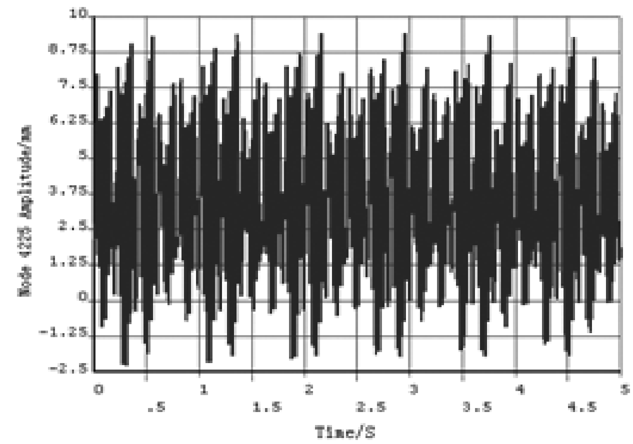
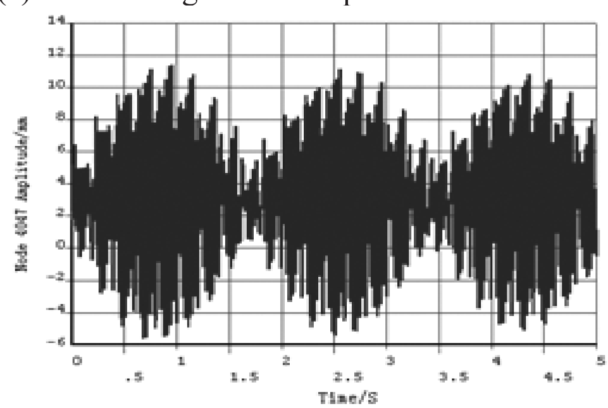


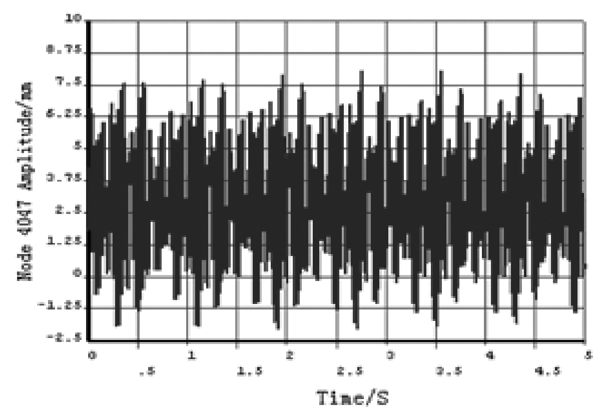
Fig.5: The amplitude of the 1st Crankshaft on No.2 configuration



(a) No.1 Configuration Amplitute-2.4~9.0mm



(b) No.2 Configuration Amplitut-5.8~11.5mm



(c) No.3 Configuration Amplitut-2.0~8.0mm

Fig.6 (a,b,c): The Time History Variables of The Torsional Vibration Amplitude at the 1st Crank Pin on Different Configurations

(2) Transient Dynamic Analysis of Crankshaft System

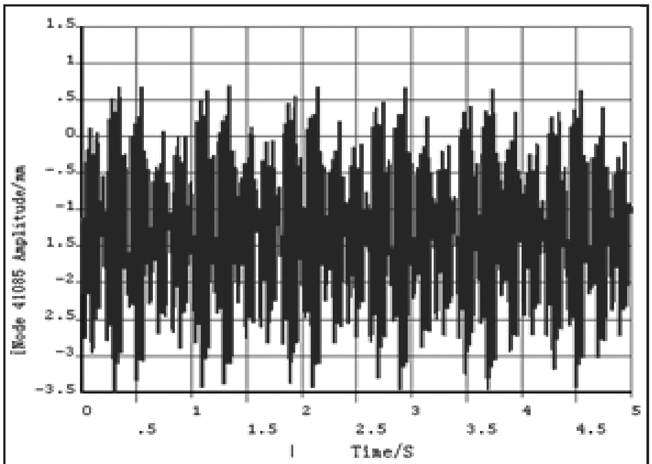
In the transient dynamic analysis of compressor crankshaft, the key to solve problems is to define the dynamic loading and the initial conditions of the compressor crankshaft. Dynamic loading accomplishes by defining the loading steps. The loading model of the transient dynamic analysis of the crankshaft is shown in figure 3 (a). The initial condition defined by the result of static analysis on two static loads when the rotating angle of crankshaft is respective 0^o and 5^o, contains the initial node displacement and the speed. When the compressor rotates one revolution, the static result of the torsional displacement of No.4047 node on the 1st crank pin in No.2 configuration, is shown in Figure 5. The initial condition of the node in dynamic analysis is defined as follow.

$$UX_0 = UX_1$$

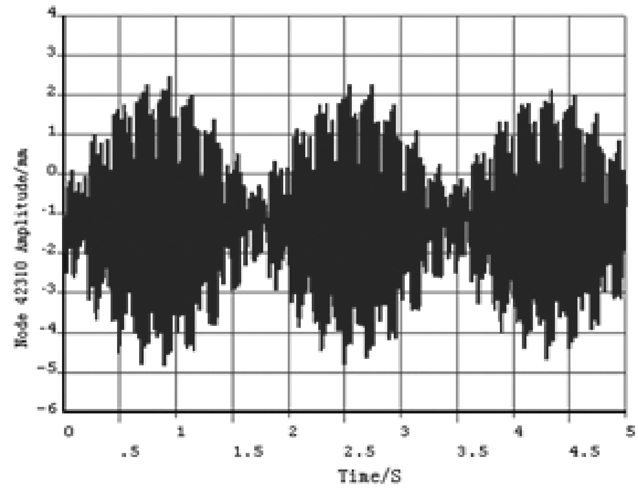
$$VX_0 = (UX_2 - UX_1)/\Delta t \quad (2)$$

Here UX_0 is the initial node displacement; VX_0 is the initial speed; UX_1 is the node amplitude of 0^o rotating angle; UX_2 is the node amplitude of 5^o rotating angle; Δt is time interval of two loading step.

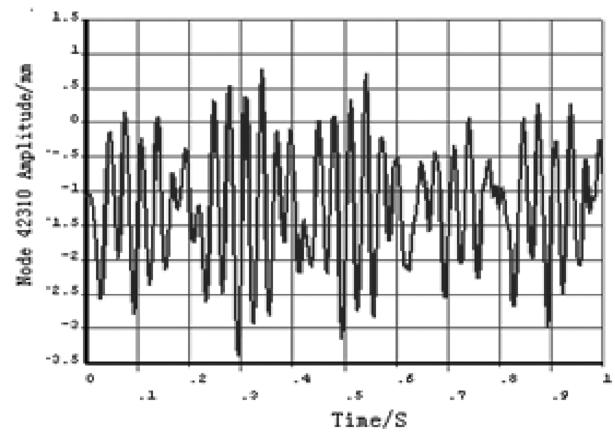
Some key technical parameters, such as the torsional vibration amplitude at the node of each crank pin and the equivalent stress σ_0, the normal stress σ and the shearing stress τ at the location where the stress concentration exist, can be obtained through the dynamic response analysis. The node vibration amplitude is adopted to identify if the impact load occurs. The equivalent stress σ_0 is the fact for checking the static strength of the crankshaft while the normal stress σ and the shearing stress τ are used for checking the fatigue strength. The time history variables of the torsional vibration amplitude at the 1st crank pin on different configurations are shown in figure 6 and that at the 6th crank pin in figure 7.



(a) No.1 Configuration , Amplitut-3.5~0.6mm



(b) No.2 Configuration, Amplitut-4.8~2.4mm



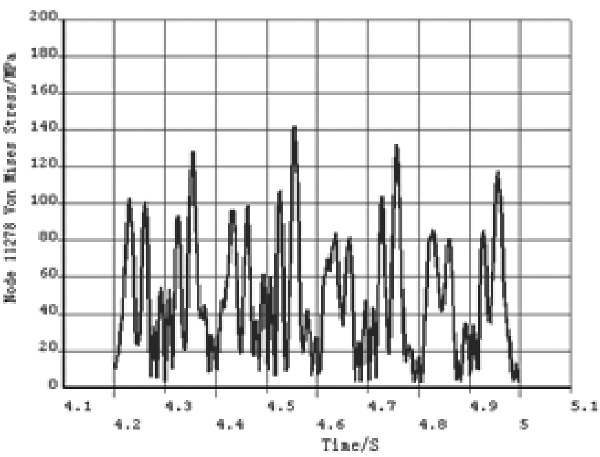
(c) No.3 Configuration, Amplitut-3.4~0.8mm

Fig.7 (a,b,c): The Time History Variables of the Torsional Vibration Amplitude at the 6tht Crank Pin on Different Configurations

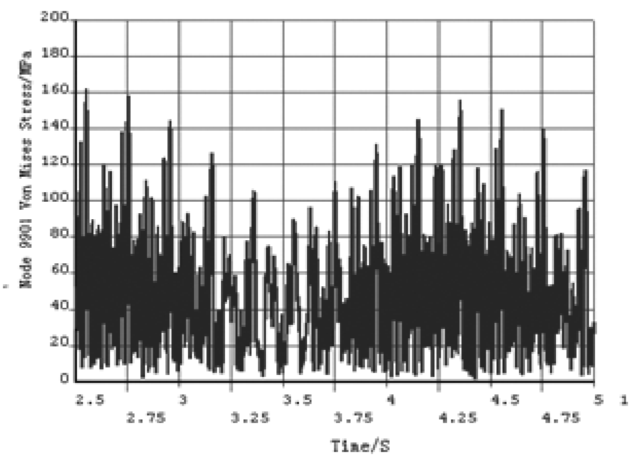
For the crank pin of different crankshaft systems, periodic amplitude beats are formed. From formula $Tb = 2\pi / |\omega_n - \omega|$, it can be known that as the harmonic frequency ω of the compressor approaching the 1st order torsional frequency ω_1, the period of the beat will become longer and more serious torsional resonance vibration will appear. On the contrary, when the frequency of the beat is equal or closer to the rotating speed of the compressor, it can be thought that no resonance vibration or weak resonance vibration exist, which can be verified in figures 6(c) and 7(c).

Varying curve of the equivalent stress at the transition fillet between the 2nd crank and main shaft (simplified as loc2 in fig.1) is shown in Figure 8 and that between the 6th crank and main shaft (simplified as loc6) is shown in figure 9. In the two figures, it is shown that the more seriously crankshaft vibrates, the larger is the node value of the equivalent stress, and the periodic amplitude beats are also formed.

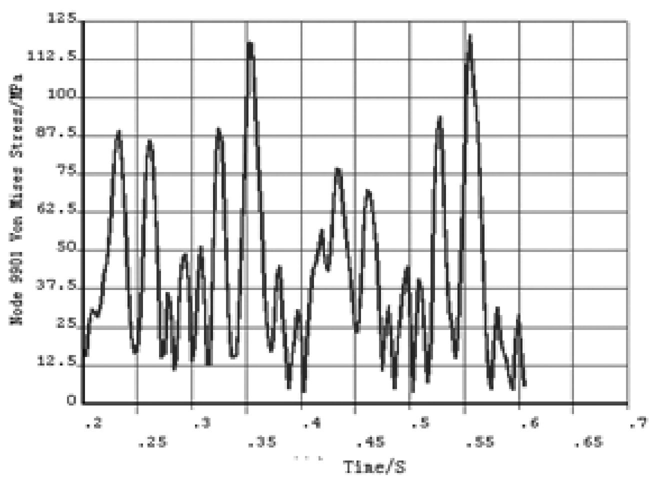
Varying curves of the normal stress and the shearing stress at loc2 are shown in figure 10; varying curves of the normal stress and the shearing stress at loc6 are shown in figure 11. In the two figures, it is shown that the more seriously crankshaft vibrates, the larger is the node values of the normal stress and the shearing stress, and the periodic amplitude beats are formed at the same.



(a) No.1 Configuration, Max. Stress 145 MPa

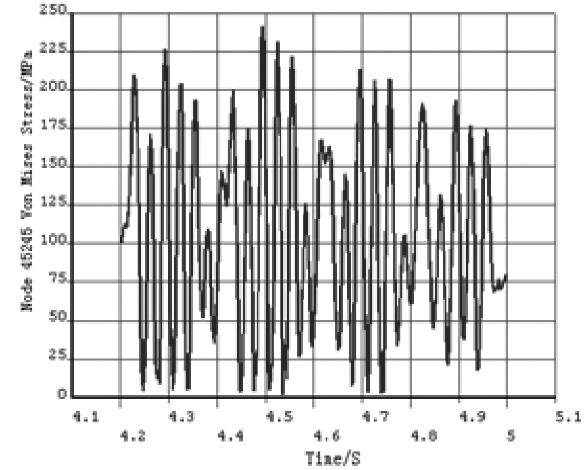


(b) No.2 Configuration, Max. Stress 160 MPa

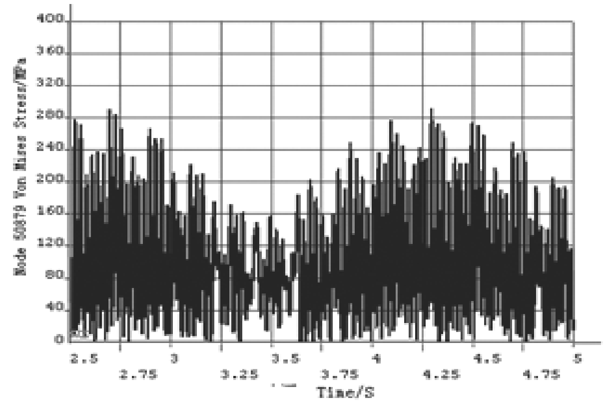


(c) No.3 Configuration, Max. Stress 120 Mpa

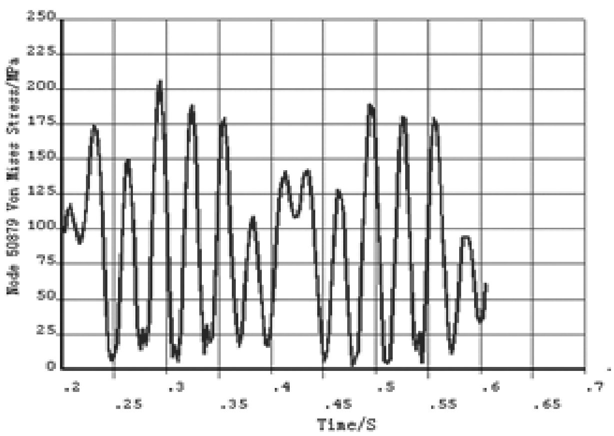
Fig.8 (a,b,c): Varying curve of the equivalent stress at the transition fillet between the 2nd crank and main shaft



(a) No.1 Configuration, Max. Stress 240 MPa

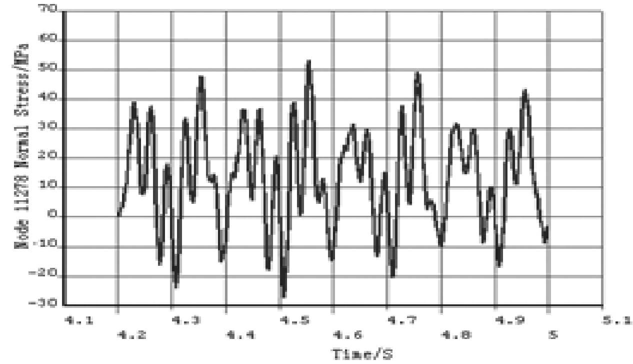


(b) No.2 configuration, Max. stress 290 MPa

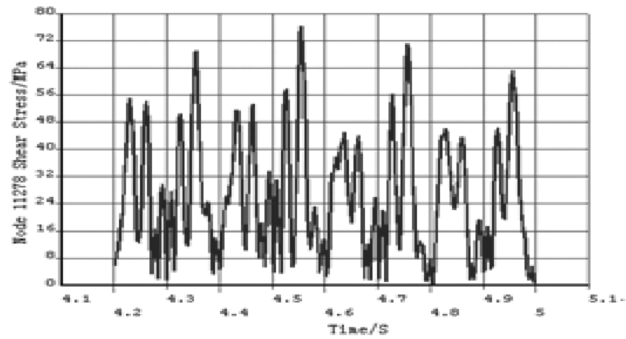


(c) No.3 configuration, Max. stress 206 MPa

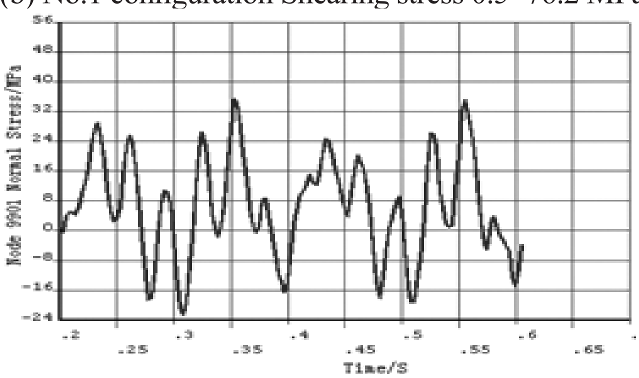
Fig.9 (a,b,c): Varying curve of the equivalent stress at the transition fillet between the 6th crank and main shaft.



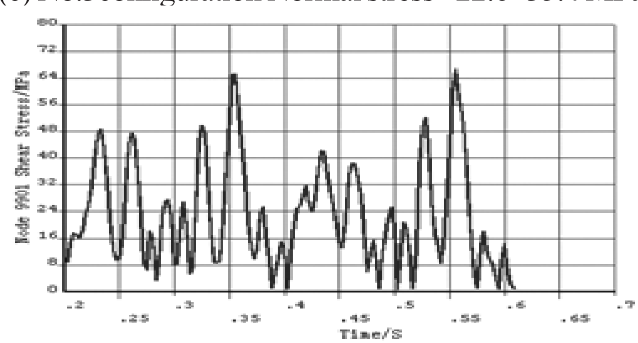
(a) No.1 configuration Normal stress -27.1~53.1 MPa



(b) No.1 configuration Shearing stress 0.5~76.2 MPa



(c) No.3configuration Normal stress -22.6~35.4 MPa



(d) No.3configuration Shearing stress 0.7~66.7 MPa

Fig.10 (a,b,c,d): Varying curves of the normal stress and the shearing stress at the transition fillet between the 2nd crank and main shaft

Conclusion

According to Figures 6 and 7, the following conclusion is gotten by the vibration analysis of the compressor crankshaft:
(1) For the node amplitude of the crank pin on the same crankshaft, that of the 1st crank is larger than of the 6th crank. The result states clearly that if the crankshaft generate the resonance, the vibration is obvious very much on the free end of the crankshaft, and the impact load must exist at the 1st and 2nd crank.
(2) When the diameter of crankshaft is modified from 280mm to 300 mm, the natural frequency of the crankshaft changes and the resonance occurs. The main performance is that the amplitude of 1st crank is larger than other configuration.
According to the result of node stress in Figures 8 - 11, the following conclusion is gotten by the transient dynamic analysis of the compressor crankshaft:
(2.1) For No.1 crankshaft to check the static strength of the crankshaft, the node safety coefficient n at loc2 is 2.68 while at loc 6 is 1.39. To check the fatigue strength, the node safety coefficient n at loc 2 is 2.38. The coefficient n at loc 6 is gotten as 1.48. Such results indicate why the crankshaft crack at loc 6 after a few weeks of production.
(2.2) For No.3 crankshaft to check the static strength of crankshaft, the node safety coefficient n at loc 2 is 3.24 while at loc6 is 2.91. To check the fatigue strength, the node safety coefficient n at loc2 is 1.8. And the coefficient n at loc6 is gotten as 1.77. According to the above result, the static strength and the fatigue strength are closed to the limit value. Such results can further explain why the crankshaft cracks at loc 6 after 4 years of production.
(2.3) For the crack of No.2 crankshaft, because the amplitude of the 1st and 2nd crank is over the allowable limit, this amplitude is made a primary criterion whether the crankshaft is safe, while the node stress of the crankshaft is not the parameter for safety evaluation, but can be one for unsafe evaluation. Such results can explain why the the crankshaft cracks quickly at loc 2 after modifying the diameter.

To sum up, the strength check of the compressor crankshaft should do as following:
(A) When the rotation speed of the compressor is not in the resonant region, the check of static strength and fatigue life must be done;
(B) When the rotation speed of the compressor is in the resonant region, the amplitude check must be done, which can estimate whether the amplitude is over the limit value. When the amplitude is not larger than the limit value, the stress must also satisfy the need of the static and fatigue strength; when the amplitude is over the limit value, the change of rotation speed or crankshaft configuration must be done.

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