Identification of Rotor Crack Faults by Vibration Analysis

Xueyan Zhang¹,* and Guoan He²

⁰Chief expert of Xi'an Thermal Engineering Research Institute Co., Ltd., Xi'an, China
²Deputy director of Vibration Technology Research Institute of Xi'an Thermal Engineering Research Institute Co., Ltd., Xi'an, China

*Corresponding author email: zhangxueyan@tpri.com.cn

Abstract. According to the vibration equation of the cracked rotor, the main vibration characteristics of the cracked rotor are summarized. On this basis, combined with the field diagnosis experience, a practical method of identifying the cracked rotor through vibration analysis is proposed. Furthermore, the proposed method is illustrated by three cases of rotor crack fault identification in the operation of steam turbine and boiler primary air fan. Successful diagnosis avoids the occurrence of potential major accidents.

Keywords: Rotor; Vibration characteristics; Crack fault; Fault identification; Diagnosis case.

1. Introduction

Transverse crack of rotor is a major fault which seriously threatens the safe and stable operation of rotating machinery. If it cannot be found and handled in time, the cracks on the rotor will continue to expand, which may eventually lead to the catastrophic accident of broken shaft and machine failure, causing great economic losses and adverse social impact[1].

It is usually found that the rotor cracks are detected by metal flaw detection during the shut-down of the unit. However, the practice of metal flaw detection shows that the accuracy of the detection results of these parts will be affected due to the geometric limitations of the rotor seal, impeller root, process groove and other parts of the finished steam turbine, and sometimes the existing cracks may not be detected. In addition, the crack growth rate is nonlinear, and it develops rapidly in the last stage. Before the unit shut-down inspection, the crack has developed to the dangerous degree that seriously affects the safe operation of the unit[2][3]. Therefore, there is always the risk of crack generation and propagation between the two metal flaw detection. If the rotor crack fault can be accurately diagnosed and early warned according to the abnormalities found in the state parameter monitoring and analysis during the operation of the unit, the suspected rotor can be shut down in time for targeted flaw detection, and the existing cracks can be found, and the corresponding treatment measures can be taken to avoid the major accident that may occur due to the continuous operation of the unit.

After the rotor crack develops to a certain extent, it usually has an impact on the shaft vibration and rotor dynamic characteristics, such as the vibration spectrum and other characteristics will change. However, when there are transverse cracks in the shaft, due to the different stress state in the crack area, the transverse cracks may present three forms: opening, closing and breathing[4]. Besides the vibration stress, the crack shape of steam turbine rotor is mainly determined by the thermal stress. The vibration characteristics of cracked rotor are different due to different crack shapes under different working conditions. Therefore, it is the key technology to find the vibration characteristic map which can best reflect the change of transverse crack state of rotor under relevant operating conditions[5-7].

In this paper, the vibration equation of cracked rotor is established, and the vibration characteristics of cracked rotor are summarized. Based on the vibration characteristics and the diagnosis experience on...
site, a practical method of vibration analysis to identify rotor cracks is proposed. Three cases of rotor crack identification in operation of steam turbine and boiler primary air fan are given to illustrate, so as to avoid the occurrence of potential major accidents.

2. Vibration Characteristics and Identification Method of Cracked Rotor

2.1. Vibration Equation of Cracked Rotor

For a simple linear Jeffcott rotor, the vibration differential equation is as follows:

$$m\dddot{x} + c\ddot{x} + kx = me\omega^2 \sin(\omega t + \varphi)$$  \hspace{1cm} (1)

In equation (1), \(m\) is the concentrated mass of the rotor; \(c\) is the damping coefficient; \(k\) is the stiffness coefficient; \(e\) is the eccentricity, \(\omega\) is the angular velocity; \(\varphi\) is the phase angle.

After the rotor cracks, the effective cross-sectional area of the cracked rotor and the anti deformation stiffness decreases, that is, the stiffness is \(k - \Delta k\). The \(\Delta k\) is related to the crack depth and the crack opening and closing state when the rotor is rotating. Due to eccentric centrifugal force and gravity, the state of crack opening and closing is often changed, so the stiffness of the cracked rotor is an irregular variable.

When the damping coefficient \(c\) changes, the amplitude also changes. Similarly, because the effective area of the cracked rotor decreases, the intermolecular friction damping force in the material decreases, but the external friction damping force occurs when the two cracked surfaces close, so the damping coefficient \(c - \Delta c\) is also a variable after the crack arise.

Because the stiffness and damping coefficient are unknown, the vibration equation of transverse cracked rotor cannot be established accurately. But its equation can be written by analogy.

$$m\dddot{x} + (c - \Delta c)\ddot{x} + (k - \Delta k)x = me\omega^2 \sin(\omega t + \varphi) + F(\Delta k)\sin(2\omega t + \varphi)$$  \hspace{1cm} (2)

In equation (2), \(F(\Delta k)\) is the disturbing force generated when the crack is closed and opened. For each rotation of the rotor, the crack is closed and opened once, that is, two disturbing forces are generated. Therefore, the disturbing frequency of the crack is \(2\omega\). Equation (2) is a nonlinear equation and has no exact analytic solution, but it can bring the changing factors into the expression of the solution of the original equation (1) for qualitative analysis.

Taking \(k - \Delta k\) and \(c - \Delta c\) as constants, the following results are obtained after omitting them. Vibration amplitude:

$$A = \frac{me\omega^2}{(k - \Delta k)\sqrt{1 - \lambda^2} + \frac{2(\xi - \Delta \xi)\lambda}{1 - \lambda^2}}$$  \hspace{1cm} (3)

Vibration phase:

$$\varphi = tg^{-1}\frac{2(\xi - \Delta \xi)\lambda}{1 - \lambda^2}$$  \hspace{1cm} (4)

In equations (3) and (4), \(\lambda = \frac{\omega}{\omega_n}\) is the ratio of speed to critical speed; \(\xi = \frac{c}{2\sqrt{mk}}\) is the damping ratio coefficient; \(\Delta \xi = \frac{\Delta c}{2\sqrt{mk}}\) is the reduced damping ratio coefficient.

It can be seen from equations (3) and (4) that the amplitude increases with the decrease of stiffness \((k - \Delta k)\) or damping ratio coefficient \(\xi - \Delta \xi\), especially under the condition of critical speed \(\lambda = 1\). The decrease of damping ratio coefficient also makes the vibration phase change, and the abrupt change is \(180^\circ\) at the critical speed. At the same time, because the decrease of damping coefficient is not proportional to the depth of crack, the change of friction damping coefficient between sections is also nonlinear, so the change of vibration phase is irregular.

Although the stiffness and damping of the rotor change after cracks appear, and the vibration characteristics are affected to a certain extent, the change of stiffness and damping is closely related to the crack opening degree and crack growth rate, and the crack growth rate is nonlinear. In addition, the
transverse cracks of the rotor sometimes change the shape of the rotor and the unbalanced mass distribution, resulting in the change of the balance state of the rotor, so the vibration characteristics of the cracked rotor are very complex.

2.2. Vibration Characteristics of Cracked Rotor
The typical vibration characteristics of cracked rotor are as follows:
(1) The stiffness asymmetry of the shaft is caused by the crack. With the continuous expansion of the crack area on the shaft, the remaining section without cracks gradually forms an asymmetric shape, resulting in higher stiffness in one direction of the section and lower stiffness in the other direction perpendicular to it. Under steady state conditions, the most important sign of the existence of cracks in the rotor is the continuous increase of the fundamental frequency vibration component and the second and third harmonic vibration components, especially the appearance and increase of the second harmonic, which is a key sign of the transverse crack propagation on the rotor. The results show that the vibration characteristics are different when the crack is in the state of time opening and time closing and when the crack is always in the state of opening. In the former, although both the first and second harmonic vibrations increase steadily with the crack propagation, the increase of the first harmonic vibration is the main factor. In the latter, the first harmonic vibration response remains almost unchanged until the crack depth extends to a substantial depth, while the second and third harmonic vibrations, especially the second harmonic vibration, increase steadily with the crack propagation.

In many cases, even if cracks appear in the rotor, if the cracks remain closed or only open to a relatively small extent, the contribution of harmonic components to the vibration of the rotor operating at normal speed is relatively small, but mainly due to the influence of unbalanced vibration. The vibration measurement and analysis of many actual steam turbine rotors show that only in a few cases, the obvious second harmonic component will appear in the vibration of the actual cracked rotor at working speed, and at this time, the rotor crack has generally developed to a very serious degree (usually the crack depth is 30% or more of the shaft diameter). The deeper the crack is, the larger the second harmonic is.

(2) The crack causes the reduction of the shaft stiffness. The bending stiffness of the shaft is related to its cross-sectional area. With the continuous propagation of cracks on the shaft, the remaining cross-sectional area becomes smaller and smaller, which will lead to the continuous reduction of the bending stiffness of the shaft. The reduction of shaft stiffness will lead to greater bending in response to static or dynamic loads (such as rotating unbalance), resulting in the increase of the first harmonic vibration (1X). According to the crack orientation, the original unbalance position and the relationship between the operating speed and the resonance speed, the rotor can bend in almost any direction. Therefore, it is possible for the crack propagation to change both the amplitude and phase of 1X at the same time. At the initial stage of crack development, the vibration vector of 1X may change slowly, but as the crack on the shaft continues to expand, the shaft stiffness further decreases, and the growth rate of 1X amplitude may be faster. In the later stage of crack development, the phase of 1X may change significantly.

Because the natural frequency of the rotor is related to the stiffness and mass of the system, the natural frequency is also affected by the change of the shaft stiffness. In general, with the crack propagation, the resonance speed point will gradually move down. The effect of the crack on the resonance mode depends on the rotor mode and the crack orientation. If the crack appears in the region near the node of the rotor mode under a certain mode (there is no obvious bending), the resonance frequency of the mode will hardly change; if the crack appears in the region near the higher mode (generally with larger bending), the effect of the crack on the resonance frequency of the mode will be greater.

2.3. Crack Identification Method by Vibration Analysis
According to the vibration characteristics of cracked rotor, combined with the field diagnosis experience, the following abnormal vibration analysis methods are given to identify the rotor cracks.
(1) The first criteria is whether the second harmonic vibration is developing continuously. With reference to the original rotor vibration data of the machine, the change of 2X vibration during start-up
and shut-down and load operation is observed. When there is obvious 2X vibration and its value increases steadily, it should be suspected that the rotor may have cracks. It should be noted that for the generator and exciter rotor, the uneven magnetic field can also cause obvious 2X vibration, so the vibration data before excitation should be used for comparison. In addition, due to the asymmetry of rotor section stiffness, some generator rotors will produce obvious 2X vibration at 1/2 critical speed, which should be compared with the original data of new machine commissioning.

(2) Continuous vibration climbing and rotor dynamic balance are not effective to reduce vibration. When the rotor cracks develop to a certain extent, the mass center of the rotor will change, resulting in some unbalanced vibration. Generally, the linearity of dynamic balance weighting response of cracked rotor is poor, and the influence coefficient and lag angle of the same type weighting may be quite different from that of normal rotor. Therefore, on the premise of correctly judging the unbalanced axial position and vibration mode of the rotor, if the balancing weight is applied repeatedly, the better effect can not be achieved, especially when the balancing effect is too far from the expected value, it can be suspected that the rotor may have cracks.

(3) The vibration climbs quickly after the phase change suddenly. When the crack develops to a certain critical moment, the change of rotor mass center caused by crack propagation may lead to the change of rotor unbalance direction, and then cause the instantaneous obvious change of vibration phase, and then the vibration increases rapidly. If the vibration phase changes suddenly and sharply during the operation of the machine, and the vibration climbs quickly afterwards, it can be suspected that there may be cracks on the rotor on the premise of eliminating the parts falling off and displacement (for example, the dislocation or opening of the two coupling halves due to insufficient tightening force of coupling connecting bolts during the off design condition of the machine).

(4) During the start-up and shut-down process, the critical speed of the rotor decreased significantly and the vibration characteristics changed evidently. Usually, the stiffness and equilibrium state of the shaft will change after cracks appear, resulting in the corresponding changes of the critical speed and vibration characteristics. If the value of the 1st rotor critical speed during start-up and shut-down is significantly reduced, the vibration under the 1st rotor critical speed is significantly increased, and there are obvious 2X,3X higher harmonic components at 1/2 critical speed, then the rotor crack can be suspected.

3. Case1-Crack Identification of LP Rotor of a 660MW Steam Turbine

Before the middle of July 2011, the vibration condition of No.3-6 bearings of two low pressure rotors of a 660MW supercritical unit had been good. However, it was found that the shaft vibration of No.3 bearing increased after July 18, 2011, and the maximum shaft vibration reached more than 200μm before the unit shut down on August 5. Table 1 lists the historical shaft vibration data of X and Y direction of No.3 to No.6 bearings recorded by DCS system.

The vibration of the unit mainly presented the fundamental frequency component, and the vibration change showed that the shaft vibration of No.3 bearing rose in broken steps. At the same time, the shaft vibration of No.4-6 bearing and the bearing pad vibration of No.3-6 bearing also increased synchronously. However, the shaft vibration of No.3-6 bearing was very small when the unit passed through the critical speed of LP rotor during shut-down, and the maximum vibration was not more than 75μm. Moreover, at low speed (about 300r/min), the journal run-out of No.3-6 bearing was relatively small, which was about 20μm.

**Table 1. Shaft vibration data of each bearing supporting two LP rotors (direct, p-p, μm)**

| Date  | direction | LPA rotor | LPB rotor |
|-------|-----------|-----------|-----------|
|       |           | No.3      | No.4      | No.5      | No.6      |
| 07-16 | X         | 18        | 54        | 48        | 57        |
|       | Y         | 18        | 42        | 57        | 71        |
| 07-26 | X         | 64        | 66        | 58        | 61        |
|       | Y         | 58        | 64        | 64        | 83        |
| 08-05 | X         | 202       | 128       | 100       | 100       |
|       | Y         | 176       | 153       | 97        | 134       |
Due to the relationship between the vibration climbing and the input of desuperheating water in the shaft seal, it was initially diagnosed that the rubbing between the LP rotor and the shaft seal caused the thermal bending of the rotor. The reason might be that after putting in the desuperheating water of the shaft seal, the uneven temperature field appeared in the steam seal section, resulting in the deformation of the steam seal body, the disappearance of the dynamic and static clearance. On August 5, 2011, the unit was shut down for maintenance, including the inspection of #3-#6 steam seal body, steam seal desuperheating water nozzle, etc. No abnormality was found except for the wear marks on the lower steam seal teeth of #3 and #4 steam seals and the Journal of No.3 and No.4 bearings. After the above inspection, the unit started on August 9, 2011. The shaft vibration of No.3-6 bearing was very small during start-up, and the shaft vibration of No.3 bearing was larger when the speed was constant at 3000r/min, and the shaft vibration of X and Y directions were 142um and 125um respectively. Considering that the vibration of No.3 and No.4 bearings of LPA rotor was mainly reversed phase component, a group of 0.85kg anti-symmetric balance weight is added to LPA rotor. After weighting, the maximum shaft vibration of No.3 bearing decreased to 90um at rated speed (see Table 2), and then the unit was connected to the grid and operated with load. With the increase of load and running time, the shaft vibration of No.3 bearing continued to climb. Finally, the unit tripped when the limit value of 254um was reached, and the vibration at critical speed and journal run-out at low speed were still very small.

Table 2. Shaft vibration data before and after the field dynamic balance (direct, p-p, µm)

| Remarks | direction | LPA rotor | | LPB rotor | |
|---------|-----------|-----------|-----------------|-----------|
|         | No.3      | No.4      | No.5            | No.6      |
| Before  | X         | 142       | 94              | 140       | 76 |
| balance | Y         | 125       | 120             | 84        | 97 |
| After   | X         | 90        | 49              | 111       | 70 |
| balance | Y         | 58        | 54              | 75        | 80 |

According to the above analysis and treatment process, it was found that there were some doubtful points in the vibration characteristics that cannot be explained.

- In general, the rotor radial rub impact will cause transient thermal bending, the vibration at critical speed and journal run-out at low speed should be increased significantly during shut-down compared with that during start-up, but this phenomenon was not found during multiple start-ups and shut-downs of the unit.

- The rotor with transient thermal bending can generally return to its original state after turning for a certain period of time. In other words, the vibration of the rotor can basically reach the original vibration level after starting and at the constant speed again. But every time the unit started up, the vibration value increased greatly after the constant speed of 3000r/min, which indicated that the rotor balance state had changed greatly.

- The influence coefficient of the second-order weighting calculation of the LPA rotor was smaller than that of the same type of unit.

Therefore, on the surface, it seemed that the vibration fault of the unit was caused by the friction vibration, but in fact, the balance state of the rotor had changed, which was caused by the internal cause of the rotor, that was, the rotor may have cracks.

On August 25, the LPA cylinder was opened, and it was finally found that there was a circumferential opening crack (about 40% of the circumference) with a length of 770 mm and a depth of more than 30 mm in the middle of the R arc between the shaft seal and the last stage impeller at the electric end of the rotor (No.4 bearing side). The shaft diameter of the crack was 600 mm, as shown in Fig.1. In view of the serious cracks on the LPA rotor, after negotiation between the owner and the manufacturer, the rotor was scrapped, and the manufacturer provided a new LPA rotor for the power plant.
Figure 1. Schematic diagram of the LPA rotor crack location

4. Case 2-Crack Identification of HP-IP Rotor of a 300MW Steam Turbine

The HP-IP rotor vibration of a 300MW unit gradually climbed after cold startup and load on May 21, 2019. The vibration was mainly based on the fundamental frequency, with a certain 2X component. At the same time, the vibration was very sensitive to the main steam pressure and temperature. When the pressure increased or the temperature decreased, the vibration increased. The maximum vibration of No.2 bearing is close to the trip value of 254μm.

Subsequently, the power plant changed the sequence valve to single valve operation, adjusted the valve sequence and other measures to try to control the vibration, but there was no effect. Finally, the unit was shut down on June 22, 2019. The maximum shaft vibration of No.1 bearing reached the full scale (500μm) and that of No.2 bearing exceeded 420μm at critical speed of HP-IP rotor. The maximum run-out at 400r/min of No.1 and No.2 journal were 78μm and 41μm respectively. After the shut-down, the No.1 and No.2 bearings were removed for inspection, and no substantial abnormality was found.

The unit would be started again on July 9, 2019. At 400r/min, the maximum run-out of No.1 and No.2 journal were 39μm and 34μm respectively, the maximum shaft vibration of No.1 and No.2 bearings at the first critical speed of 1560-1610r/min were 176μm and 102μm respectively. After a constant speed of 3000r/min, the shaft vibrations of 1X, 1Y and 2X, 2Y were 103,105μm and 48,45μm, respectively. After the unit was loaded, the vibration started to climb, and it was still very sensitive to the main steam pressure and temperature. Then, the main steam pressure was maintained at 13.5MPa, the load was 150 MW, and the shaft vibration of 1X, 1Y and 2X, 2Y were 135, 150μm and 124, 91μm respectively. Although the vibration decreased with the decrease of main steam pressure and the increase of main steam temperature, the overall vibration base had been increasing. In the process of vibration climbing, the first harmonic component was the main component, but the second harmonic component was also climbing synchronously. By July 12, the maximum value of shaft vibration in X direction of No.1 bearing was close to 190μm, and the second harmonic component was close to 40μm. It was ready to shut down in the afternoon of July 12 due to uncontrollable vibration. Before shut-down, the load was 30 MW, and the maximum shaft vibration of No.1 bearing was about 130μm. The maximum shaft vibration of No.1 and No.2 bearings exceeded 700μm and 320μm respectively at the first critical speed of HP-IP rotor during shut-down(see Fig.2). When the rotating speed reached 400r/min, the maximum run-out of No.1 and No.2 journal were 66μm and 38μm, respectively.

By comparing the Bode curves of cold start-up on July 9 and shut-down on July 12, it was found that there was a great difference. The results showed that the vibration of the HP-IP rotor at the first critical speed during the shut-down was significantly higher than that during the start-up, and the value of the first critical speed during the shutdown was also significantly lower than that during the start-up. The first critical speed was 1560-1610r/min at start-up and 1365-1400 r/min at shut-down, with a difference of about 200r/min. In addition, during the shut-down period of 760-780 r/min, the shaft vibration of No. 1 and No. 2 bearings also had a peak value, which was close to 200μm and 80μm, respectively. The peak value was mainly the second harmonic component, and this speed basically corresponds to half of the first critical speed of HP-IP rotor (commonly referred to as subcritical speed).
The critical speed depends on the mass and stiffness distribution of the rotor. Compared with the cold start-up, the rotor mass would not change in the hot shut-down process, but only the rotor stiffness distribution could be changed (stiffness reduction). Although the stiffness of rotor would decrease slightly in hot state, and the value of the speed rate in shut-down was different from that in start-up, the critical speed difference of HP-IP was usually only tens of RPM during start-up and shut-down, and the difference of critical speed was about 200 r/min, which was obviously abnormal. Moreover, the appearance of the second harmonic vibration peak at the sub-critical speed of the HP-IP rotor also showed that the stiffness symmetry of the rotor had changed.

Due to the significant increase of vibration at critical speed during shut-down, and the increase of No.1 journal run-out at low speed was less than 30 μm, and No.2 journal run-out had almost no change, it showed that there was no obvious thermal bending of HP-IP rotor, and the main reason for the significant increase of vibration at critical speed was the serious deterioration of rotor balance state. All of these indicated that there was serious transverse crack in the rotor. Moreover, when the main steam temperature decreased, the surface of the rotor bore tensile stress, and the cracks on the surface of the rotor should expand, which lead to the increase of vibration.

On July 23, the HP-IP cylinder was opened, and it was found that there was a visible transverse crack at the R angle of the bottom of the stress relief groove (notch diameter of 668 mm) after the HP-IP rotor regulating stage, with the length covering more than 3/4 of the circumference and the maximum depth of 194 mm (see Fig.3). The inspection results confirmed the accuracy of the diagnosis conclusion. Timely shut-down inspection found cracks, to avoid the continuous operation of the unit may occur at any time the shaft broken machine catastrophic accident. Because the rotor crack was too deep to carry out cutting treatment, the HP-IP rotor was scrapped.
5. Case 3-Crack Identification of a Boiler Primary Air Fan Rotor

The primary air fan of a 600MW subcritical unit boiler is a double impeller, adjustable blade fan with rated speed of 1470r/min. The fan shaft is connected with the intermediate shaft and driven by motor. The fan rotor is supported by two lubricated bearings.

The fan was put into operation on October 4, 2020 after maintenance. At the beginning of operation, the horizontal vibration of the bearing at the driving end was basically maintained at about 1.5mm/s. However, since November 16, the vibration began to climb continuously and obviously. The maximum vibration on November 17 exceeded 2.3mm/s, and then the vibration climbing rate became faster and faster. On November 18, the maximum vibration reached 3.75 mm/s.

In view of the continuous climbing of the fan vibration, although the vibration did not reach the trip value, the fan was shut down on November 18. During the shut-down, the maximum horizontal vibration of the bearing at the drive end and the free end were 8.78 mm/s and 5.51mm/s, respectively. After readjusting the center and checking that the bearing seat was normal, the vibration was larger when the fan was started up again than that before shut-down, and the vibration climbing phenomenon still existed. The fan was shut down again on November 20, and the maximum horizontal vibration of the bearing at the drive end and free end was 12.82 mm/s and 8.48 mm/s respectively.

Then, the vibration measurement system was arranged on site to measure the horizontal and vertical shell vibration of the fan at the drive end and free end. The measurement results showed that the overall vibration of the horizontal and vertical shell at the drive end were 171μm and 98μm respectively, in which the 1X and 2X vibration components were 168μm, 57μm and 78μm, 24μm respectively. The overall vibration of the horizontal and vertical shell at the free end were 144μm and 66μm respectively, in which the 1X and 2X vibration components were 128μm, 52μm and 53μm, 24μm respectively.

During the shut-down process, obvious vibration peaks appeared at 909-974r/min, in which the maximum horizontal overall,1X,2X and 3X vibration components of the drive end shell were 279μm, 80μm, 218μm and 46μm respectively, and the maximum horizontal overall,1X,2X and 3X vibration components of the free end shell were 198μm, 63μm, 126μm and 30μm respectively. The maximum horizontal vibration of the drive end and free end bearings recorded by the online vibration monitoring system were 12.51mm/s and 9.32mm/s respectively.

Considering that the vibration kept climbing at the working speed, and there were obvious 1X and 2X vibration components of synchronous climbing, and there was an obvious 2X vibration peak at about half of the first critical speed (design value 1984r/min) of the fan shaft system (910-945r/min) during the shut-down process, which indicated that the rotor shaft of the fan had an increasing stiffness asymmetry fault and the balance condition deteriorates. The most likely reason was that the rotor had a significant crack fault.

On November 28, the fan disintegrated. It was found that there were obvious visible cracks on the shaft of the first stage impeller side near the drive end bearing. The crack direction was perpendicular to the axial direction, and the arc length was about 70 degrees (see Fig.4).

![Figure 4. Crack diagram of fan rotor](image-url)
6. Conclusions
(1) The cracks in the rotor can be identified in time through the vibration analysis. The criteria include the continuous increase of the second harmonic vibration, the fundamental frequency vibration of continuous climbing and the invalid dynamic balance of the rotor, the rapid climbing of the vibration after the 1X phase mutation, the significant increase of the vibration at the critical speed, the decrease of the critical speed and the appearance of the peak value of the second harmonic vibration at the sub critical speed.

(2) Using the crack diagnosis method proposed in this paper, three cracks in the rotor of steam turbine and boiler primary air fan in operation were successfully identified, which confirmed the accuracy of the method and avoided the possible major accidents in the continuous operation of the equipment.

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