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Investigation of blade failure in a gas turbine †

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Abstract

A gas turbine abruptly tripped to a stop during its daily morning start-up period. Out of a total of 81 blades in the first row only one blade was broken at its root. Prior to this accident, there were three blade failure accidents in the same plant during the last 10 years. First, the fracture surface of the troubled blade was investigated. Stress analysis of the blade showed that the maximum stress occurred due to the pressure profiles developed during operation. Modal analysis for one blade and the assembly of blades was performed and Campbell diagram and Interference diagram were drawn to check the dynamic characteristics of the blades. The vibration of the turbine was measured using accelerometers during the operation condition. The result shows that the fatigue fracture of the blade was originated during transient events internal to the combustion chamber which was close to the resonance condition of the assembled blades.

Keywords: Blade failure; Campbell diagram; Condition monitoring; Gas turbine; Interference diagram; Transition piece

1. Introduction

There have been many gas turbine failures. Investigations of the failures have been carried out from the metallurgical and mechanical standpoint [1-5]. Metallurgical investigations generally start from visual and/or scanning electron microscopy (SEM) observations to determine the failure mechanism. Failures generally occur due to cracks that started during exposure to high temperatures or stress concentrations. The cracks initiated from the surface of the blade and propagated to a critical length resulting in catastrophic fracture.

Mechanical investigations usually conclude that the resonance during the vibration of the blade is the cause [6-8]. Numerical simulation for the stress distribution or natural frequencies and their mode shapes of the blade is often used to determine the cause of failure. However, if the failure happens after a long period of operation, it might be concluded that the dynamic characteristics of the blades (mechanical resonances) did not contribute to blade failure.

A gas turbine failure is investigated in this paper from metallurgical and mechanical perspectives.

2. Background

A gas turbine abruptly tripped to a stop during the daily morning start-up period. Out of a total of 81 blades in the first row, only one blade was broken at its root area, and the freed upper platform part was found in the bottom of the exhaust section. Almost all the blades and vanes of the subsequent rows and other areas of the machine also suffered extensive damages by the shock and flying fragments of the blades as shown in Fig. 1(a). The heart of the damage was easily traced to the blade, which was the only blade that was broken at the root as shown in Fig. 1(b). Within a fraction of a second after the blade broke away, the neighboring blades and vanes were subjects to breakage. The debris from the blades and vanes flew in all directions, causing a chain reaction of damage.

Fig. 2 shows recorded operational data such as the pressure and the temperature of combustion gas, the rotational speed of rotor, the generated electric power and the vibrations of the turbine. The start-up operation of the turbine in the morning of the failure was not particularly different from those of the prior week. However, the day before the failure, the amount of electricity produced was at 100% of turbine capacity, where roughly 50% capacity was produced on other days. The vibration level was slightly higher than on the previous day.

Prior to the accident, there had been three cases of blade failure in the same plant since the start of commercial operation. They provide similar results in terms of the location of the crack and the nature of the fracture. The first accident was not sufficiently investigated. The root causes of both previous cases were concluded due to high cycle fatigue. The sources of the malignant frequencies that initiated and propagated the cracks were attributed to the transition pieces and bolts that

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(a) Broken rotor from the first to the fourth row blades

(b) Broken blade in the first row

Fig. 1. Damaged gas turbine blades.

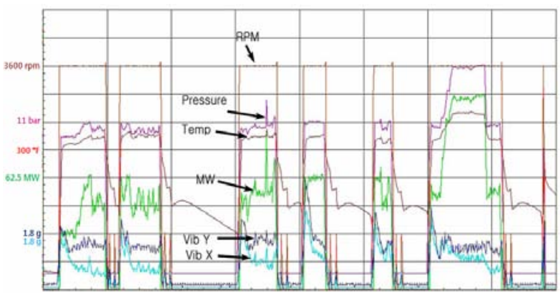
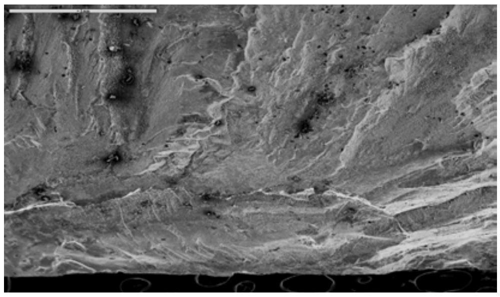
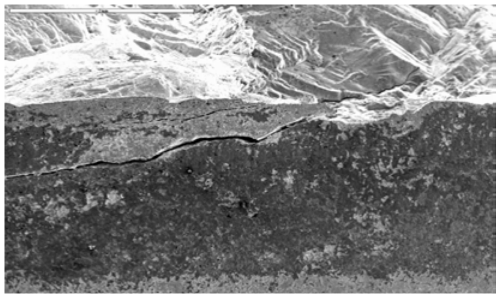


Fig. 2. Operational data during breakage.

Fig. 3. Transition piece and bracket.



(a) Crack origins at fracture surface



(b) Crack initiations at top serration surface

Fig. 4. SEM examination.

were presumed to have failed before the blades [9].

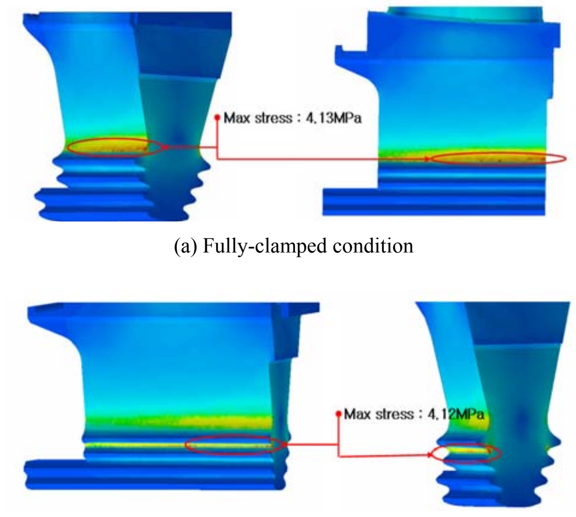
In this case, the damage to the transition pieces was relatively minor compared to other parts as shown in Fig. 3. All 14 transition pieces and their holding bolts were intact with bent brackets and thin gaps between the transition piece and the turbine body after the accident. These findings are the greatest difference from the two other recent turbine failure cases, where the transition piece bolts were found broken after the respective accidents. The only reasonable explanation for the bending of these brackets is the powerful reverse flow of gas due to the abrupt explosion inside of the turbine.

3. Metallurgical examination and stress analysis

If the nucleus of a crack occurs on a metal surface, it must grow over a certain critical size to become an initiated crack. Once a crack is initiated, then its behavior depends on the external stress state. Therefore, if crack initiation occurs and a cyclic tensile load is great enough to propagate the crack in the metal, then the final rupture will be a matter of time. The root cause of the failure can be determined based on where and how the crack nucleated and initiated.

The fracture surface of the failed blade root was examined by a scanning electron microscope (SEM). Fig. 4(a) shows the areas where the crack began. The primary crack was branched laterally, and numerous secondary cracks appeared to have developed simultaneously. It is considered that multiple origins of cracks nucleated independently on the top serration surface that was in contact with the disc.

Fig. 4(b) shows the midpoint of the crack from start to final rupture demonstrating representative morphology of crack propagation. Most of the fracture surface prior to the final rupture shows granular cleavage, or faceted fractures with river lines and ratchet marks. Occasionally beach marks and fatigue striations were also visible indicating that non-ductile fatigue cracking was the means of crack propagation. The location of fatigue crack initiation is generally related with the stress concentration; however, it is hard to explain where the stress concentration originated. Stresses were calculated to determine the location where the maximum stress occurred.



(a) Fully-clamped condition

(b) Partially-clamped condition

Fig. 5. Maximum stress distribution depending on clamp conditions.

Pressure profiles were developed for nozzle wakes, transition piece wakes and flow distortion. Stresses were calculated for a fully-clamped condition, and for the case which the upper portion of the blade was not clamped, based on the assumption of the existence of a small clearance between the blade and the rotor clamp. It means that if the location of the crack initiation coincides with that of maximum stress at the condition of partial clamp, we can conclude that the looseness between the blade and the rotor clamp was the source of the crack initiation. The bending stress due to the pressure profile on the airfoil and the tensile stress due to the centrifugal force during rotation were calculated and combined using SAMCEF software [10]. The pressure was given as 13.7 psi according to the specification of the turbine. The results show that the maximum stress of the fully clamped case is marginally greater than that of the partially clamped case as shown in Fig. 5(a) and Fig. 5(b). And the location of the crack initiation coincides with the position where the maximum stress was occurred in the case of fully clamped in Fig. 5(a). It means that the blade was fully clamped until the breakage. This stress concentration could lead to the initiation of crack and the dynamic situation during operation would tend to enlarge the size of the crack over time.

As the stress analysis shows, the primary cracks at the blade started from the top serration surface, and initially propagated into the blades perpendicular to the exterior surface, and then changed direction horizontally. In addition to the primary crack that eventually led the blade to final rupture, a few secondary cracks were also visible at the top serration surface as illustrated in Fig. 6. When minute elastic deflections or slight motion actually occurs, the cyclic motion of extremely small amplitude is enough to cause micro welding on both surfaces. Fretting wear occurs on mating surfaces that are essentially stationary with respect to each other. Fatigue crack can be initiated on these contacting surfaces. Fretting wear took place

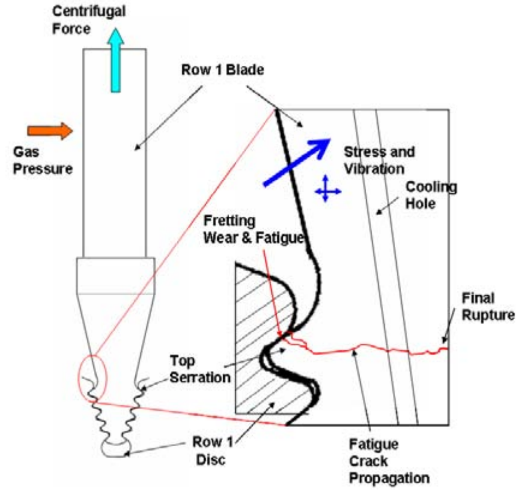


Fig. 6. Illustration of fretting fatigue crack initiation and propagation.

on both the pressure (concave) side and the suction (convex) side of the blades. The crack from the current accident initiated on the pressure side, but blades in the two previous cases were actually cracked at the suction side top serration contact borderline.

The blades were rotating at 3,600 rpm with the centrifugal force and the combustion gas pressure that all blades were equally subjected to. Also, if there were vibration, the cracks will outgrow to a critical size.

4. Modal analysis and interference diagram

The metallurgical investigation showed that the blade rupture was due to high cycle fatigue. This means that the blade was exposed to dynamic loading. Rupture due to dynamic loading suggests that the blade can be subject to resonance. A modal test was conducted to find the natural frequencies and its mode shape for one blade and blade assembly. Modal analysis for one blade and the assembly of blades in the first row was performed. The natural frequencies and mode shapes for a fixed boundary condition were found experimentally and numerically. Modal analysis for the full assembly of the entire blade in the first row was performed by using SAMCEF software. However, the resulting natural frequencies cannot be exact because the high temperature inside the combustion chambers and the rotating condition can change the natural frequencies of the blade assembly [11]. Nonetheless, the results can be informative as to why the accident happened.

To find the natural frequencies of the blade, an impact test for the blade was performed at the fixed boundary condition as shown in Fig. 7. The fixed condition was established using a vice. The frequencies were calculated using SAMCEF software. A comparison of the natural frequencies is given in Table 1 which shows that there are good agreements except for the second mode between the experimental results and the numerical simulation. This is why the experimentally determined shape of the second mode came from the lateral motion

Table 1. Comparison of natural frequencies of the blade for fixed boundary condition. (Hz)

Mode order	Experimental	Numerical
1	576	598
2	977	1316
3	2451	2327
4	2910	2458
5	4661	3715
6	5000	4546
7	5840	4689

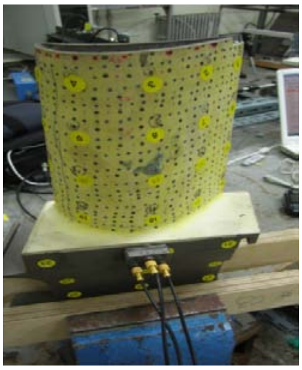


Fig. 7. Modal test for fixed boundary condition of a blade.

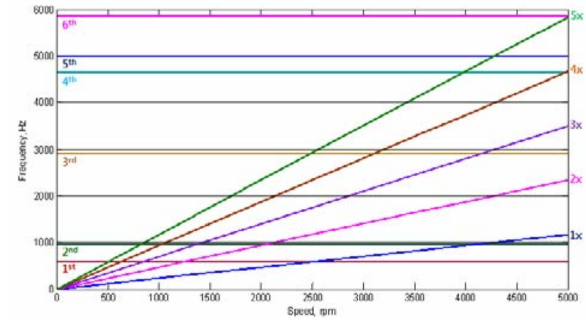


Fig. 8. Campbell diagram.

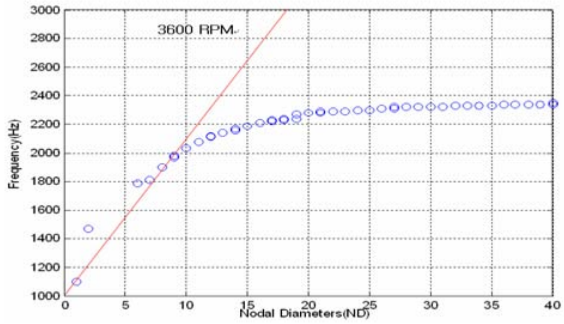


Fig. 9. Interference diagram.

of the fixed blade because the vice could not fully constrain the lateral motion.

If the excitation frequency is close to the natural frequency of a bending mode, then rupture just above the contact line between the blade and the rotor is possible. The values of yield strength and Young's modulus for Inconel 738 of the blade material are reduced as the temperature increases. The first natural frequency can be expected to drop from the room temperature-measured value of 576 Hz to approximately 466 Hz during normal operation from the reasoning of the change in Young's modulus of the blade material due to temperature rise [9]. This shift in natural frequencies assuming all natural frequencies are equally affected by operating temperature. The Campbell diagram (Fig. 8) shows this shift in natural frequencies by assuming that all natural frequencies are equally affected by operating temperature. The closest intersections of frequencies and significant integral order excitation components are marked with circles on the Campbell diagram. It can be noted that many of these do not align well with the operating speed of 3,600 rpm; however, this alone is not sufficient to claim, as this Campbell diagram is only for one blade with fixed boundary condition. In reality, the natural frequencies and the nodal diameter of the assembled blades can be more important.

There are 81 blades at the first row of the gas turbine. The blade assembly has its own modal properties. This was investigated using the cyclic symmetric method for modal analysis for a cyclic symmetric structure [12]. Depending on the mode, large deflections or contacts between neighboring blades are possible. The results show that there are too many natural frequencies compared to those of one blade. There are various types of mode shape depending on the natural frequencies; some of mode shapes show large deflection, some are in the same direction and others are in the opposite direction of the neighboring blades. Interference diagram [13, 14] can be used to avoid the potential excitation of a particular mode of vibration for turbine blades, thus reducing the risk of fatigue failures. Such diagrams are an excellent tool with which to combine information on blade natural frequencies and mode shapes, excitation sources and the operating speed of the machine on the same graph. In interference diagrams, the condition of resonance for the bladed disk is that the natural frequency of the bladed disk must be equal to the frequency of the exciting force and the number of nodal diameters must coincide with the harmonic of the force n. Nodal diameter is a sinusoidal diametric pattern of the displacement of the tip of each blade when plotted with angular position. Each nodal diameter mode shape occurs in a pair. The maximum number of nodal diameters in a bladed disk assembly is half of the number of blades for an even number of blades. For a disk having an odd number of blades, the maximum nodal diameter is half of the total number of blades minus one. Our turbine has maximum 40 nodal diameters as shown in Fig. 9. It shows that the natural frequencies of the first, fourth and sixth nodal diameters produce the frequency close to those when it runs 3600 rpm with the consideration of 14 nozzles. This is the crucial conjecture of this accident. However, if this conjecture is correct, this gas turbine could not sustain for longer period of time. There is a need to investigate the running condition of the gas turbine in operation.

5. Condition monitoring

In general, a gas turbine experiences internal pressure fluctuation due to the combustion process. Combustion is a kind of chaotic process which can lead to the blade vibration induced by an unstable flow excitation as a contribution to failure. If the design and operation condition between two gas turbines are the same, the vibration response should be a similar pattern since the vibration characterizes the state of the combustion chamber.

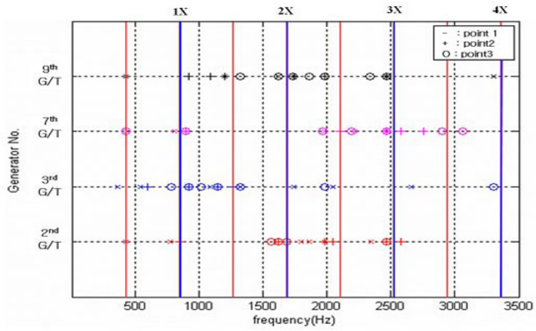
The gas turbine has 14 transition pieces, and each transition piece has one nozzle. The gas turbine runs at 3600 rpm in normal operation. Therefore, the possible fundamental frequency of the vibration can be 840 Hz. The Fourier coefficient of the impulse-like vibration signal can have multiple order of the fundamental frequency. 1/2x, 1x, 2x, 3x… components can appear. The frequency of the measured vibration should be the same since the gas turbines are the same specification and the measured points were the same. In this respect, the vibrations of the four different gas turbines (G/T 2nd, 3rd, 7th, and 9th) at the site, which have the same specifications, were measured using 3 accelerometers at the surface of the turbine casing in operation as shown in Fig. 10(a). However, Fig. 10(b) shows different frequency components depending on the measured position even in normal operation condition. And the multiple order of the fundamental frequency is hard to find in its frequency component. This implies that there are factors affecting the operation condition inside the combustion chambers of the gas turbine, and the operation condition, especially the combustion process inside the chamber, is completely different though the gas turbines have the same specification.

The combustion process can easily be unstable and show different spectra depending on burning conditions. Considering that the blades were in operation for more than 6 years, the fatigue fracture of the blade originated probably during transient events and not during continuous stable operation under vibratory stresses. As shown in Fig. 2 of the monitored signals, the gas turbine was operated with full capacity on the previous day just before the breakage. That was the final test after the overhauling. It is believed that the full operation on the previous day led to more unstable combustion process. Nevertheless the monitored vibration signals at the designated bearings do not show any special indication as the operation condition changes. These monitored signals represent the operation condition of the rotor only, not the combustion chamber. Therefore, it is recommended that the condition monitoring should include a larger variety of signals.

6. Conclusions

The failure of a gas turbine was investigated in order to find the root causes and seek countermeasures that can prevent further accidents. The following conclusions were drawn based on the analyses.

(a) Vibration measurement on the operating gas turbine



(b) Measured frequencies at the gas turbines

Fig. 10. Vibrations of the gas turbine during operation.

(1) Excessive fretting wear occurred on the mating surfaces of the blade root serration and the disc. The contacts were essentially stationary with respect to each other but when slight movement occurred, the cyclic motion of extremely small amplitude was enough to cause micro welding on both surfaces. Fatigue cracking was initiated on the contacting surfaces.

(2) To avoid resonance, excitation frequencies during normal operation condition should be far from the natural frequencies of the structure which can be checked by using Campbell diagram and Interference diagram. The Campbell diagram for one fixed blade shows that the closest intersections of frequencies and significant integral order excitation components were not found. The Interference diagram shows that some modes of the blade assembly are close to a resonance condition. However, this alone is not sufficient since the blade natural frequencies can be affected by operating load and firing temperature.

(3) The vibrations measured on four different gas turbines of the same specifications during stable operation show different frequency components. Considering that the real blades were in operation for more than 6 years it may be concluded that the dynamic characteristics of the blades such as mechanical resonance did not contribute to blade failure. The fatigue fracture of the blade was probably originated during transient events and not during continuous stable operation under vibratory stresses. The combustion process can easily be unstable and show different spectra depending on burning conditions. As a result, high vibratory stresses could be developed in the blades structure. It is believed that the full

operation on the previous day led to unstable combustion process.

(4) Considering the extent of damage of current blade failure, it is difficult to think that the operators had no means of detecting and preventing the failure before it actually happened. It is noted that a more precise turbine rotor vibration monitoring technique must be developed and used for safer operation.

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