J Fail. Anal. and Preven. (2012) 12:391–401
DOI 10.1007/s11668-012-9576-y

TECHNICAL ARTICLE—PEER-REVIEWED

# Failure Analysis and Repair of a Catastrophically Damaged Gas Turbine Compressor Disk Using SEM Technique and CFD Analysis

**A. R. Torabi · M. Bayati**

Submitted: 14 November 2011 / in revised form: 3 April 2012 / Published online: 9 May 2012


**Abstract** During a major overhaul of an 85 MW gas turbine unit in Iran-Rey power plant, 39 cracks were detected with different lengths and locations on the compressor disk of stage 11. All of the cracks initiated from the dovetail regions. Preliminary visual inspections and further microfractography using the scanning electron microscope demonstrated that the fretting fatigue phenomenon was the main cause of failure. Four repair methods were suggested to restart the unit. The first one was to remove all of the cracks from the disk by machining, or the so-called blending. The second, third, and fourth ways were to remove the entire rotor blades of stage 11, to remove the entire rotor and stator blades of the stage 11 simultaneously, and to remove those rotor blades of stage 11 corresponding to the damaged dovetails, respectively. Although the first way of solution was initially carried out on the damaged disk, the first author offered that restarting the unit with the blended disk is not reliable enough because of the presence of a large number of repair points on the disk. Using the numerical investigations based on the computational fluid dynamics, it was found that only the second suggestion (i.e., removing the entire rotor blades of the stage 11) might be applicable. Ultimately, the entire stage 11 rotor blades were removed from the blended disk, and the gas turbine unit was successfully restarted without encountering abnormal operation. Although the performed process resulted in approximately 20% output power loss compared with the unit's power before the blades' removal, the unit was quickly restored to be ready to restart, and the electric power could be generated during the period of peak consumption.

**Keywords** Gas turbine · Compressor disk · Failure · Crack · Fretting fatigue

## Introduction

Gas turbines are complex systems of producing power, a large number of which are utilized in the power plants with the aim to generate electric power. In a gas turbine, the ambient air enters the compressor and passes through the several stages of rotary and stationary blades to be compressed and used to prepare a fuel–air mixture for combustion. Then, the fluid flow with high velocity, pressure and temperature would be able to drive the turbine shaft and produce mechanical power. Finally, turbine shaft runs generator shaft to produce electric power.

In order to obtain high efficiency and long operational hours as well as to prevent failures in gas turbine units, they are periodically inspected and subjected to major overhauls. Although all the gas turbine components are usually inspected at specified time during service, the main attention is commonly paid to the most important parts and locations. Since the stress concentrators in the gas turbine components are the places with a high risk of nucleating and propagating cracks and finally of sudden brittle fracture of the components, these locations are usually checked

A. R. Torabi (✉)
Fracture Research Laboratory, Department of Aerospace Engineering, Faculty of New Science and Technologies, University of Tehran, P.O. Box 13741-4395, Tehran, Iran
e-mail: ali_torabi2002@yahoo.com; a_torabi@ut.ac.ir

A. R. Torabi
Rey Power Plant Research Committee, Baghershahr, Rey, Tehran, Iran

M. Bayati
Department of Aerospace Engineering, Amirkabir University of Technology (Tehran Polytechnics), P.O. Box 15875-4413, Tehran, Iran

by means of the nondestructive testing (NDT) to detect likely cracks and defects. The main stress raisers in gas turbine components are truly the notches having different shapes and sizes. The notches of V, U, O, as well as dovetail (i.e., the disk and blade connecting region) shapes are the most common ones in gas turbines as well as in the other mechanical components and structures. The brittle fracture in some of these stress concentrators have been frequently investigated by several researchers both theoretically and experimentally (see, e.g., [1–8]). Numerous reports have already been published in the past dealing with failures in different components of gas turbines, particularly in the gas and combined-cycle power plants. For instance, one can see [9–14] in which failure in disks and blades of gas turbines have been studied by means of visual inspections, stress analysis, micro- and macro-fractography, etc.

Because most of the gas turbine units in Iran's power plants, particularly the Rey power plant, are timeworn (some of them have more than 150,000 h life time), many types of failures have already occurred in these units, and in some cases, the units are not repairable because of heavy damages, and thus, they are permanently exited from service. Kermanpur et al. [15] investigated the fracture of gas turbine compressor blades, which occurred in Hesa power plant in Isfahan and found that the fracture initiated at the contact between the disk and blades in the dovetail region and because of weak resistance of blade's roots against wear. Recently, a catastrophic damage has occurred for a 32 MW gas turbine unit in Rey power plant during which the compressor disks of stages 16 and 17 were broken, and several cracks of various lengths (from 10 to 300 mm) were detected [16]. Field inspections and technical investigations done by Farrahi et al. [16] demonstrated that the disk failure in the region of connecting to the compressor shaft was due to the fretting fatigue crack initiation and fatigue crack propagation. Moreover, they found that the cracks detected in the dovetail regions were initiated by fretting fatigue mechanism and then they propagated because of the fatigue loading applied [16].

The first Author of the present paper is the chief consultant of the Rey power plant research committee (RRC) and has sighted many incidents in gas turbine units. These events related to the compressor have usually occurred in the last stages (see, e.g., [16]). Widespread investigations performed in RRC dealing with failure in compressor disks have revealed the great contribution of the fretting fatigue phenomenon. The latest failure that occurred in Rey power plant was a catastrophic damage in the compressor disk of stage 11 in an 85 MW gas turbine where 39 cracks having different lengths were detected in the dovetail regions. The type of machine was MHI 701B which operates in Tehran where the temperature is normally between $-10$ and $+40$°C. Its actual rated power in summer under base load conditions was 72 MW. This gas turbine machine is used as a base load unit, and it is programmed to become a peaking unit in the cases of emergency (the number of cases is a few during a year). The numbers of successful and unsuccessful starts before failure were 2,482 and 54, respectively. The unit is normally off during winter because of the low demand of electric power, and it is reserved to be used more reliably during summer. The operational characteristics of the unit were carefully extracted from the maintenance documents, and it was obtained that the effective operational hours (EOH) before failure was about 163,203. The EOH depends on several parameters, such as the number of starts and trips, fast loads, type of fuel (e.g., natural gas or gasoil etc.), etc. which has different computational formulas for various gas turbine models. One of the accepted standard formulas to calculate EOH is—EOH = number of fired hours + 20 × number of starts—which could be used for obtaining the approximate value of EOH, if the model-specific formula is not available for the unit.

Based on the fractographic investigations and by taking into consideration the previous extensive evidences exist in the Rey power plant archive, the cause of failure was identified by RRC to be the fretting fatigue crack nucleation and fatigue crack propagation. In this study, a repair procedure is proposed on the basis of the computational fluid dynamics (CFD) analyses to predict the behavior of the gas turbine compressor. The successful restart of the unit 32 gas turbine demonstrated that although the proposed method resulted in output power loss of approximately 20%, it might be a good temporary solution to the problem in emergency situations where the beneficiaries could not afford the unit to be idle for rather a very long time.

## Description of the Incident and Failure Analysis

### Description of the Incident

During a major overhaul of an 85 MW gas turbine unit with more than 163,000 h life time in the Rey power plant, after opening the turbine casing, 39 cracks with different sizes in length and depth were detected by means of preliminary visual inspections followed by nondestructive magnetic particle testing (MT) on the compressor disk of stage 11 entirely in the dovetail regions. Figure 1 shows a dovetail region corresponding to the investigated compressor having six hypothetical cracks of lengths $L_i (i = 1, 2, \ldots, 6)$.

Note that the hypothetical cracks shown in Fig. 1 are presented herein to characterize more conveniently the real

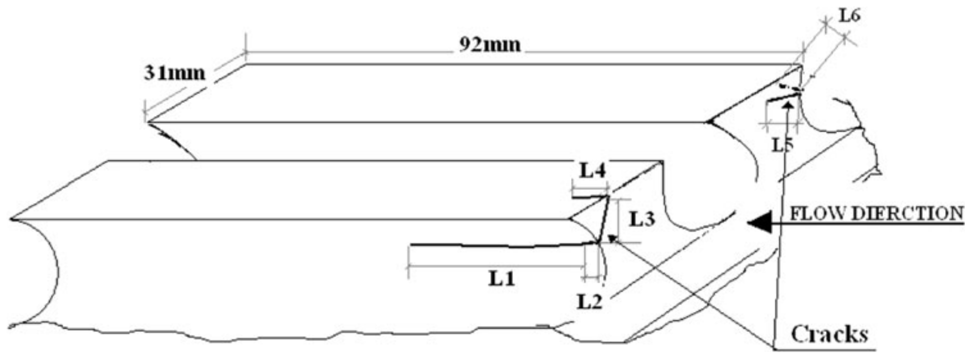


**Fig. 1** Disk dovetail region with six hypothetical cracks

cracks detected in the dovetails. The parameters $L_i$ for the cracks are presented in Table 1.

Figures 2 and 3 display the rotor part of the damaged compressor and some of the cracks detected in the dovetail regions, respectively. As shown in Fig. 3c, those cracks located in the back side of the dovetail region and hence could not be directly seen with eyes, were detected and characterized by means of a mirror.

It is noteworthy that the dovetail regions on a gas turbine compressor disk are the places with a high risk of nucleating cracks due to the existing contact between rotor blades and disk. The nucleated cracks can be propagated by fatigue mechanism (because of the fluctuating nature of the loads applied to the components) till detection at overhaul or violent sudden fracture during operation. This type of failure has frequently occurred in the Rey power plant gas turbine units, the most recent of which has been investigated in [16], and the latest one is described in the present study. Figure 4 shows a few cracks initiated from the dovetail regions of the compressor disks of stages 17 and 18 in a 32 MW gas turbine unit in the Rey power plant [16].

### Failure Analysis

In order to investigate the cause of failure more accurately, several cracked pieces removed from the dovetails during blending process were subjected to micro-fractography by scanning electron microscope (SEM). Figure 5 reveals some of the studied pieces.

As seen in Fig. 5, each piece contains two crack lines branched from the crack origin. The cracked surfaces were studied by SEM and during investigations; the attention was mainly paid to find likely trace of fatigue failure. Relatively similar surface patterns were found for the pieces and all of them demonstrated clearly the fatigue failure mechanism. Figures 6 and 7 represent two micrographs of a cracked surface taken by SEM with different magnifications (×500 and ×3000, respectively). Both the micrographs were taken from the zones of crack growth very close to the crack lines.

As seen in Figs. 6 and 7, obvious fatigue striations demonstrate the dominance of fatigue failure mechanism. The main difference between these figures is that Fig. 7 shows only the fatigue striations, but Fig. 6 represents both the fatigue striations and the surface effects of wear.

## Repair

After detecting the cracks on the compressor disk of stage 11, the Rey power plant maintenance and repair committee (RMRC) decided to repair the disk in a short time interval because, the need of the electric power was very serious and the disk replacement in a short time was not possible. The first repair procedure considered was to remove the entire cracks from the disk by means of a well-known process, called blending. A review of the guidelines prescribed in the maintenance manual of the gas turbine revealed that 37 detected cracks were allowable to be removed, and two cracks were not so because of having larger lengths than permitted. Therefore, the failure report was sent to the manufacturing company by the committee, and then, a positive response dealing with removing the entire cracks was received. The manufacturing company had also stated that the unit was not guaranteed to be safe in service because of the presence of two long cracks. Anyway, the entire cracks were removed from the disk by blending. Figure 8 shows a blended dovetail of the damaged disk of stage 11.

The first author, as the chief consultant of the RRC, stated that the blended disk was not reliable to be used because of the presence of a large number of crack removals on the disk. He suggested that the stage 11 to be removed from the compressor cascade. For this purpose, three repair options were offered, which comprised removal of (1) the entire rotor blades of stage 11, (2) the entire rotor and stator blades of stage 11 simultaneously, and (3) those rotor blades of stage 11 corresponding to the damaged dovetails. The RRC was deeply concerned with this serious question of whether the compressor would experience any severe instability, such as serge, stalls, etc.

**Table 1** The parameters $L_i$ for the cracks detected in the dovetail regions on the compressor disk of stage 11

| Crack number | Blade number | Crack length | | | | | | Remark |
|---|---|---|---|---|---|---|---|---|
| | | $L_1$ (mm) | $L_2$ (mm) | $L_3$ (mm) | $L_4$ (mm) | $L_5$ (mm) | $L_6$ (mm) | |
| 1 | 1 | 5 | 5 | 0 | 0 | ··· | ··· | ··· |
| 2 | 5 | 12 | 7 | 0 | 0 | ··· | ··· | ··· |
| 3 | 13 | 26 | 0 | 3 | 0 | ··· | ··· | Two cracks on one root |
| 4 | 13 | 18 | 0 | 9 | 4 | ··· | ··· | |
| 5 | 15 | 16 | 0 | 8 | 0 | ··· | ··· | ··· |
| 6 | 16 | 25 | 0 | 9 | 2 | ··· | ··· | ··· |
| 7 | 18 | 23 | 0 | 9 | 0 | ··· | ··· | ··· |
| 8 | 22 | 34 | 0 | 9 | 0 | ··· | ··· | ··· |
| 9 | 23 | 3 | 1 | 0 | 0 | ··· | ··· | ··· |
| 10 | 24 | 13 | 0 | 0 | 0 | ··· | ··· | ··· |
| 11 | 26 | 30 | 0 | 9 | 0 | ··· | ··· | ··· |
| 12 | 27 | 2 | 0 | 0 | 0 | ··· | ··· | ··· |
| 13 | 30 | 2 | 2 | 0 | 0 | ··· | ··· | ··· |
| 14 | 35 | 8 | 2 | 0 | 0 | ··· | ··· | ··· |
| 15 | 37 | 30 | 0 | 9 | 1 | ··· | ··· | ··· |
| 16 | 39 | 10 | 0 | 0 | 0 | ··· | ··· | ··· |
| 17 | 43 | 9 | 0 | 0 | 0 | ··· | ··· | ··· |
| 18 | 45 | 4 | 2 | 0 | 0 | ··· | ··· | ··· |
| 19 | 47 | 3 | 2 | 0 | 0 | ··· | ··· | ··· |
| 20 | 48 | 8 | 0 | 0 | 0 | ··· | ··· | ··· |
| 21 | 53 | 23 | 0 | 8 | 0 | ··· | ··· | ··· |
| 22 | 54 | 20 | 0 | 0 | 0 | ··· | ··· | ··· |
| 23 | 56 | 36 | 0 | 9 | 4 | ··· | ··· | ··· |
| 24 | 58 | 27 | 0 | 9 | 1 | ··· | ··· | ··· |
| 25 | 59 | 7 | 0 | 0 | 0 | ··· | ··· | ··· |
| 26 | 61 | 13 | 0 | 0 | 0 | ··· | ··· | ··· |
| 27 | 63 | 30 | 0 | 9 | 3 | ··· | ··· | ··· |
| 28 | 64 | 9 | 0 | 4 | 0 | ··· | ··· | ··· |
| 29 | 65 | 35 | 0 | 9 | 9 | ··· | ··· | ··· |
| 30 | 66 | 3 | 0 | 0 | 0 | ··· | ··· | ··· |
| 31 | 67 | 30 | 0 | 9 | 0 | ··· | ··· | ··· |
| 32 | 68 | 12 | 0 | 2 | 0 | ··· | ··· | ··· |
| 33 | 69 | 24 | 0 | 8 | 0 | ··· | ··· | ··· |
| 34 | 70 | 20 | 2 | 0 | 0 | ··· | ··· | ··· |
| 35 | 71 | 40 | 0 | 9 | 1 | ··· | ··· | ··· |
| 36 | 73 | 15 | 2 | 0 | 0 | ··· | ··· | ··· |
| 37 | 76 | 26 | 0 | 9 | 0 | ··· | ··· | ··· |
| 38 | 79 | 25 | 0 | 5 | 0 | ··· | ··· | ··· |
| 39 | 81 | 4 | 0 | 0 | 0 | ··· | ··· | ··· |

To answer this question, several qualitative gas flow analyses were performed on a simple three-stage axial compressor found in the literature by means of CFD, and the fluid behavior was investigated and modeled with the removed blades in accordance with the three ways of solution suggested in RRC. In the next section, the CFD analyses are elaborated.

## Analysis of Fluid Flow

Investigation of the compressor internal flow and performance at on-design and off-design conditions has been done experimentally, as well as by means of tentative observations [17–19]. Nowadays, the CFD is a fool-proof method for investigation of cascade at both on-design and

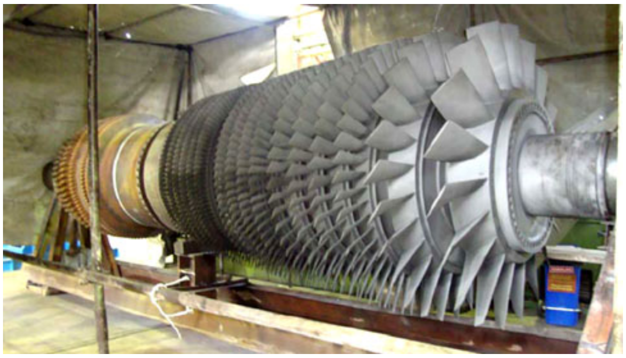
Fig. 2 Damaged compressor rotor

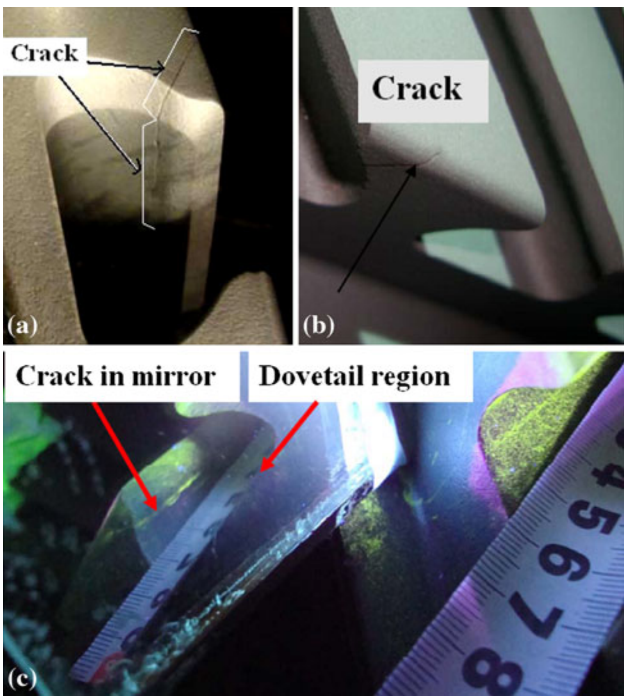

Fig. 3 Cracks detected in the dovetail region. (a, b) Directly visible with eyes. (c) Directly invisible and detected in mirror

off-design conditions. CFD has been improved by Sisto et al. [20], Jonnavithula et al. [21], and other investigators. In CFD, the Navier–Stocks (NS) equations at a discrete domain with a turbulence model, such as Boldwin–Lomax or $k$–$\varepsilon$ [22], are solved by the different methods such as finite volume method (FVM). One or more compressor stages containing several blades can be modeled to study the interaction between vanes themselves, as well as the rotor–stator interaction on the fluid flow. In this section, we studied the effects of removing some rotor blades as well as an entire stage on the compressor performance.

In order to study the fluid flow behavior in the compressor cascade, and hence, the compressor performance, when the three specified blade removals (i.e., the three

Fig. 4 Cracks initiated from the dovetail regions on the compressor disks and propagated toward the disks body [16]

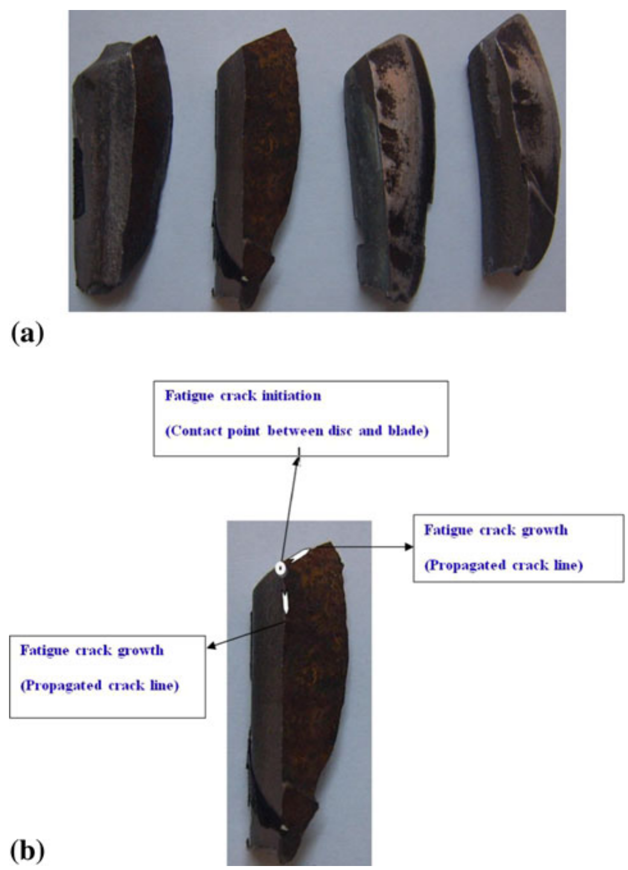

Fig. 5 (a) Some of the cracked pieces removed from the dovetails during blending. (b) The regions of crack initiation and crack growth

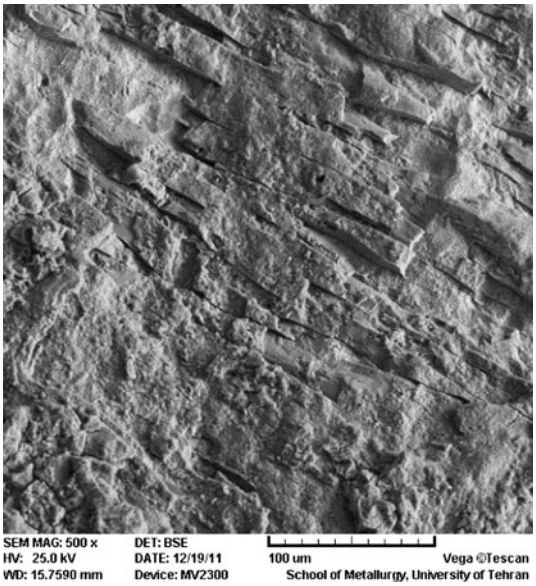


**Fig. 6** Fatigue striations on the surface of a removed piece corresponding to the zone of crack growth near the short crack line (SEM magnification ×500). The photo was taken from an area very close to the region of crack initiation

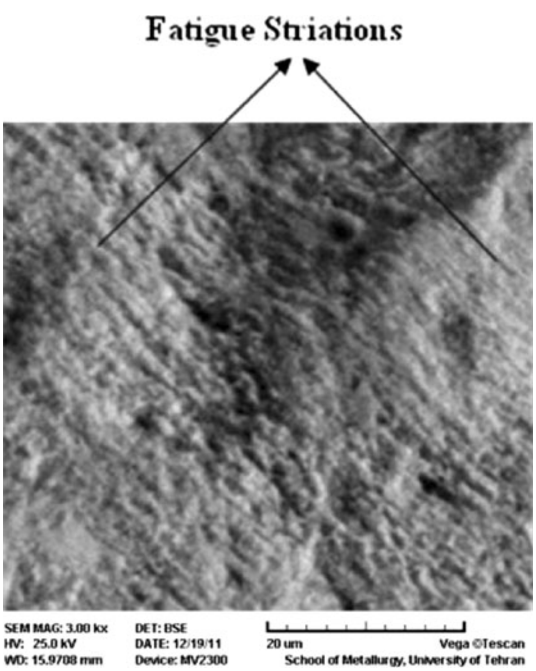


**Fig. 7** Fatigue striations on the surface of a removed piece corresponding to the zone of crack growth very close to the large crack line (SEM magnification ×3,000)

ways offered by RRC) are applied, we need to model at least three stages for which the removals are necessarily located in the middle. To do this and by taking into account that we required only a sample axial compressor with at least three stages for investigating the flow behavior qualitatively, a literature survey was performed to see if we


**Fig. 8** A blended dovetail of the damaged disk of stage 11

can find a simple model. Ultimately, a three-stage axial compressor cascade for which each stage consists of seven rotor and seven stator blades was found in [23] and utilized to analyze the fluid flow. Three cases of study corresponding to the three ways of blade removals were considered for the flow analysis. The obtained results were compared to those for a complete three-stage axial compressor cascade presented in [23]. In the created model, the entire rotor blades of the middle stage, the entire middle stage of compressor cascade containing rotor blades and stator vanes, and two rotor blades corresponding to the two hypothetical cracked dovetail regions were removed, respectively. In the forthcoming subsections, the details of modeling and flow analysis are described.

## Numerical Method

The two-dimensional, compressible NS equations were solved by FVM. Since no coriolis acceleration exists, the governing equations at relative coordinate system are equal to the absolute governing equations. Also, no energy transfer exists, and the total entropy is considered to be constant. Therefore, we used the absolute velocity by adding the rotational velocity to the relative velocity. Periodic boundary conditions at up and down domains as well as the velocity and pressure at inlet and outlet boundaries were used, respectively. The domain was divided into several separate blocks corresponding to the rotor and stator. The rotor blocks are rotational, and the stator blocks are fixed. Figures 9 and 10 show the physical domain, and the unstructured grids around the blades, respectively.

In this study, the RNG $k$–$\varepsilon$ turbulence model [22] was used, and the fluid was assumed as a perfect gas to relate the pressure and density. The grids were refined at the leading and trailing edges of the airfoils because; the gradients of the flow parameters are high at these locations.

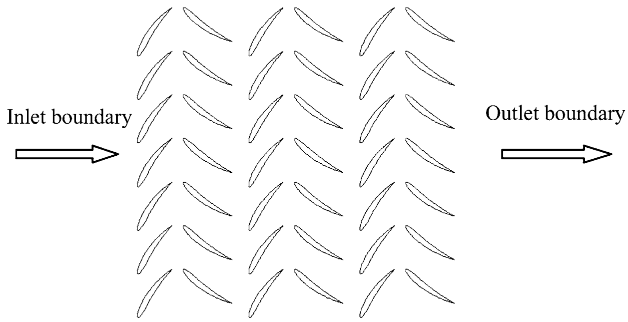


**Fig. 9** Three stages of the cascade blades containing rotors and stators [23]

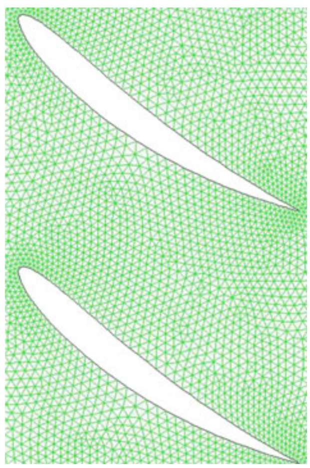

**Fig. 10** Triangle unstructured grids around the blades

**Table 2** The cascade characteristics [23]

| | |
|---|---|
| Stagger angle of rotor blade | 55 |
| Stagger angle of stator blade | 35 |
| Rotor blade profile | NACA65-(A10) |
| Stator blade profile | NACA65-(A10) |
| Solidity | 1.35 |

## Flow Analysis Between Three Stages of a Simple Axial Compressor Cascade

Three stages of a 2D cascade containing seven rotor blades and seven stator vanes were simulated as shown in Fig. 9. The cascade blades characteristics and the steady-state conditions of the fluid flow are presented in Tables 2 and 3, respectively [23]. The commercial software Fluent was utilized for the flow analysis.

Note that the parameters presented in Table 3 have been elaborated in [23], and hence, they are not explained herein. Figures 11, 12, and 13 show respectively the static pressure contours, the Mach number distribution, and the stream lines for the three-stage compressor in its complete

**Table 3** The values of the compressor operating parameters corresponding to the steady-state flow conditions [23]

| | |
|---|---|
| $r_h/r_t$ | 0.6 |
| $R$ | 0.56 |
| $P_{in}$ | 100 kPa |
| $T_{in}$ | 300 K° |
| $P_{exit}/P_{in}$ | 1.07 |
| $\beta_1$ | 62 (°) |
| $\Psi$ | 0.28 |
| $\Phi$ | 0.4 |
| $Vx_{in}$ | 36 (m/s) |
| $M_{in}$ | 0.223 |
| $U_r$ | 90 (m/s) |
| RPM | 1,240 |

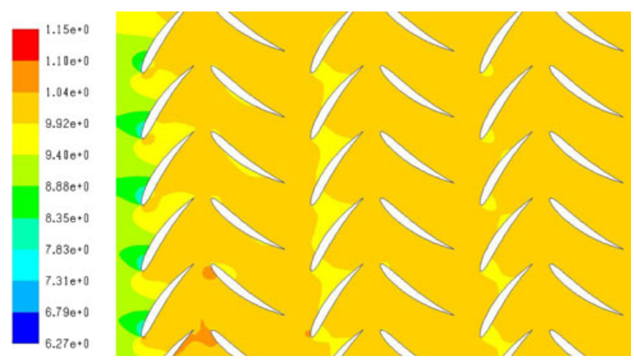


**Fig. 11** Static pressure contours in complete cascade

(i.e., no blade removal) condition. As seen in Fig. 11, no flow separation occurs between the blades.

When removing the entire rotor blades of the middle stage, there are no blades in the gap between the first and the second stator vanes. This can be the result of increasing the incidence angle to second stator vanes. This condition might increase flow separation probability, but as seen in Fig. 14, this possibility seems to be low because the fluid flow travels in its proper path.

Figures 15 and 16 display the velocity contours and the static pressure distribution in the cascade, respectively, when the entire rotor blades of the middle stage are removed. The results showed that the total pressure rise decreased approximately by 9% because of the absence of the middle rotor that could increase the pressure and velocity in the space between the two stators. It is well believed that the diminishing of the total pressure rise will result in decreasing the compressor efficiency and trivially the output power of the gas turbine. Fortunately, no adverse pressure gradient or flow separation is seen in Figs. 15 and 16 and consequently, no instability is expected to occur.

In the next step, the entire rotor blades and stator vanes of the middle stage were removed simultaneously. In this

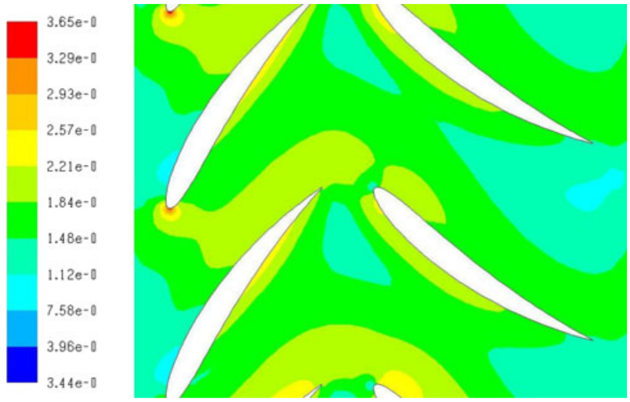


**Fig. 12** Mach number distribution at complete cascade

**Fig. 13** Stream lines in complete cascade

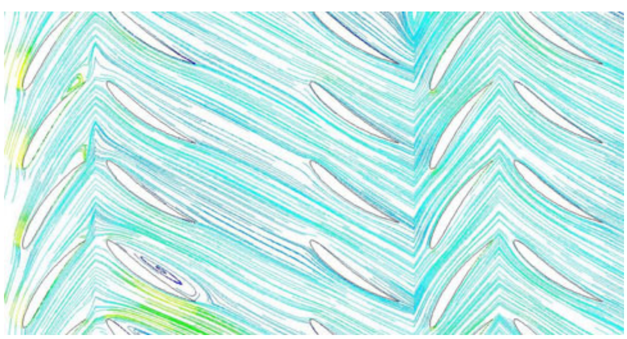

**Fig. 14** Stream lines when removing the entire rotor blades of the middle stage

condition, flow regime is completely changed, and the incident angle raises up to nearly the stall angle. The velocity distribution in the cascade is shown in Fig. 17. The gap in the compressor cascade increases, and the rise in the loss of the total pressure reaches about 23%.

In the final simulation, only two rotor blades of the middle stage were removed. As shown in Fig. 18, there is no uniform flow as in prospect in axial compressor cascade. As seen in Fig. 19, there are many large eddies in the flow, and the rotational stall is seriously expected to occur.

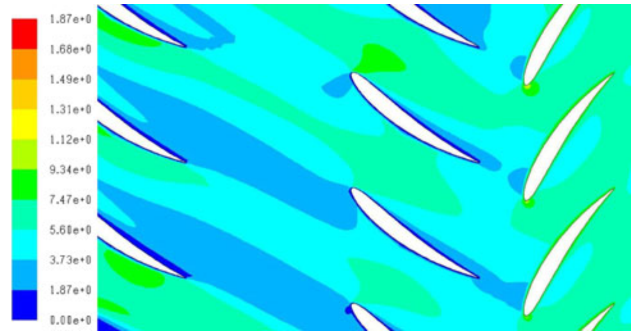


**Fig. 15** The velocity contours when removing the entire rotor blades of the middle stage

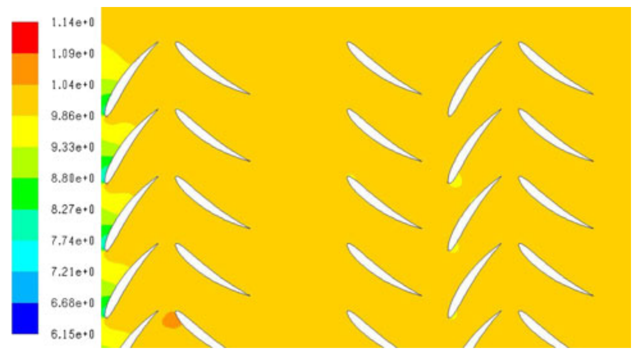


**Fig. 16** Static pressure distribution when removing the entire rotor blades of the middle stage

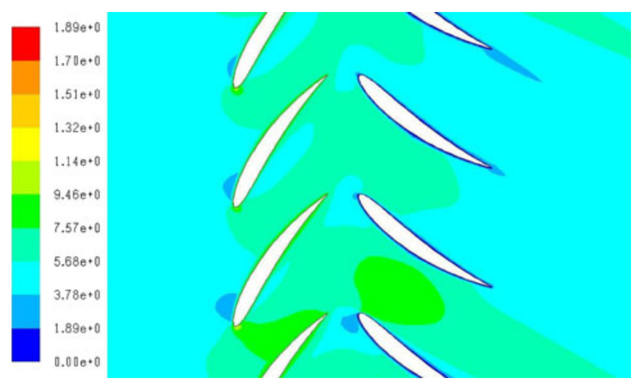


**Fig. 17** Velocity contours in the cascade when removing one stage

It is important to highlight that, although we removed only two rotor blades in the third simulation, it could be expected that removing 39 rotor blades of the stage 11 in the gas turbine compressor studied in the present research will result in more critical fluid flow conditions from the view of eddy formation and instability. As discussed above, between the three different ways of solution suggested in RRC, only the first one (i.e., removing the entire rotor

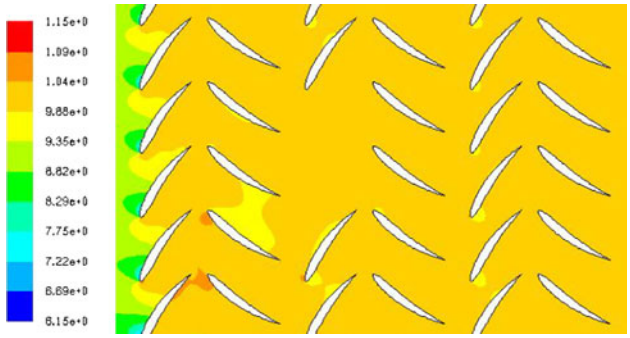


**Fig. 18** Static pressure contours when removing only two rotor blades of the middle stage

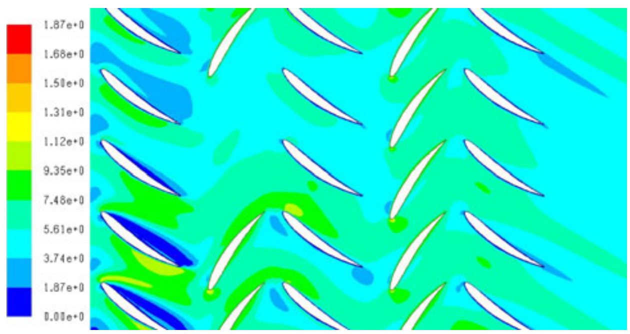


**Fig. 19** Velocity contours when removing only two rotor blades of the middle stage

blades of the stage 11) might be prescribed tolerating the inevitable loss of the compressor efficiency.

## Results and Discussion

Crack nucleation from dovetail regions in the gas turbine compressor disks due to fretting fatigue mechanism, resulting from the wear and tear due to the time that the unit had been in service, is a common failure at the Rey power plant units. One of the most important parameters in controlling the contact stress between the compressor rotor blade and disk, and hence, the crack nucleation life time of the region, is the clearance between these two components such that it must be kept in an allowable range prescribed by the manufacturing company. If the clearance is not carefully checked to be in the permitted range, then the excessive contact between the disk and blade may result in premature crack nucleation at the disk and growth in the crack will lead to disk's final fracture (when the clearance is too small). On the other hand, the larger clearance may result in two adverse phenomena. The first one is the large-amplitude blade vibration and hence the premature blade failure' and the second one is the severe wear of the blade lock pin. Another important event is expected to occur when cracks initiate in dovetail region. It may be that the cracked dovetail cannot apply an adequate reaction force to the blade at the contact area, and hence the crack growth in disk, despite the disk fracture, can lead to the escapement of blade. This may lead to heavy damage in the most of the turbine rotary and stationary parts by the foreign object damage (FOD).

Note that from the viewpoint of engineering design, it is necessary that the cracking resistance of the blade material be higher than that for the disk material. In other words, in addition to the considerations taken in selecting these materials based on the general mechanical and metallurgical properties, it is essential to note that the disk material should be more brittle than the blade material. Under this condition, it is expected that the contact forces between disk and blade result in the deformation of the blade (i.e., the rather ductile material) and the cracking of the disk (i.e., the rather brittle material). This fact is perfectly considered in the design of the gas turbine compressor investigated in the present study. A review of the materials' properties of the unit 32 gas turbine components in Rey power plant revealed that the disk and blade were made of Ni–Cr–Mo–V alloy steel and an alloy steel with 12% Cr, respectively. As given in [24], the disk material has a higher strength, and is more brittle than the blade material (note that the strain to failure and the area reduction in the uni-axial tensile test is considered to evaluate).

Preferring the disk cracking to the blade cracking can be justified thanks to the fracture mechanics as an advanced engineering design tool. Let us consider two imaginary cracks with same lengths, one of which is initiated in the blade, while the other in the disk at the contact region. The crack in blade is located in a position having a high far-field tensile stress level; however, the crack in disk is subjected to a very low far-field tangential stress level mainly generated from the centrifugal forces (see the stress distribution in a centrally hollow rotating disk [25]). Extensive finite element analyses performed previously in the Rey power plant on the disks and blades with a toward-the-center crack and a transverse crack, respectively, in the dovetail regions having the same lengths and subjected to pure mode I (i.e., opening mode) loading conditions, demonstrated that the mode I stress intensity factor ($K_I$) for the blade is meaningfully larger than that for the disk [26]. Taking into account that the values of the plane-strain fracture toughness ($K_{Ic}$) for both the blade and disk materials are not considerably different [26], it can be concluded that the blade fracture is more serious than the disk fracture. It is trivial that blade fracture during gas turbine operation will result in catastrophic damage of the other components via FOD that most of them will not usually be able to be refurbished or repaired to be used again [27].

Note that from a maintenance and repair view point, the gas turbine components (e.g., the compressor disk and blades) should be designed such that their critical crack lengths (i.e., the crack length at the fracture onset) become as large as possible. This is because the possibility of likely cracks being detected during periodic inspections and overhauls enhances, and hence, one can repair or replace the cracked component before final fracture. Since the critical crack length depends on both the far-field stress and the plane-strain fracture toughness ($K_{Ic}$) of material, it is expected that for a constant value of $K_{Ic}$, the critical length of the toward-the-center crack in the disk (initiated from the dovetail region) of the currently studied gas turbine becomes considerably long compared with the disk radius (see also in [16] a crack of about 300-mm length). This is because of the existing very low far-field tangential stress near the disk dovetail region. The evidences in this subject are the failures that occurred in several compressor disks in the Rey power plant; most of which contain long cracks initiated from the dovetails and propagated toward the disk body such that they could be replaced before final fracture [16, 27].

As mentioned earlier, if several rotor blades corresponding to the damaged dovetail regions are removed from the disk, then the fluid flow in the compressor may experience instability leading to wake formation and stall. Since removing simultaneously the entire rotor and stator blades of a stage of compressor would result in decreasing the pressure rise and causing likely stall, it could not be a practical solution to the existing repair problem. The only practical solution seems to be the removal of the entire rotor blades of the stage 11 for which we expected an out-of-instability compressor operation and a tolerable efficiency decrease. Trivially, the compressor efficiency decrease would result in turbine output power loss: approximately 20% in the studied case. A comparison of the output powers obtained from the gas turbine unit before and after the removal of the stage 11 rotor blades revealed that 20% power loss was tolerable in a short period of time with the aim of restarting the unit quickly. It is worth mentioning that the solution effected was prescribed to exit the unit from its idle state temporarily. After restarting the unit, the Rey power plant sent a purchase request to the manufacturing company, and the company undertook delivering a new rotor within the next six months. The RMRC intends to substitute the repaired rotor with the new one to prevent continuous power loss.

It should be finally highlighted that in all of the investigated conditions in the present article, the probability of the stall event is mentioned. It is very difficult to predict the stall event accurately without the availability of the compressor map of the investigated gas turbine machine (the compressor map is the private design information for the producing company which is not usually given to the purchaser). The stall in compressor has direct relation to the fluid flow inside the compressor. As a result, by investigating the flow inside the compressor and the total pressure rise of the compressor, it is possible to anticipate the instability of the compressor partly. In this study, we surveyed generally the probability of the stall event in a simple three-stage compressor model and cited that by removing the entire rotor blades of the middle stage of the compressor, the probability of the stall event is less than in the other investigated conditions. Our conclusions were fundamentally based on the incident angle as well as on the observation of the fluid flow, and we did not consider the other parameters that may affect the stall prevision such as when changing the fluid dynamics location. In the four cited states, the flow path line and the contours of flow parameters were compared together, and we concluded that, among the entire possibilities of blade removals, the prescribed rotor blade removal option has the fewer tendencies for the instability than the other conditions. Although removal of the rotor blades of one stage does not guarantee that it will not make the compressor prone to stalling, it is best to remove the total rotor blades of stage 11 instead of removing only the damaged blades or removing the total stage 11 of the compressor including rotor and stator. It is worth mentioning that this method of repair is not a strong suggestion since it was applied based on only some important fluid flow parameters and not on the comprehensive characteristics of the compressor, which are normally understood from the compressor's map as well as from other scientific literature. It is expected that the presented approach could be used in the similar gas turbine disk failures in the future as a temporary repair method.

## Conclusions

The main conclusions obtained from this research are

1. Micro-fractography using SEM demonstrated that the main cause of crack initiation from the disk dovetail regions was the fretting fatigue phenomenon, and the crack propagation was due to the fatigue loading applied.
2. It was qualitatively found by CFD analyses that only the removal of the entire rotor blades of the stage 11 might be allowable for repairing the compressor by blade removal.
3. The applied repair process led to the output power loss of actually 21% without any severe operational instability in the gas turbine unit.

**Acknowledgments** The authors wish to thank Mr. Eng. Fariborz Teimouri, the managing director of the Rey power plant, for providing the support to this study. They also thank Mr. Eng. Hamid Reza Jafarinia and Mr. Eng. Ali Ghiasipour for their assistance in the field studies. Mrs. Eng. Elnaz Ghorbani is warmly thanked for her help in typesetting and the final review of the manuscript.

## References

1. Strandberg, M.: Fracture at V-notches with contained plasticity. Eng. Fract. Mech. **69**, 403–415 (2002)
2. Shin, C.S., Smith, R.A.: Fatigue crack growth from sharp notches. Int. J. Fatigue **7**, 87–93 (1985)
3. Lazzarin, P., Tovo, R., Meneghetti, G.: Fatigue crack initiation and propagation phases near notches in metals with low notch sensitivity. Int. J. Fatigue **19**, 647–657 (1997)
4. Aratani, M., Knott, J.F.: The growth of short fatigue cracks ahead of a notch in high strength steel. Eng. Fail. Anal. **17**, 200–207 (2010)
5. Ayatollahi, M.R., Torabi, A.R.: A criterion for brittle fracture in U-notched components under mixed mode loading. Eng. Fract. Mech. **76**, 1883–1896 (2009)
6. Ayatollahi, M.R., Torabi, A.R.: Determination of mode II fracture toughness for U-shaped notches using Brazilian disc specimen. Int. J. Solids Struct. **47**, 454–465 (2010)
7. Ayatollahi, M.R., Torabi, A.R.: Brittle fracture in rounded-tip V-shaped notches. Mater. Des. **31**, 60–67 (2010)
8. Ayatollahi, M.R., Torabi, A.R.: Investigation of mixed mode brittle fracture in rounded-tip V-notched components. Eng. Fract. Mech. **77**, 3087–3104 (2010)
9. Carter, T.J.: Common failures in gas turbine blades. Eng. Fail. Anal. **12**, 237–247 (2005)
10. Hou, J., Wicks, B.J., Antoniou, R.A.: An investigation of fatigue failures of turbine blades in a gas turbine engine by mechanical analysis. Eng. Fail. Anal. **9**, 201–211 (2002)
11. Khajavi, M.R., Shariat, M.H.: Failure of first stage gas turbine blades. Eng. Fail. Anal. **11**, 589–597 (2004)
12. Kubiak, J., Urquiza, G., Rodriguez, J.A., Gonzale, G., Rosales, I., Castillo, G., et al.: Failure analysis of the 150 MW gas turbine blades. Eng. Fail. Anal. **16**, 1794–1804 (2009)
13. Shlyannikov, V.N., Iltchenko, B.V., Stepanov, N.V.: Fracture analysis of turbine disks, computational–experimental background of the operational decisions. Eng. Fail. Anal. **8**, 461–475 (2001)
14. Poursaeidi, E., Aieneravaie, M., Mohammadi, M.R.: Failure analysis of a second stage blade in a gas turbine engine. Eng. Fail. Anal. **15**, 1111–1129 (2008)
15. Kermanpur, A., Sepehri Amin, H., Ziaei-Rad, S., Nourbakhshnia, N., Mosaddeghfar, M.: Failure analysis of Ti6Al4V gas turbine compressor blades. Eng. Fail. Anal. **15**, 1052–1064 (2008)
16. Farrahi, G.H., Tirehdast, M., Masoumi Khalil Abad, E., Parsa, S., Motakefpoor, M.: Failure analysis of a gas turbine compressor. Eng. Fail. Anal. **18**, 474–484 (2011)
17. Inue, M., Kuroumaru, M., Tanino, T., Furukawa, M.: Propagation of multiple short-length stall cells in an axial compressor rotor. ASME J. Turbomach. **122**, 45–53 (2000)
18. Day, I.J., Breuer, T., Escuret, J., Cherrett, M., Wilson, A.: Stall inception and the prospects for active control in four high-speed compressors. ASME J. Turbomach. **121**, 18–27 (1999)
19. Silowski, P.D.: Measurements of rotor stalling in a matched and mismatched multistage compressor. GTL report, vol. 221. Gas Turbine Laboratory, Massachusetts Institute of Technology (1995)
20. Sisto, F., Wu, W., Thangam, S., Jonnavithula, S.: Computational aerodynamics of oscillating cascade with evolution of rotating stall. Am. Inst. Aeronaut. Astronaut. J. **27**, 462–471 (1989)
21. Jonnavithula, S., Thangam, S., Sisto, F.: Computational and experimental study of stall propagation in axial compressors. Am. Inst. Aeronaut. Astronaut. J. **28**, 1945–1952 (1990)
22. Choudhury, D.: Introduction to the renormalization group method and turbulence modeling. Fluent Inc., Technical Memorandum, TM-10 (1993)
23. Amanifard, N., Farhanieh, B., Ghorbanian, K.: Numerical investigation of rotating-stall in a stage of an axial compressor with two different approaches. J. Adhes. Sci. Technol. **4**, 39–47 (2007)
24. Maintenance manual of the MHI gas turbine units available in Rey power plant (2010)
25. Boresi, A.P., Schmidt, R.J.: Advanced Mechanics of Materials, 6th edn. Wiley, New York (2002)
26. Research documents of the gas turbine units available in Rey power plant (2009)
27. Failure reports document of the Mitsubishi gas turbine units available in Rey power plant (2008)