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# Effects of temperature gradient on compressor casing in an industrial gas turbine

E. Poursaeidi [a], H. Ghaemi [a,*], M. Charmchi [b]

[a] Department of Mechanical Engineering, University of Zanjan, Zanjan, Iran
[b] Faculty of Mechanical Engineering, Semnan University, Semnan, Iran

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ABSTRACT

The main aim of this work is devoted to numerical study of temperature distribution effects on the distortion of the compressors casing and consequently the fracture of these blades using ANSYS software. The gas turbine operation history shows that the most fractures occur at the beginning of the cold seasons which may corroborate the probability that the compressor casing distortion due to the non-uniform temperature distribution, can lead to touch the blades tip section with the casing and their failures. Results show that the temperature distribution on the compressors casing is non-uniform at the cold season and this causes distortion on the casing. The casing distortion reduces clearance between first-row blades and compressor casing. Hence, it is concluded to modify the cooling procedure to prevent the occurrence of such a harmful problem.


## 1. Introduction

The tip clearance between a compressor rotating blades and its casing has an unfavorable effect on performance, which causes the energy losses. The gradual increase of blade tip clearance over time is a significant cause of compressor performance deterioration. But in modern high efficiency compressor, a critical requirement for optimal performance relies on minimizing radial clearances between the rotating bladed disk and the casing. So the risk of contact between the rotating bladed disk and the casing increases, significantly. In the previous works, scholars have studied the risk of interaction between rotating blade and casing using non-linear dynamics [1–4]. Giovannetti et al. [5] proposed a procedure for the evaluation of clearance reduction using abradable seals. All operating conditions phenomena such as vibrations, thermal mechanical expansion and rotor eccentricity assembly tolerance, were considered in FE modeling of all gas turbine components that contributes to the first stage radial clearance. All full operating cycles for the gas turbine were considered and both thermal and mechanical loads were applied. Analysis of distortions in the inner side of the steam turbine casing, in the presence of 3-D high temperature gradients was presented by McElhaney [6]. Wei et al. [7] simulated the effects of Reynolds number on the tip clearance flow in a certain turbine rotor under a real working condition characterized by high temperature, high pressure ratio and high rotation. Douville et al. [8] experimentally studied the effect of three different tip clearance heights and two Reynolds number on the tip clearance loss of the plat tip platform. An experimental investigation on the tip clearance flow in a low-speed single-stage axial turbine rotor was performed by measuring the distribution and development of the pressure, loss, velocity and turbulence field [9,10]. Ma and Jiang [11] investigated three dimensional

* Corresponding author. Tel.: +98 9128410740.
*E-mail address:* hossein_ghaemi_89@yahoo.com (H. Ghaemi).

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turbulent flow of the tip leakage vortex in an axial compressor rotor which was studied by a three component laser doppler velocimetry. As pressure of the fluid from inlet to outlet of the compressor increases, temperature will increase through the compressor. It is common to expect 300 °C, temperature gradient between inlet and outlet of the compressor. However, any local and one-sided cooling of compressor casing may result an elastic distortion. It seems that the cooling procedure of turbine casing, in the last part of the chamber, fades clearance between compressor blades and compressor casing. GE-frame 6 gas turbine compressor which is an axial kind of turbo machines can be considered as a conic cylinder with 17 rows of rotor and stator blades.

In the current work we face with GE-frame6 gas turbine with less than 10 years operation. According to power plant engineers reports the one set gas turbines were terminated in the beginning of February 2008 because of sudden failure and complete breakdown of compressor. Previously, two set gas turbine were respectively terminated in the middle of September 2006 and November 2007 in a 13-month interval on account of the similar accidents. The purpose of this paper is to find the factor or factors causing the failure.

## 2. Governing equations

The Reynolds stress model (RSM) was selected to solve transport equations. In RSM, the eddy viscosity approach has been discarded and the Reynolds stresses are directly computed. The Reynolds averaged momentum equations for the mean velocity, are listed below:

$$\frac{\partial \rho U_i}{\partial t}+\frac{\partial}{\partial x_j}(\rho U_i U_j)-\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial U_i}{\partial x_j}+\frac{\partial U_j}{\partial x_i}\right)\right]=-\frac{\partial p''}{\partial x_i}-\frac{\partial}{\partial x_j}(\rho\overline{u_i u_j})+S_{M_i} \tag{1}$$

where

$$p''=p+\frac{2}{3}\mu\frac{\partial U_K}{\partial x_K} \tag{2}$$

In Eq. (1) $\rho\overline{u_i u_j}$ is known as Reynolds stresses and must be modeled in order to close this equation. The Reynolds stress model involves calculation of the individual Reynolds stresses $\overline{u_i u_j}$ using differential transport equations. The individual Reynolds stresses are then used to obtain closure of the Reynolds-averaged momentum. The transport equations for the transport of the Reynolds stresses, $\rho\overline{u_i u_j}$ can be written as follows:

$$\frac{\partial \rho\overline{u_i u_j}}{\partial t}+\frac{\partial}{\partial x_k}(U_k\rho\overline{u_i u_j})-\frac{\partial}{\partial x_K}\left(\left(\delta_{kl}\mu+\rho C_s\frac{k}{\varepsilon}\overline{u_k u_l}\right)\frac{\partial\overline{u_i u_j}}{\partial x_l}\right)=P_{ij}-\frac{2}{3}\delta_{ij}\rho\varepsilon+\phi_{ij}+P_{ij,b} \tag{3}$$

where $P_{ij}$, the exact production term, is given by the following equation:

$$P_{ij}=-\rho\overline{u_i u_k}\frac{\partial U_j}{\partial x_k}-\rho\overline{u_j u_k}\frac{\partial U_i}{\partial x_k} \tag{4}$$

The production term due to the buoyancy is

$$P_{ij,b}=B_{ij}-C_{buo}\left(B_{ij}-\frac{1}{3}B_{kk}\delta_{ij}\right) \tag{5}$$

where

$$B_{ij}=g_i\left(-\frac{u_t}{\rho\sigma_\rho}\frac{\partial\rho}{\partial x_j}\right)+g_i\left(-\frac{u_t}{\rho\sigma_\rho}\frac{\partial\rho}{\partial x_i}\right) \tag{6}$$

and $\phi_{ij}$ denotes the pressure–strain correlation.

The pressure strain term can be splitted into two parts:

$$\phi_{ij}=\phi_{ij,1}+\phi_{ij,2} \tag{7}$$

where $\phi_{ij,1}$ is the 'slow' term, also known as the return-to-isotropy term, and $\phi_{ij,2}$ is called 'rapid' term and given by

$$\phi_{ij,1}=-c_1\rho\frac{\varepsilon}{k}\left(\overline{u_i u_j}-\frac{2}{3}\delta_{ij}k\right) \tag{8}$$

$$\phi_{ij,2}=-c_2\left(P_{ij}-\frac{2}{3}P\delta_{ij}\right) \tag{9}$$

Also the equation for $\varepsilon$ can be written as follows:

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\frac{\partial}{\partial x_k}(\rho U_k\varepsilon)=\frac{\varepsilon}{k}(C_{\varepsilon 1}P_k-C_{\varepsilon 2}\rho\varepsilon+C_{\varepsilon 1}P_{\varepsilon b})+\frac{\partial}{\partial x_k}\left[\left(\mu\delta_{kl}+c_\varepsilon\rho\frac{k}{\varepsilon}\overline{u_k u_l}\right)\frac{\partial\varepsilon}{\partial x_l}\right] \tag{10}$$

The Reynolds averaged energy equation is

$$\frac{\partial\rho h_{tot}}{\partial t}-\frac{\partial p}{\partial t}+\frac{\partial}{\partial x_j}(\rho U_j h_{tot})=\frac{\partial}{\partial x_j}\left(\lambda\frac{\partial T}{\partial x_j}-\rho\overline{u_j h}\right)+\frac{\partial}{\partial x_j}[U_i(\tau_{ij}-\rho\overline{u_i u_j})]+S_E \tag{11}$$