Experimental Investigation on Choking and Combustion Products' Swirling Frequency Effects on Gas Turbine Compressor Blade Fractures

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Abstract- *Hydroforming has emerge as an more and more appealing production method for the production of hole bodies. Numerous programs are recognized, in particular within the automotive industry where the trend is increasingly closer to more tricky geometries shaped from tubes and extrusions. In this contribution, a new class of hydroforming strategies is offered. They are characterized by way of the use of sheet metal pairs, consequently permitting an prolonged kind of shapes, but they require unique sealing and docking gadgets. Models, simulations and experiments have targeting the feasibility of 1 precise manner (the hydroforming of un welded sheet metallic pairs) and on the have an effect on of the various parameters at the procedure window as well as on the part geometry.*

Keywords- Sheet metal forming; Hydroforming; Modeling; Finite elements; Hollow bodies

I. INTRODUCTION

With the steady increase of humans' need for energy, the techni-cal development of industrial gas turbines has gained momentum in the past several decades. Technical development aspects such as increased output and efficiency, greater power density, etc., have been the central design goals for gas turbines over the past 25 years.Because the combustion process has the greatest entropy change in every power cycle, it is one of highest heat loss sources in it. Thus, decreasing combustion losses, such as decreasing unburnt fuels, is one of the most common methods of increasing gas turbine efficiency. Using swirling fuel injection, which increases the mixing process, is one of the best ways to achieve this goal. When using the swirling injection of fuel and air in the combustion chamber, if combustion swirl frequencies (CSF) become equal to the natural frequencies of the combustion chamber, it causes resonance in the chamber structure, which is very dangerous for both turbine and compressor blades.

Moreover, the instability of flow inside the combustion chamber depends on many other characteristics such as the flow Mach number in the transition piece area, detonation dependence parameters, and so on. Unfortunately, due to the existence of unfavorable pres-sure gradients along the compressor axis, combustion instabilities can cause stall, or in the worst case scenario, surge problems. Therefore, in order to avoid destructive problems that cause unpredictable failures in engine instruments, a high level of safety should be considered in the design of gas turbine combustion chambers.

There are two distinct aspects in the instability of gas turbine combustion chambers:

- instability of chamber structure
- instability of combustion flow inside the chamber

Instability of chamber structure can cause instability of combustion flow and vice versa. Therefore, investigation of both structure and combustion flow instabilities is necessary for demonstrating the combustion chamber instabilities. Developing an analytical solution for both aspects of chamber instability is very complex due to a variety of operating conditions of the engine and the complexity of geometry and boundary conditions governing it, whereas, numerical studies are very common and suitable due to their abilities in applying complex boundary conditions to complex geometries. In order to study the instability of combustion structure the finite element method can be used; while, for combustion flow instabilities, computation fluid dynamics (CFD) codes are more suitable.Combustion flow instabilities in gas turbine engines can be di-vided into two common distinct parts: choking and detonation. If the flow Mach number in a combustion chamber's exit throat, which is located in the transition piece, reaches 1, the existent acoustic waves push the combustion products into the compressor side and cause a backflow inside it. This situation is called choking. When this happens, the compressor's custom function is disordered and the inversed flow is seen as a moving pressure wave toward the compressor entrance throat. Detonation is the instability of combustion process during the startup operation of the engine.

There are suitable engineering studies published in the literature that investigate the combustion instabilities in gas turbine combustion chambers. Paxson and Quinn [1] performed a comprehensive study of the effect of pressure waves on the acoustic modes of the combustion chamber and how it instigates various forms of instability. In addition, Paxson [2] studied the effect of abrupt change of area in a lean premixed combustor rig and how this change of area expedites the phenomena of combustion instabilities. Schuermans et al. [3] used numerical tools to model the gas turbine combustion chamber and experiments to model the response of combustion. Recently, Schmitt et al. [4] studied the interaction of turbulence, acoustics, and combustion on a scaled gas turbine combustor with a single burner.

Fig. 1 Failed blades in first row of four frame-6 gas turbines

Resonance is the other most common and often overlooked instability in gas turbine chambers; it happens when one of the structure’s natural frequencies become equal to its CSF. A few papers and studies have been published that address this aspect. Caraeni et al. [5] developed a fast method, based on the Arnoldi algorithm, to determine the resonance frequencies of a combustion chamber. More recently, Bethke et al. [6] determined the resonance frequencies using finite element-based methods.Numerous reports of fractures in first-stage moving compressor blades in frame-type gas turbines, such as in plants in Pakistan, China, Korea, USA (New Jersey and Newark Bay) [7] and recently in Iran, show that similar fractures are due to compressor design errors. Due to the importance of this problem, conferences have been dedicated to examine the issues with frame-type com-pressors [8]. Premature fracture of a compressor blade after about 18,000–31,500 hs of operation occurred in four gas turbines at a seaside refinery in Iran [9]. As shown in Fig. 1, all of the fractures occurred in the first-stage moving blades.

All the fractures occurred during normal engine operation; thus, it was clear that detonation had not occurred and therefore it was not investigated. Vibration gauges show high amplitudes during a brief period of time before accidents; these can be due to combustion chamber resonance or perhaps choking. Therefore, in order to distinguish chamber resonance probability, CFD analysis of combustion and finite element modeling of the chamber structure were done to calculate CSF and chamber natural frequencies, respectively. Moreover, Mach number distribution inside the chamber was studied to determine choking probabilities. The results are presented in this paper.

II. CASE-STUDIED GAS TURBINES

Frame-6 heavy duty gas turbines are formed in 17 stages of compressor disks and 3 stages of turbine disks, as shown in a side- view in Fig. 2

The combustion chambers of the gas turbine are of can-annular type. Each turbine has ten combustion chambers that are located around the rotor between the compressor and the turbine section.

The chamber must be in a through-flow position. It is designed so that the air near the nozzle stays close to the front wall of the liner. Figures 3(a)and 3(b)show the combustion chamber and its arrangement around the turbine rotor, respectively.Air and fuel flow rates during operation of the engine and other geometrical values that have significant effect on chamber instability are reported in Sec. 4

III. FINITE ELEMENT MODELING

Modal analysis was done to determine the combustion chamber’s vibration mode shapes and natural frequencies. First of all, a complete 3D model of a combustion chamber including the fuel nozzle collar, liner, spring seal, liner stops, liner cap, and transition piece was constructed with Solid-Works software as shown in Fig. 3(a).The study was continued utilizing the ANSYS finite element code. The generated FE model consists of 121,407 quadratic elements that were defined in a Cartesian coordinate system. Mean-while, the three first natural mode shapes and frequencies were calculated via modal analysis (Fig. 4). The modal analysis basically solves the simplest vibration equations, which do not include the damping ratios. In other words, the calculated local deformations (mostly epicycloid instead of circular in shape) in the modal analysis that are shown in Fig. 4are qualitative, but the magnitudes of natural frequencies are in exact values due to the independency of natural frequencies on the damping ratios

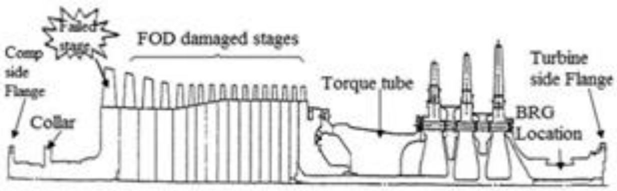

Fig. 2 Side view of the turbine

Due to the low thickness of combustion liners, they are often susceptible to resonance by the stimulation of swirling flow inside the combustion chamber. In order to avoid this problem, protective rings are used to increase chamber stability. Using protective rings in gas turbine combustion chambers substantially increase the chambers' risk of resonance in the first three natural modes, in which their magnitudes are about 331, 339, and 369 Hz, respectively.

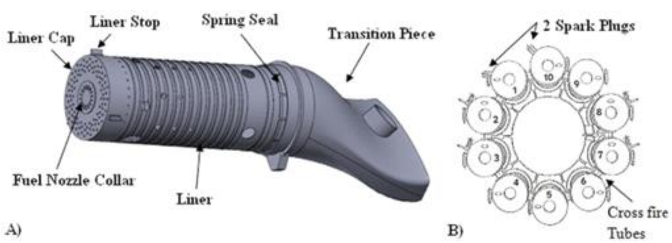

Fig. 3 (a) Schematic of the combustion chamber; (b) arrangement of the chambers on the rotor

IV. THEORETICAL ANALYSIS OF CSF

When air and fuel are swirled into the combustion chamber, the tangential momentum of swirling flow causes a tangential shear force in the chamber liner. Thus, the source of swirling of combustion products is due to the tangential momentum of the swirl vane flows inside the chamber, which varies due to operation load and condition of the engine. As the flow goes to the chamber outlet, the swirling energy is decreased due to the existence of chamber wall shear stress and interaction of secondary air inlet flows with the swirling flow; therefore, studying resonance probability in a gas turbine's combustion chambers is a critical issue both practically and pedagogically; therefore, having an exact analysis of swirling flow characteristics helps designers to have lower cost, more stable combustion chambers.

Theoretical calculation of CSF is not suitable because of existence of the complex boundary layers at the chamber walls and increasing combustion product velocity due to combustion reactions; therefore, numerical analysis must be used for it. The purpose of theoretical analysis of CSF is to determine and estimate its source/s and magnitudes, respectively. Distribution of air flow in the combustion chamber air inlets is shown in Fig. 5.About 7% of compressor outlet air flow is used for turbine buckets and nozzles cooling, and 93% enters the combustion chamber; its distribution is shown in Fig. 5. About 58% of combustion air is used to form an air blanket around the burning gases and to dilute the temperature, 12% of which enters from swirl vane injectors and 30% of which enters from the inlets on the conic shape plate (Fig. 5). The combustion chamber has 16 swirl vanes with an angle of hs¼ 35 deg with respect to the tangential coordinate. An inner view and position of fuel injectors in the swirl vanes area are shown in Figs. 6(a)and 6(b), respectively.
There is a need to characteristics of combustion chamber inlet flow in order to calculate CSF variations under various engine operation loads. Figure 7shows the characteristics of combustion chamber inlet flow in miscellaneous engine operation loads. Combustion fuel momentum is negligible in comparison with huge air momentum and thus its effects on CSF under every operating load conditions can be neglected.

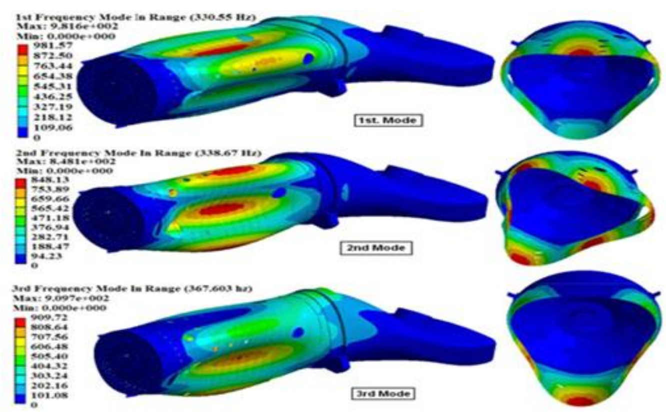

Fig. 4 Three first natural mode shapes and frequencies

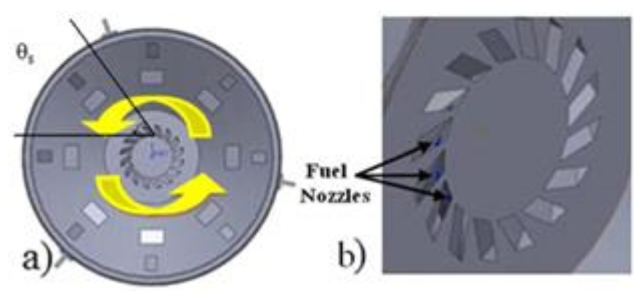

Fig. 5 Air inlet distribution along the combustion chamber

Distance of swirl vane tips from the centerline of the chamber reads as R_s¼ 0.275 m and the sum of their area in each combustion chamber (injector tips area per each chamber) reads S_a¼ 0.0011844 m^2. Injectors' air velocity, V_a, is given by:

V_a ¼ Air flow rate=ðq S_a10Þ (1)

Consequently, CSF can be derived as follows:

CSF ¼V_a Cos h_s=2p R_s (2)

CSF was calculated under various engine operation load conditions as shown in Fig. 7.Regarding Fig. 7, as gas turbine operating load increases, the compressor outlet temperature, pressure, and air flow increase. However, CSF, which was calculated from Eq. (2), decreases from 0 to 15 MW and then increases by increasing the operation load. Calculated CSF varies from 80 to 102 (Hz) in miscellaneous operation loads.

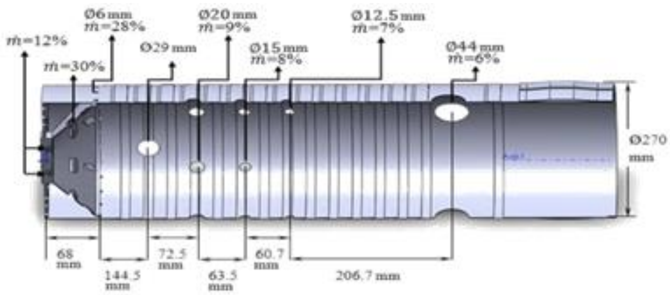

Fig. 6 (a) Chamber inner view of swirl vanes; (b) fuel injectors

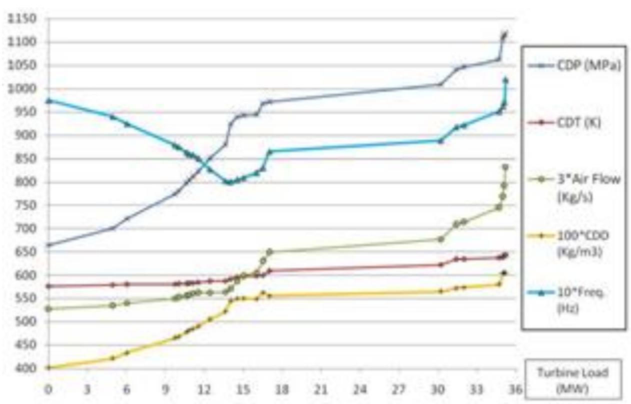

Fig. 7 Characteristics of combustion chamber inlet flow in various operation load of engine

According to calculated natural frequency of chamber structure using ANSYS software and results of theoretical calculated CSF, there is no probability of resonance for the combustion chamber under these conditions. But, as mentioned above, theoretical calculated CSF is not exact and it should be calculated using exact CFD analysis, which is under consideration in Sec.5

V. CFD ANALYSIS

To demonstrate fuel characteristics, Table 1shows natural gas characteristics during combustion [10]. A commercial FLUENT soft-ware package [11] was used to simulate combustion, fluid flow, and heat transfer inside the chamber. The constructed chamber ge-ometry was covered with an unstructured hybrid tetrahedral/ wedge by three different meshes (749,983, 883,857 and 1,065,790 cells), and a parametric analysis was accomplished to confirm whether the numerical results were grid-independent or not. After determining mesh independency, the medium-size mesh (883,857 cells) was chosen for the simulations. Mesh density is refined in the neighborhood of fuel and air inlet sections, being progressively decreased inside the chamber.Because 14 MW is the most common operating load in South Pars Gas Co. refineries 2 and 3 and all the fractures have occurred under these operating conditions in the aforementioned refineries, it was chosen for the simulation. Furthermore, some arbitrary operating conditions were simulated in order to determine how CSF varies versus various operating loads. The nonpremixed combustion with the PDF model was used to simulate the chamber under these two different operation load conditions. To calculate the fields of velocity, pressure, and temperature the discretized conservation equations for mass fractions, momentum, and energy were solved. Flow was assumed to be stable and the segregated.

Table 1 Fuel gas analysis

Analysis SCM: standard cubic meter	Unit	Gas specification limit	Quantity @ 30-Sep-2005
C1	%mol	Min 82	86.67
C2	%mol	Max 12	5.35
C3	%mol	Max 4	1.94
C4	%mol	Max 1	1.00
C5p	%mol	Max 0.4	0.03
CO2	%mol	Max 2	1.30
N2	%mol	Max 5.5	3.71
Gross heating value	MJ/SCM	Min 35.6	37.30
H2S	mg/SCM	Max 4.5	1.68
Total sulphur	mg S/SCM	Max 96.4	22.10
Mercaptans as sulfur	mg S/SCM	Max 14.2	11.08
Water content	mg/SCM	Max 103	8.00
HC dew point	C	Max 5 C @ 55 barg	33 C @ 85 barg

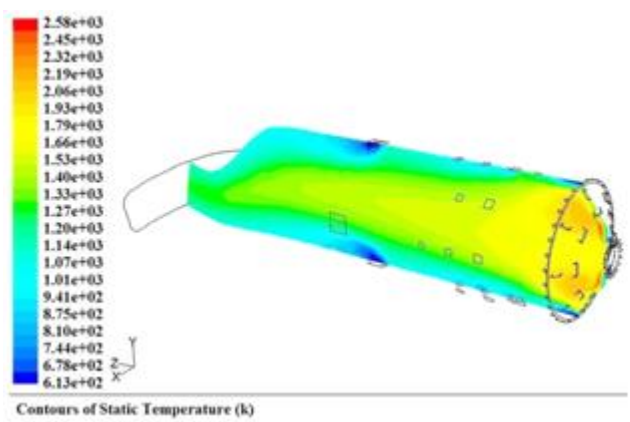

Fig. 8 Contours of temperature in a vertical plane inside the combustion chamber under 14 MW operation.

solver was used because the instability of every conservation equation does not affect other equations' stability.Turbulence closure was achieved by means of the standard k-e model [12,13]. For boundary conditions, mass flow rates, tempera-ture, and mixture fractions were fixed on the inlet sections [11]. The liners of the chamber were fixed to be thermally insolated. The con-stant value of the roughness ratio of the walls (ratio of roughness height to the length scale of geometry (hydraulic diameter of the chamber) was set at 0.05. The pressure outlet condition was chosen for outlet boundary and the gauge pressure was set at 9 bars.

The homogeneous combustion gas phase was modeled using the equilibrium hypothesis while applying rich flammability limits to evaluate the nonequilibrium reactions as a partial equilibrium model [11]. Instantaneous mass fractions were determined in terms of the instantaneous mixture fraction. Mean mass fractions of fuel, oxidizer and products were obtained from the mean and variance of instantaneous values, assuming a b-PDF distribution. For pressure-velocity coupling, the SIMPLE method was used and the second order upwind differential scheme was employed to ap-proximate the convective terms. The solutions were considered to be converged when the sum of the normalized residuals for each control equation was on the order of 1 10 ^{6}. All the models and procedures used in FLUENT software to solve a various ranges of fluid mechanics and heat transfer problems are clearly explained in the FLUENT6.3.26 User Manual [11].

Figure 8presents contours of temperature for a vertical plane in the chamber under the 14 MW output power condition. The large amount of heat released by oxidation of the fuel is pointed out by the near-red regions at point where the flames originate near the injectors. As the flow moves to the outlet, the heat is exchanged with the air that is injected from the liner holes, creating the temperature gradient shown in Fig. 8. The contours of temperature along the length of all the liners of the combustion chamber shows that the temperature levels increase at first, reaches to their maxi-mum, and thereafter decreases. Regarding Fig. 8, using swirling vane injectors to form swirling flow in the combustion chamber makes mixing processes faster and consequently combustion improves.

Figure 9shows the contours of the Mach number for a vertical plane in the chamber under the 14 MW output power condition. As shown in Fig. 9, the Mach number of flow inside the burner is less than the unit at the transition piece area and thus, the flow is subsonic in this area.Regarding Fig. 9, the Mach number is more than the unit in prior stages of the chamber. This is due to a high amount of swirling energy in the swirl vane flow and the intensity of the combustion reaction in this area, which causes a huge swirling momentum. As the flow goes to the outside of the chamber, the Mach number is decreased through the entrance of the transition piece area. The increase in the flow Mach number in the transition piece area is due to special geometry of this area which should increase the flow velocity in order to use it in the turbine. It should be considered that, during starting and stopping of the engine, the combustion flow properties are more unstable than the steady operation condition of the gas turbine. Therefore, more attention must be paid to the starting and stopping operation condition.

In order to identify CSF, Eq. (2)was calculated in the chamber domain. Figure 10shows the contours of CSF for a vertical plane in the chamber under the 14 MW output power condition.As shown in Fig. 10, CSF inside the chamber near the swirl vanes of fuel injectors is much greater than in other regions. For investigating the effective CSF, histograms of frequencies were calculated and the results showed that the 200 Hz frequency covers about 70% of the flow very close to the chamber wall. Therefore, it is clear that the effective CSF is about 200 Hz under 14 MW output power of the engine. Simulations show a range of 100 to 250 Hz for effective CSF in variety of engine operation loads.

Comparison of natural frequencies of the chamber structure with the CSF values (acting as exiting force frequency) show that design of chamber have been suit and there is no possibility of resonance for the chamber during the normal operation of the engine.

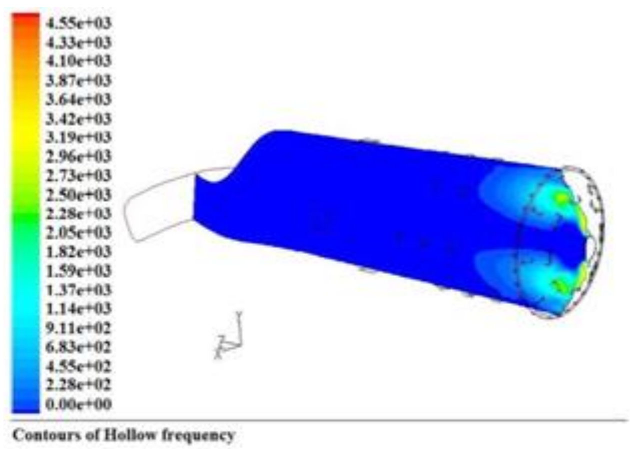

Fig. 9 Contours of Mach number in the combustion chamber under 14 MW operation

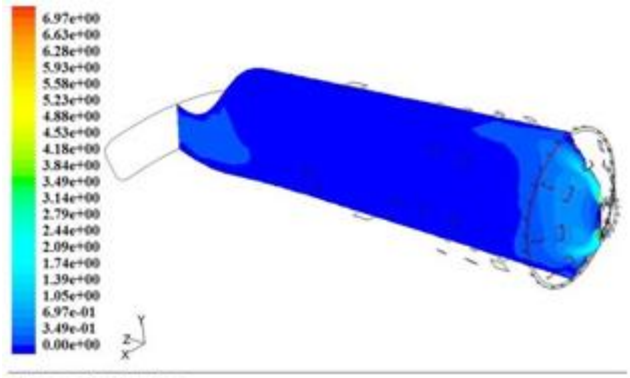

Fig. 10 Contours of hollow frequencies in the combustion chamber under 14 MW operation

VI. RESULTS AND CONCLUSIONS

The problem of premature fracture failure of blades occurred in four gas turbine compressors at a refinery. In order to determine the probability of combustion instability effects on the failure of these blades, a 3D model of the combustion chambers was studied with both finite element (FE) and CFD codes.Distribution of Mach number inside the combustion chamber showed that flow is subsonic in the transition piece area of the chamber. Due to the existence of supersonic flow conditions in prior stages of the chamber; e.g., the zones near the swirl vanes, some critical unpredicted conditions can cause supersonic flow in the transition piece area. Moreover, the operation of the combustion chamber during the starting of the engine is more unstable than steady load operation and thus more attention must be paid during the starting periods. It was concluded that choking cannot be a cause of compressor blade failure in common steady opera-tion of these engines.

Results of CFD simulations showed that CSF varies from 100 to 250 Hz during the normal operation of a gas turbine. Comparison of chamber structure natural frequencies by CSF show that there is no possibility of chamber resonance under range of operation load conditions. Furthermore, according to the simulation results listed above, it can be concluded that the design and manufacturing of this kind of combustion chamber is suitable for improving combustion mixing in the chamber.

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