3D Finite Element Modeling and Vibration Analysis of Gas Turbine Structural Elements

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Abstract
This paper is dedicated to the topical problem of vibration reliability of gas turbine units. One of the most urgent problems during design and manufacturing of gas turbine units is the provision of reliability and durability of unit's joints by development and application of perspective turbomachines constructions with application of up-to-date CAD/CAE/CAM-technologies. This paper is devoted to finite element analysis of rotors and casings vibration analysis for various gas turbine units.

In the present paper natural frequencies of the gas turbine unit rotor with varying bearings stiffness are analyzed for axisymmetrical model. Transient process of rotor vibrations after 4-th cascade blade breakaway is also analyzed. Finite element modeling and analysis of natural frequencies of full-scale gas turbine turbocompressors models and power turbine spatial model is carried out.

Introduction
At present, the power plant machinery construction development is characterized by the tendency towards the increase of unitary power, the decrease of specific capital expenditure, the increase of operational media temperature, the wide use of combined power sets with vapor and gas turbines.

The rotor is the most responsible component of a gas turbine set, as it transfers mechanical energy from rotor blades to a generator for transforming it into electric energy or transfers mechanical energy for the use in technological processes by other machines. Operating life of a rotor machine, breakdown prevention, productivity accuracy of technological operations implemented by a machine depend, first of all, on the level of rotor vibrations and a pressure values between a rotor and bearings. In order to increase productivity and meet high requirements of reliability, different ways are used for creating rotor machines:

1) Power increase in a single unit without a change of nominal angular velocity of a rotor.

2) Angular velocity increase (e.g. gas turbine transport motors, grinding and polishing machines, turbocompressors, etc).

These tendencies in the rotor machine designing led to the problem of fighting against vibroactivity.

The present work includes the research of dynamic behavior of a turbocompressor rotor of a gas turbine unit as well as the investigation of natural frequencies and vibration modes of rotors combined with various gas turbine units. Among three types of turbocompressor vibrations – bending, torsional and lengthwise – only the first type is decisively important for running turbomachines. So, only bending vibrations were considered in the present work. The hyroscopic moments were not considered.

As the construction of the modern gas turbine sets is sufficiently complex, it was expedient to use approximation methods of computation. In this work the finite element method (FEM) was used – mostly wide-spread nowadays among methods for solving mathematical physics problems. This method presents an effective numerical method for solving physics, engineering and technology problems.

A powerful eigenvalue extraction method for natural vibration analyses of large-scale structures is used. This method is based on a block Lanczos algorithm with shift implementation [1,2,3].
Finite element research of rotor natural frequencies and vibration modes

Investigation of rotor natural frequencies and vibration modes

Let us consider a disco-drum rotor of a turbocompressor of a gas turbine set. The main parts of this rotor, aside from operating blades, are disks and shafts. Disks with operating blade rows, fixed to them, in connection with shafts form a single construction of a turbomachine rotor (See Figure 1). The rotor revolves in elastic bearings, which diminish system rigidity and resonance amplitudes. In order to simplify computations of a turbocompressor rotor, we replace elastic backing-up shanks by equivalent springs with appropriate rigidities.

![Figure 1. 3D rotor model](image)

The rotor contains sixteen stages of a compressor and four stages of a gas turbine. Inasmuch as only lowest frequencies are important, and each stage carries from 30 to 90 blade apparatus, it is possible to homogenize the blades of each rotor stage (See Figure 1) introduce into consideration equivalent density and efficient elastic characteristics of a blade apparatus.

For dynamic computation of a turbocompressor rotor, the ANSYS FE – software was used. Figure 2 demonstrates the FE – rotor model. It is axisymmetric, consisting of 14030 8-node axisymmetrical elements PLANE83, 2 spring-damper elements COMBIN14. The total number of degrees of freedom – NDF = 93900.

![Figure 2. FE axisymmetric rotor model](image)

The following assumptions were made in computations:

1) Rigidities of bearing were assumed to be equal.
2) Casing resonance characteristics were not taken into account.

For selecting an elastic bearing models, which is nearest to real ones, the values of natural frequencies for various methods of bearing simulation (See Figure 3) with experimentally defined rigidity $k'$ were correlated with experimentally determined natural frequencies $F^e_1$ and $F^e_2$ (Table 1).

![Figure 3. Various methods of bearing simulation](image)

<table>
<thead>
<tr>
<th>Fastening variant</th>
<th>Natural frequencies</th>
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<tbody>
<tr>
<td>Figure 3a</td>
<td>$0.98F^e_1$</td>
</tr>
<tr>
<td>Figure 3b</td>
<td>$0.97F^e_1$</td>
</tr>
<tr>
<td>Figure 3c</td>
<td>$1.00F^e_1$</td>
</tr>
<tr>
<td>Figure 3d</td>
<td>$0.98F^e_1$</td>
</tr>
<tr>
<td>Experimental</td>
<td>$F^e_1$</td>
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</table>

Comparing experimental and computed natural rotor frequencies, one may conclude that computed frequencies relating to model $b$ in Figure 3 most closely approach to those obtained by experiments. So, this particular bearing elastic model was considered in further researches.

The first vibration mode of a rotor on absolutely rigid bearings (See Figure 4a) presents a classic beam mode with one lengthwise half-wave. The second (See Figure 4b) and the third (See Figure 4c) modes are a combination of bending mode and coupler vibration mode.

The first natural frequency of vibration of a beam-like rotor model consisted of 34 parts, that was computed with the use of the Rayleigh-Ritz method, differs from that computed with the use of FE-models by 8%.

This result gives evidence of the necessity of conducting a frequency analysis based on the full-scale FE-model.
The picture for vibrations of a rotor on pliable bearings with $k'$-rigidity is principally different. The first mode (See Figure 5a) is a combination of a rigid body translation and the first bending mode. The second mode (See Figure 5b) is a combination of a rigid body rotation and the second bending mode. The third mode (See Figure 5c) is coupler vibration. The fourth mode (See Figure 5d) is a complex interaction of the bending mode of the rotor and coupler vibrations.
Figure 5. Vibrations modes of the rotor on pliable bearings

Figure 6 demonstrates dependence of rotor frequency on bearing rigidity. The results show that a satisfactory frequency offset (10%) out of critical frequency $\omega$ [4] is reached in the range of bearing rigidities below $10^5$ kgf/mm. As a result of the natural frequency analysis, one may conclude that the rotor is bendable, as operational frequency $\omega$ exceed the first bending frequency.

Figure 6. Dependence of rotor frequency on bearing rigidity
For the analysis of centrifugal forces influence on natural frequencies of the rotor on pliable bearings with $k'$-rigidity in Figure 7 the frequency diagram (so-called Campbell diagram) is presented. At this diagram one can see the dependency between dynamical natural frequencies and angular velocity $\omega$ of the rotor. Rays of 4 lower harmonics of the disturbing force are also shown. Crossing of these rays with graphs of dynamical natural frequencies defines critical angular velocities at which resonance occurs for the analyzed type of actuation.

**Figure 7. Campbell diagram of the rotor**

FE-analysis convergence study was carried out on the rotor model with $k'$-bearing rigidity for various numbers of degrees of freedom NDF. Figure 8 presents dependence of the first natural frequency on NDF of FE axisymmetric rotor model, from where one may conclude that practical convergence is reached at NDF>100000. Analogous results take also place in researches of practical convergence of other natural frequencies of a turbocompressor rotor.
A revolving rotor presents a source of vibrations. Such vibrations may be caused by unexpected dynamic actions. Such actions may occur, for example, as a result of an unexpected damage and a break-away of operating blades. It was analyzed a rotor transient process at the break-away of a blade of the fourth stage of a turbine rotor. The following assumptions were made for calculations:

1) Bearing rigidities were taken equal.
2) Disks were assumed to be uniformly heated up to 250°C.
3) Tension of the coupler was not taken into account.
4) An unexpected break-away of a blade causes appearance of a radial unbalance force $F$.
5) Damping coefficient corresponding to a half-energy loss per vibration period is taken from experimental data.

Rotor movement comprises two vibrating movements: transient & steady state. The first stage of this movement (several starting cycles, during which free vibrations participate in movement) is related to a transient mode. Owing to damping, after a short time interval, free vibrations disappear, and remains only a steady process of forced vibrations constantly sustained by the action of the exciting force. Most dangerous parts of a rotor, i.e. four turbine stages, are considered below.

Figure 9 shows dependence on time of a radial displacement amplitude at the place of exciting force application, i.e. at the place of a blade break-away.

For the unsteady process, the tensile stress field at the time instant $t_{\text{out}}$ is shown in Figure 10. The average tensile stress at the joint between the fourth turbine rotor stage and its right-end part is equal to $2.2 \sigma_{\text{comp}}$, where $\sigma_{\text{comp}}$ – compressive stress in a coupler providing rotor integrity.
Figure 9. Transient process at the break-away of a blade

Figure 10. Tensile stress field for the unsteady process
The tensile stress field for a steady process is presented in Figure 11. The average tensile stress at the butt-joint between the third and the fourth turbine rotor stages is equal to $0.96 \sigma^{\text{comp}}$.

Figure 11. Tensile stress field for the steady process

An analysis of steady state rotor vibrations within the frequency range from $\theta$ to $\omega$ was carried out. It was stated that constrained amplitude is proportional to square of rotation number per minute that may cause appearance of high rotor vibrations at natural frequencies. Continuous work at resonance conditions or at those close to them may lead to an accident. Therefore, it is necessary to determine the dangerous zones of critical rotation number per minute.

Tensile stress amplitudes are presented in Figure 12. Average values of these stresses in turbine joints are indicated correspondingly for $F_1$, $F_2$ and $F = \omega$ (angular velocity).
Comparing tensile stresses with the coupler compressing stress, it was stated that for steady conditions – tensile stress is less than compressing, but for unsteady conditions – tensile stress exceeds compressing, that may cause a rotor breakage.

A harmonic analysis in the frequency range from $\theta$ to $\omega$ shows that it is impossible to operate within the range of first two frequencies because of initiation of intolerable stresses and strains.

**Finite element simulation and research of natural frequencies and vibration modes of gas turbine unit**

An actual problem of vibroreliability of a high-power set is considered. The complex vibration state estimation of the turbo compressor rotor and the casing was fulfilled.

A FE-research of natural frequencies and vibration modes of the rotor combined with the casing of a gas turbine unit (GT-1) turbo compressor was carried out. The main GT-1 components are the following: the converting tube; compressor, turbine, combustion chamber casings; the rotor containing 16 compressor stages and 4 gas turbine stages (See Figure 13). Homogenization of blades for each rotor stage was performed. Masses of unaccounted structural elements in the analyzed model were simulated by inclusion of point masses.

Figure 13 presents the GT-1 FE-model consisting of 14920
8-node axisymmetrical elements PLANE83, 2 spring-damper elements COMBIN14 and 40 dot mass elements MASS21.

Computations were performed for NDF=160827 (NDF – total number of degrees of freedom for FE – axisymmetric GT-1 model).

Figure 14 shows model vibration modes for real bearing rigidity \( k \). The first three modes conform to bending-solid-state rotor vibrations, the fourth one – to bending casing vibration.
Figure 15 presents dependence of model frequencies on bearing rigidity. The data obtained give evidence that satisfactory frequency offset (10%) out of critical frequency $F_{cr}$ may be achieved within the bearing rigidity range below $0.65k$.

Analysis of natural frequencies and vibration modes of gas turbine turbocompressor

The 3D FE-model of the gas turbine turbocompressor (GT-2) is shown in Figure 16. It includes 11315 20-node 3D-elements SOLID95, 8 2-node spring-damper elements COMBIN14, 1190 8-node shell elements SHELL93 and 6 3-node beam elements BEAM4. Natural frequency computations were performed for NDF = 210312.

The results obtained compared with experimentally determined frequency values are presented in Table 2.

<table>
<thead>
<tr>
<th>Natural frequencies</th>
<th>Computed</th>
<th>Experimental range</th>
</tr>
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$F_4 = 0.46 \, F_{cr}$  \hspace{1cm} $F_1^\circ \subset [0.46 \, F_{cr}, 0.48 \, F_{cr}]$

$F_6 = 0.69 \, F_{cr}$  
$F_7 = 0.73 \, F_{cr}$  
$F_8 = 0.82 \, F_{cr}$

$F_2^\circ \subset [0.73 \, F_{cr}, 0.81 \, F_{cr}]$

Figure 16. GT-2 3D FE model

Computed FE-model frequencies lie in the range of experimental frequencies. The analysis performed shows that frequency offset out of critical value ($F_{cr}$) is unsatisfactory at ninth frequency ($F_9 \cong F_{cr}$) that corresponds to bending rotor vibration modes.

Figure 17 presents the first twelve GT-2 natural vibration modes. Proceeding from the data obtained, it may be noticed that not only rotor and casing bending modes, but also axial rotor vibration modes are present (seventh and tenth modes).
Finite elements simulation and research of power turbine natural frequencies and vibration modes

The 3D FE power turbine model is presented in Figure 18. It consists of 5920 20-node 3D elements SOLID95, 4 2-node spring-damper elements COMBIN14 and 323 8-node shell elements SHELL93. Natural vibration computations were performed for NDF = 106884.

The main construction components are: a scroll, a turbine casing, a power body, rigidity ribs, a bearing casing cylinder and rotor containing four turbine stages (See Figure 19). Each stage carries large number of blade apparatus that enables blade homogenization for each rotor stage.
Figure 18. Power turbine 3D FE model

Figure 19. The first vibration mode of the power turbine
The first two modes (See Figure 20,21) comply to rotor rigid body vibrations. The third mode (See Figure 22) is bending mode of bearing casing cylinder vibrations. Figure 22 presents power turbine natural vibration mode corresponding to the first bending mode of rotor vibrations.

Figure 20. The second vibration mode of the power turbine

Figure 21. The third vibration mode of the power turbine
Analyzing the results obtained, one may conclude that a natural frequency offset out of operation frequency $F_{op}$ is satisfactory.

**Conclusion**

Based on the ANSYS FE-software, the computation of natural frequencies and vibration modes of turbocompresors of gas turbine units was performed. The complete vibroreliability analysis of a revolving turbocompressors rotor was carried out.

By comparing computed values of natural rotor frequencies with those experimentally defined, the type of elastic bearing fastening was chosen. As a result of the natural frequency analysis performed, it was established that the investigated rotor is bendable, i.e. its operating frequency $\omega$ does not exceed first bending frequency. It was stated that satisfactory rotor frequency offset out of the critical frequency may be reached in the bearing rigidity range below $10^5$ kgf/mm.

It was investigated the transient process of rotor vibrations at the break-away of the turbine fourth stage blade. The tensile stress field analysis was carried out for both unsteady and steady conditions. Comparing tensile stresses with coupling compressing stresses, it was stated that in the steady condition tensile stress is less in value than compressing, but in the unsteady condition it exceeds compressing one, what might lead to a rotor’s breakdown at turbine disks joints. By the harmonic analysis performed in the frequency range from $\theta$ to $\omega$, it was stated that it is impossible to operate within two first critical frequencies because of exceedingly large stresses and strains.

The analysis of gas turbine unit GT-1 and power turbine vibrations showed that a frequency offset out of critical frequency $F_{cr}$ is satisfactory.

The finite element research of gas turbine unit GT-2 made it possible to state that in this case a frequency offset out of $F_{cr}$ is unsatisfactory at ninth frequency corresponding to rotor bending vibration mode.

**References**