Method of engine structural frame vibrations analysis during fan blade-out

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Abstract. The probability of fan blade destruction is extremely small but not equal to zero. The consequences of fan blade-out (FBO) can be catastrophic. The absence of dangerous consequences during FBO event should be confirmed by expensive tests. This causes urgency of developing accurate computational techniques for engine structural frame vibrations analysis. In this paper, we propose an approach to determining rotor-casing system dynamic response to unbalance associated with FBO. The problem is solved in a nonlinear non-stationary formulation. Rotor-casing contact interaction, blades crown stiffness and changes in rotational speed are taken into account.

1. Introduction

Fan blades of aviation gas turbine engines (GTE) are subjected to significant centrifugal loads. Their working conditions are further complicated by the possibility of damage due to ingress of foreign objects. These factors can lead to complete or partial destruction of the blade and to blade-out.

The probability of a fan blade-out (FBO) is extremely low, but not equal to zero. Regulatory industry documents [1, 2] oblige aviation GTE developers to experimentally confirm the absence of hazardous aftereffects for an aircraft in case of FBO. Blade-out engine test is an extremely expensive task. This determines the relevance of the engine structural frame dynamics analysis methods development in order to ensure successful testing at the first attempt.

The following factors can be identified as the main factors leading to an increase in loads on the elements of the engine structural frame components after FBO:

- The interaction of separated blade’s fragments with a fan casing and succeed blades;
- The deflection of the overhung shaft part and the impact of the remaining fan impeller blades on the casing;
- Almost instantaneous unbalance occurrence in the low pressure (LP) rotor, many times higher than the nominal value;
- The passage of the unbalanced LP rotor through a critical speed when rotational frequency is slowed due to the fuel cut-off and intensive blades-casing contact interaction.

The first of the listed factors appears within the first 0.1 seconds after FBO, while the resonant mode passing can take place in 1-2 seconds. Different time scales make it possible to approximately consider these groups of factors separately, despite the nonlinearity of the system.
The tasks of the blade-out simulation and its interaction with the remaining blades and the fan casing are successfully solved by modern finite element analysis complexes [3, 4]. The main interest in them is confirmation of the casing’s ability to hold destroyed blade fragments.

The response of the rotor to an instantaneous unbalance and the passage of the unbalanced rotor through the resonant mode, with contact interaction being considered, have been studied for several decades and examined both in monographs and modern articles [5-7].

In this work, the focus is on the second group of factors. An approach is proposed to simulate the response of the aviation GTE rotor-casing system to an instantaneous unbalance associated with the FBO, taking into account possible contact interaction and changes in the rotor rotational frequency.

2. Rotor model development

The design model includes a fan casing and a shaft of a LP rotor which have typical structure for modern aviation GTE. The rotor is mounted on three elastic supports and carries a cantilever disk with a set of fan blades. 3D solid model of the design model is shown at figure 1.

![Figure 1. 3D solid model of rotor-casing system (a), rotor bearing planes (b)](image_url)

The rotor model is built using the finite element (FE) method and subsequent static condensation according to the Guyana method [8]. The three-dimensional (3D) FE rotor model is created using four-node shell elements and has 11,1648 degrees of freedom. As condensation nodes, 15 nodes, that lie on the axis and do not belong to the three-dimensional FE model directly, are chosen. These nodes are connected to the nodes of the 3D model through multipoint constraints (MPC) of RBE2 and RBE3 types [9]. Each of the condensation nodes has 6 degrees of freedom. The reduced rotor model has 90 degrees of freedom. Fan blades and LP turbine structural components are not included in the model directly. The point inertial elements are used for their modeling. The rotor supports are modeled by binodal linear elastic elements. 3D FE model with MPC links and condensation nodes is shown at figure 2. MPC are shown by pink lines, condensation nodes are highlighted with green circles. Condensation nodes with point inertial elements are additionally highlighted with orange circles.
Figure 2. 3D FE rotor model, MPC and condensation nodes are highlighted.

The quality of the model obtained using the Guyana static condensation method is determined by the choice of condensation nodes and has to be evaluated. This estimation is made on the basis of calculating the rotor critical speeds using the full and reduced models, table 1.

Table 1. Rotor critical speeds (full and reduced models).

| №  | Mode shape   | Critical speed, RPM | Error, % |
|----|--------------|---------------------|----------|
|    |              | Full FE model       | Reduced FE model |         |
| 1  | 1st bending  | 1514                | 1567      | 3.5     |
|    |              |                     |           |         |
| 2  | 2nd bending  | 4349                | 4356      | 0.2     |

The error in the critical frequencies determined using the reduced model is no more than 4%. That indicates a fairly accurate reflection of the inertial and stiffness characteristics of the rotor by the reduced model and the possibility of using it for dynamics simulation.

3. Contact interaction modelling

The contact interaction of the rotor with the casing through the fan blades is taken into account using the penalty method. The normal component of the contact force $F_n^{cont}$ was considered to be a nonlinear function of the difference between the radial displacement of the rotor node $r_o$, which lies in the plane of contact, and the initial value of the radial gap $\Delta$. The tangential force component $F_\tau^{cont}$ was considered linear proportional to the normal component with the proportionality coefficient $\mu$:

$$\delta = r_o - \Delta, \quad F_n^{cont} = \begin{cases} 0, & \delta < 0 \\ F_n^{cont}(\delta), & \delta \geq 0 \end{cases}$$

(1)

The moment, created by the tangential component of the contact force relative to the axis of rotation, was taken into account:

$$M_x^{cont} = -\text{sgn}(V_A') F_\tau^{cont}(r_{\text{blade}} + \Delta - r_o)$$

(2)

Here $r_{\text{blade}}$ is the distance from the rotation axis to the most distant point at the periphery of the fan impeller, $V_A'$ – the tangential speed component of this point. Contact forces are shown at figure 3, explanations of the speed $V_A'$ calculation are given there, for simplicity the fan wheel is shown as a round disk.
Velocity $V_A'$ can be determined by:

$$V_A' = V_O' + \omega r_{\text{blade}}$$

$$V_A' = -\dot{y}_O \frac{z_O}{r_O} + \dot{z}_O \frac{y_O}{r_O} + \omega r_{\text{blade}}$$  \hspace{1cm} (3)

The contact forces projections on the axis of a fixed Cartesian coordinate system are defined as follows:

$$F_y^{\text{cont}} = -F_n^{\text{cont}} \left( \delta \right) \frac{y_O}{r_O} + \text{sgn} \left( V_A' \right) \mu F_n^{\text{cont}} \left( \delta \right) \frac{z_O}{r_O}$$

$$F_z^{\text{cont}} = -F_n^{\text{cont}} \left( \delta \right) \frac{z_O}{r_O} - \text{sgn} \left( V_A' \right) \mu F_n^{\text{cont}} \left( \delta \right) \frac{y_O}{r_O}$$  \hspace{1cm} (4)

The fan blades compliance was taken into account by specifying the nonlinear elastic characteristic of the blades crown as a dependence $F_n^{\text{cont}} \left( \delta \right)$. The elastic characteristic of single blade was obtained from the solution of a contact problem. The blade was pressed against rigid casing. The radial displacement was set to the blade root section. The dependence of the contact force normal component projection on the radial direction on the displacement value is taken as the elastic characteristic. Fan blade deformed shape is shown at figure 4. The elastic characteristic of the blades crown is obtained under the assumption that it is a system of radially oriented springs, each of which has an elastic characteristic corresponding to one blade.
Figure 5 shows the elastic characteristic of single blade and a crown consisting of 21 blades. Fractures on the blades crown elastic characteristics correspond to an increase number of the blades contacting with the casing.

![Figure 5. Elastic characteristics of one blade (a) and crown of 21 blades (b)](image)

The elastic characteristic has an extremum associated with the type of deformed blade configuration and buckling of the blade. As the radial displacement of the root section increases, the contact zone of the blade with the casing shifts. This leads to an increase in the angle between the normal component of the reaction on the casing and the direction of displacement and to a decrease in the magnitude of the normal contact force projection on this displacement direction.

4. Rotor-casing FBO response

The problem of studying the response of the rotor-casing system to an instantaneous unbalance is solved in a non-linear non-stationary formulation based on the integration of the equations of motion in the time domain. The system of equations describing the rotor motion is supplemented by an equation describing the rotation around its own axis, as a rigid body:

\[
\begin{bmatrix}
[M] \{\ddot{q}\} + [D]\{\dot{q}\} + \{\ddot{\phi}\}[G]\{\dot{q}\} + ([K] + \{\ddot{\phi}\}[C])\{q\} = \{f\}
\end{bmatrix}
\]

Equations (5) are integrated compatibly with the use of an implicit scheme according to the Newmark method. Thus, the contact forces influence on the change in rotor rotational speed is taken into account.

At each integration step, the contact conditions were checked. In the case of their accomplishment, the vector of nodal forces was supplemented by the contact forces. Otherwise, it was represented only by the load from the unbalance.

The stiffness and mass matrices of the system are obtained using MSC Nastran. To integrate the equations of motion in this formulation, a C++ program has been developed. The program correctness has been tested for the case of taking into account only the normal component of the contact force and a constant rotor rotational speed. In this formulation, the problem can be solved using MSC Nastran.

The following parameters were used in simulation: unbalance characteristics (mass and eccentricity) – \(m_{\text{unb}} = 9 \text{ kg}, e = 621 \text{ mm}\), which corresponds to the destruction of one and a half fan blades, rotor rotational speed \(\omega = 3800 \text{ RPM}\), initial gap \(\Delta = 15 \text{ mm}\). Damping matrix was considered proportional to the stiffness matrix, damping level is 7.5% of the critical at maximum rotation frequency. The time point \(t = 0\) corresponds to the unbalance occurrence. The results of the simulation using MSC Nastran and developed program are shown at figure 6.
The obtained values are in good agreement, the error does not exceed 1%, which allows to make a conclusion about the correctness of the program.

The results of the calculation, taking into account both the normal and tangential components of the contact forces, as well as the moment and rotational speed variation for the case of constant fan blades stiffness are given at figure 7. The characteristics of the unbalance, damping and the initial clearance are assumed to be the same as in the first calculation, initial rotational speed \( \omega = 3800 \text{ RMP} \), the blades crown stiffness is assumed to be equal to \( k = 3 \cdot 10^4 \text{ N/mm} \), \( F_{\text{cont}}^{\text{mm}} = k\delta \).

![Figure 6. Rotor vertical displacements at the fan section](image)

Figure 6. Rotor vertical displacements at the fan section

The largest rotor displacements in the fan section occur approximately 0.66 seconds later the unbalance appears and are achieved at the rotational frequency \( \omega \approx 2380 \text{ RPM} \). The highest displacement value is 94 mm. The difference in the frequency of achieving the largest displacements from the 1st critical speed value of the three-support rotor, given in Section 2, is due to the fact that when blade-casing contact interaction accurse the blades crown acts as an additional elastic support. This causes a change of the rotor 1st critical speed.

The most intensive change in the rotation frequency is realized, when the greatest displacements are achieved, which is associated with the adopted model of the contact forces formation and the constant rigidity of the blades crown in this design case.

The simulation results for the case of the blades crown nonlinear elastic characteristic are shown at figure 8. The elastic characteristic shown in figure 4 was used.
Figure 8. Rotor vertical displacements at the fan section (a) and calculated rotational velocity (b) for the case of a nonlinear elastic characteristic of the blades crown

In this design case, the largest displacements are observed approximately 0.49 seconds after the FBO. Their value is 147 mm. The frequency of reaching the greatest displacements is $\omega \approx 1718$ RPM. The change in rotational frequency happens to be almost linear. The absence of an increase in the intensity of deceleration during the passage of a resonance stems to the fact that the elastic characteristic of the blades has a maximum. The greatest value of contact forces, and therefore the braking torque $M_{\text{cont}}$, is limited in magnitude. Higher displacement values compared with the previous case are associated with differences in the elastic characteristic of the blades crown, as well as with a lower rate of the resonant mode passage. After $t = 0.86$ s rotation speed remains constant this corresponds to the rotor displacements in the fan section being less than the initial gap value.

5. Summary and conclusions

An approach to determining rotor-casing system dynamic response to sudden unbalance associated with FBO is proposed in this paper. The problem is solved in a non-linear non-stationary formulation based on the integration of supplemented equations of motion system in the time domain. Blades-casing contact interaction considering fan blades crown flexibility and changes in rotor rotational speed was taken into account. To integrate the equations of motion in presented formulation, a C++ program has been developed. The program correctness has been verified. The simulation results for the cases of the constant blades stiffness and nonlinear blades crown elastic characteristic are presented. In case of constant blades stiffness the most intensive change in the rotor rotation frequency is realized, when the greatest displacements are achieved, which is associated with the adopted model of the contact forces formation. In case of nonlinear blades crown stiffness rotor deceleration curve is almost linear which stems to the fact that the elastic characteristic of the blades has a maximum. The greatest value of contact forces, and therefore the braking torque $M_{\text{cont}}$, is limited in magnitude.

The proposed approach has a practical value and can be applied to assess the impact of the engine structural frame parameters on the values of the forces realized after FBO. As part of further method improvement, it is planned to take into account the flexibility of the fan casing, presence of the abradable coating, as well as improvements in the blades crown elastic characteristic determination. The experience of the authors' calculations, as well as comparison with experimental data, indicate that the rigidity of the blades crown, obtained using the proposed approach, is found to be high estimated. Other sources of rotor deceleration besides contact forces are also planned to be considered.

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