DYNAMIC BEHAVIOR OF A COUNTER-ROTATING MULTIROTOR AIR TURBINE STARTER

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ABSTRACT
This paper concerns the dynamic behavior of an Air Turbine Starter (ATS) used for starting fighter aircraft engines. The ATS is essentially composed of an envelope and two counter-rotating rotors connected by a reduction gear. Prediction, using finite element calculation based on a pseudo-modal method, have permitted to obtain : Campbell diagram and unbalance response. Experiments on the whole ATS are carried out in the operating range and the predictions compare favorably with experimental results.

NOMENCLATURE
- \( F \) : frequency Hz.
- \( N \) : speed of rotation of the turbine, rotor 1, rpm.
- \( N_S \) : nominal speed of rotation of the turbine, rpm.
- \( C_1(\Omega_1), C_2(\Omega_2), C(\Omega) \) : gyroscopic matrices of rotors 1 and 2 and of the assembly.
- \( K_1, K_2, K_S, K \) : stiffness matrices of rotors 1 and 2, of the stator and of the assembly.
- \( F_1(t), F_2(t), F(t) \) : unbalance forces on rotors 1 and 2 and of the assembly.
- \( M_1, M_2, M_S, M \) : mass matrices of rotors 1 and 2, of the stator and of the assembly.
- \( R_{1S}, R_{2S} \) : reaction forces on rotors 1 and 2.
- \( \alpha \) : reduction gear ratio.
- \( \delta_1, \delta_2, \delta_S, \delta \) : nodal displacements of rotors 1 and 2, of the stator and of the assembly.

INTRODUCTION
Air Turbine Starters (ATS) are frequently used for starting the engines of fighter aircrafts, and their dynamic behavior must be well known in order to avoid the possible problems caused by critical speeds due to mass unbalance. It is essential to determine the natural frequencies (as a function of the speed of rotation) which the Campbell diagram indicates as well as the unbalance response. The latter must also be controlled. This paper deals with predictions and experiments on an ATS containing an envelope and two counter rotating rotors connected by a reduction gear. To our knowledge no paper has been published on experiments and predictions of the dynamic behavior of counter rotating rotors with significant gyroscopic effects.

Firstly the ATS and the experimental set up are described, then the equations governing the dynamic behavior of the systems are given. Finally the numerical and experimental results on the air turbine starter are presented, compared and discussed.

DESCRIPTION OF THE AIR TURBINE STARTER
The ATS studied here is presented schematically figure 1.

Figure 1 : Air turbine starter - Functionnal diagram.
It consists of three basic modules:
- the turbine module which is the aerodynamic part of the starter and which transforms the pneumatic energy of the incoming airflow from the auxiliary power unit (APU) into mechanical power,
- the transmission module containing a planetary reduction gear train, a clutch assembly (ring gear) and an output drive, which converts the rotational power generated by the turbine shaft into the torque output required for main engine start,
- the stator which supports the turbine and the transmission module by roller bearings.

The main characteristics of the ATS are:
- Power : 130 kW
- Nominal speed (N_s) : above 60000 rpm
- Mass of the ATS : about 10 Kg
- Reduction gear ratio : \( \alpha = 4.455 \)

The view of the test set up is given figure 2.

EQUATIONS-SOLUTIONS
The finite element model, Figure 3, is based on beam elements for the shafts, rigid disc elements for discs, stiffness for the roller bearings which are isotropic and linear and a reduced stiffness for the stator.

As the geometry of the stator is axisymmetric the value of the reduced stiffness is obtained by using Fourier's serial development and the Guyan's reduction technique.

In practice the equations are obtained in a similar way to that presented in the paper by Berthier, Ferraris and Lalanne (1986).

Equations of rotor 1, turbine D4, can be written as:

\[
M_1 \ddot{\delta}_1 + C_1(\Omega_1) \dot{\delta}_1 + K_1 \delta_1 = F_1(t) + R_{1S}
\]

with \( M_1, C_1(\Omega_1), K_1 \) respectively symmetric mass, antisymmetric gyroscopic and symmetric stiffness matrices. \( \Omega_1 \) is the angular speed of rotation of Rotor 1 and \( F_1(t) \), and \( \delta \) are respectively the unbalance force and the modal displacement vector. \( R_{1S} \) is the vector corresponding to the reaction components of the assembly stator and to bearings acting nodes 15 and 22 where there are two roller bearings.

With the same kind of notation, the equations of rotor 2 can be written as:

\[
M_2 \ddot{\delta}_2 + C_2(\Omega_2) \dot{\delta}_2 + K_2 \delta_2 = F_2(t) + R_{2S}
\]

\( R_{2S} \) is the vector corresponding to the reactions of the assembly, stator and bearings, acting node 36 and 45 where there are two roller bearings.

The equations of the stator are:

\[
M_S \ddot{\delta}_S + C_S \delta_S = -R_{1S} - R_{2S}
\]

Combining equations (1), (2) and (3) allows the reactions to be eliminated and, the following general equations are obtained for the entire ATS:

\[
M \ddot{\delta} + C(\Omega_1, \Omega_2) \dot{\delta} + K \delta = F(t)
\]

where \( M, C(\Omega_1, \Omega_2), K \) are respectively the symmetric mass matrix, the antisymmetric gyroscopic matrix and the stiffness matrix. \( F(t) \) and \( \delta \) are the force and the nodal displacements vectors. In addition, as the rotors are counter-rotating, and due to the connection gears the speeds of rotation are such that:

\[
\Omega_1 = -\alpha \Omega_2
\]

where \( \alpha \) is the reduction gear ratio. Owing to the relation (5) which must be associated with (4), the equation of the motion can be considered to be:

\[
M \ddot{\delta} + C(\Omega) \dot{\delta} + K \delta = F(t)
\]

where \( \Omega \) equals \( \Omega_1 \). The equations (6) are solved, then one obtains:
- the natural frequencies as a function of the speed of rotation, the Campbell diagram and the possible zones of instability.
- the effect of the force due to the mass unbalance.

The order of the system (6) is highly reduced using the pseudo-modal method and the computer program presented by Lalanne and Ferraris (1990). The ATS model uses 45 nodes whereas 10 modes are used for the pseudo-modal reduction.

DYNAMIC BEHAVIOR OF THE ATS
The Campbell diagram presented figure 4 gives the evolution of the natural frequencies of the system as a function of turbine rotation speed.
The influence of the gyroscopic effect appears to be significant. The whirls shown on the figure (FW, forward whirl and BW backward whirl) are those of the turbine, i.e. of the rotor 1. The bearings are symmetric, so the intersection points of the natural frequencies D + E with F = N(rotor 1)/60 = N/60 are not critical speeds.

In fact, only points A and B correspond to the critical speeds due to unbalance on rotor 1. The nominal speed is Ns and then the corresponding critical speeds are N1 = 0.28 Ns and N2 = 0.56 Ns. Point C corresponds to the intersections of the critical speeds with F = N(rotor 2)/60 = N(rotor 1)/60 = N/60. The turbine is in backward whirl but rotor 2 is then in forward whirl, thus the point C corresponds to a critical speed due to unbalance on rotor 2. At this point the turbine rotation speed is N3 = 0.90 Ns.

The results of the Campbell diagram are summarized in Table 1.

<table>
<thead>
<tr>
<th>Position of the mass unbalance</th>
<th>Turbine Rotor 1</th>
<th>Turbine Rotor 1</th>
<th>Ring gear Rotor 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sense of the whirl</td>
<td>Rotor 1 FW</td>
<td>Rotor 1 FW</td>
<td>Rotor 2 FW</td>
</tr>
<tr>
<td>Frequencies</td>
<td>0.0046 Ns</td>
<td>0.0094 Ns</td>
<td>0.0034 Ns</td>
</tr>
<tr>
<td>Critical speeds (N rotor 1)</td>
<td>0.28 Ns</td>
<td>0.56 Ns</td>
<td>0.90 Ns</td>
</tr>
</tbody>
</table>

Table 1: Results obtained from the Campbell diagram.

The mode shapes corresponding to the three natural frequencies are given respectively figures 5, 6, 7.

Unbalance responses are predicted, assuming a modal Q factor of 25. The mass unbalances are 10 g.mm. Firstly the mass unbalance is situated on the turbine disc, and the response amplitudes at the turbine disc position and at the ring gear position are presented respectively figure 8(a) and figure 9(a). The mass unbalance is then situated on the ring gear and the response amplitudes at the turbine disc position and at the ring gear position are presented respectively figure 8(b), and figure 9(b). The critical speeds obtained (0.28 Ns, 0.56 Ns, 0.90 Ns) are obviously the same as those of the Campbell Diagram. Experimental results have then been obtained with the two mass unbalances acting simultaneously on the turbine disc and on the ring gear. Two proximity probes have been used for each measurement plane and no keyphasor has been utilized. The experiments have been performed in industrial situation, the run out has been observed to be not significant and then not measured. The amplitude of radial displacement of the turbine disc and the ring gear are presented respectively figure 10 and figure 11. The critical speeds (0.26 Ns, 0.57 Ns, 0.89 Ns) appear quite close to those predicted and obtained by the Campbell diagram and the mass unbalance responses, figures 8 and 9. There are two reasons for the
difference between the measured and the predicted amplitudes: the value of the Q factor is an estimate, and residual mass unbalances are not taken into account. The mathematical model has been used for increasing the third critical speed to 1.2 Ns. The corresponding experiments are predicted quite well.

**CONCLUSION**

Experiments and prediction have been performed on the dynamic behavior of an air turbine starter and the results compare favorably. Intersections of the natural frequencies with the lines in the Campbell diagram give the critical speeds when the rotor, where the mass unbalance is situated, is in forward whirl compared to its own speed of rotation.

The finite element model is quite satisfactory. It has been used for modifying the ATS dynamic characteristics by increasing the stiffness of the foundation of the turbine shaft and then the highest critical speed.

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**REFERENCES**
