THE APPLICATION OF WHOLE ENGINE FINITE ELEMENT MODEL ON CRITICAL SPEED ANALYSIS FOR THE COMMERCIAL AERO-ENGINE ROTOR

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Coupling effect between stator and rotor plays an important role in aero engine rotordynamic response, and needs to be properly characterized in rotordynamic analysis. The coupling effect can be determined with a 3D whole engine finite element model, which includes the major engine structure characteristics and can capture the modes of casing and coupling vibration between stator and rotor. A mode-sorting process, which is based on the comparison between complex mode shapes and pairs similar mode shapes calculated at two successive rotor speeds, is used to obtain the Campbell diagram. However, since the whole engine model is very complex and there exist a large number of local or global modes, errors may easily occur during the mode sorting, especially when an automated mode-sorting process is carried out with an embedded module in some commercial software. Such errors can lead to incorrect Campbell diagram and false prediction of the critical speeds of the rotor.

In this study, a new method is proposed to overcome the mode-sorting error and provide the correct Campbell diagram and critical speeds prediction based on whole engine modeling. In this method, a preliminary Campbell diagram which may have sorting error of rotor is first obtained with the whole engine model. Secondly, the static stiffness and dynamic stiffness of the stator are calculated through the stator model only. Thirdly, two Campbell diagrams of rotor are obtained based on the static stiffness and the dynamic stiffness of the stator respectively, which act like springs in the finite element analysis. Finally, the preliminary Campbell diagram is compared with the two Campbell diagrams obtained at the third steps, and is corrected accordingly based on the comparison. The correct critical speeds can then be determined based on the corrected diagram.

Finally, the critical speeds predicted through the above procedures are validated by an unbalance response analysis. The study shows that the proposed method is an effective approach to provide correct critical speed prediction based on whole engine modelling.

**Keywords:** commercial aero-engine, whole engine finite element model, Campbell diagram, critical speed

1. Introduction

Flexible rotor and thin shell structure stator casing are used in most modern commercial aero-engines. As a result, the dynamic characteristics of the rotor and the stator casing can have coupling effect, which may play an important role in aero engine rotordynamic response and the critical
speed calculation. Therefore, it needs to be properly characterized in rotordynamic analysis \[^1, 2\].

The coupling effect can be determined with a 3D whole engine finite element model, which includes the major engine structure characteristics and can capture the modes of casing and coupling vibration between the stator and the rotor. A mode-sorting process, which is based on the comparison between complex mode shapes and pairs similar mode shapes calculated at two successive rotor speeds, is necessary to obtain the Campbell diagram. However, since the whole engine model is very complex and there exist a large number of local or global modes, errors may easily occur during mode sorting, which can lead to incorrect Campbell diagram and false prediction of the critical speeds of the rotor.

Traditionally, rotordynamic analysis using finite element method does not include stator components. As such, the stator usually is replaced with springs. There are two kinds of stiffness methods in traditional rotor dynamics analysis. One is called static stiffness method and the other is called dynamic stiffness method. In rotor dynamics analysis with static stiffness method, only the stiffness but not modes of casing is considered, it means that the static stiffness springs are obtained by applying a force at the support and the spring constant is obtained as the force divided by the support displacement. In this method, only rotor modes are represented and no mode coupling between rotor and casing is considered. The advantage of this method is that the Campbell diagram is simple and clear, and the errors in mode sorting are limited and can easily be corrected by changing spin interval or MAC (modal assurance criterion) comparison. With dynamic stiffness method, both stiffness and modes of casing are considered, the way to obtain the spring constant is similar to that for the static stiffness. The only difference is the force is applied with a frequency. Therefore, the rotor support stiffness is changing with rotor speed, especially near the stator resonant frequencies. As a result, the frequencies in Campbell diagram change severely with rotor speed, which leads to significant errors in mode sorting.

In this paper, a new method is proposed to obtain the correct Campbell diagram and critical speed based on whole engine model by taking the advantages of the results of the above mentioned two methods. The method is applied to the critical speed analysis of a commercial aero-engine low pressure rotor. The critical speeds predicted through the method are validated by an unbalance response analysis. The method is simple, effective, and leads to the results with great accuracy.

2. 3D whole engine finite element modelling

The 3D whole engine finite element model of a commercial aero-engine is shown in Fig. 1, which includes the major components of the engine. There are 5 bearings in the engine, with bearing 1, 2 and 5 supporting the low pressure rotor, and bearing 3 and 4 supporting the high pressure rotor. There is no inter-shaft bearing between low pressure rotor and high pressure rotor. The Solid185 element is used in the finite element model. The spring element is used to model the elastic support connecting the rotor to the stator.

![Figure 1. The 3D whole engine finite element model](image)
3. Preliminary Campbell diagram with whole engine finite element model

The preliminary Campbell diagram based on whole engine model is shown in Fig.2. We can see that there are many mode pairs, some are coupling modes. There is a very obvious mode-sorting error in the Campbell diagram in Fig.2 as marked with red circle, and apparently the sorting process made an error by not crossing the line. As such, the critical speeds of the rotor of the whole engine model are difficult to be determined from the diagram itself.

![Campbell diagram with whole engine model](image)

**Figure 2.** The preliminary Campbell diagram based on whole engine model

4. Campbell diagram and critical speed with static stiffness and dynamic stiffness

4.1 The support static stiffness and dynamic stiffness of the rotor

If we use the rotor-support system, instead of the whole engine model, the static stiffness and dynamic stiffness can be obtained by experiment or analysis \([3,4]\). The static stiffness of the stator casing is obtained by the linear static analysis and the dynamic stiffness is obtained by the harmonic response analysis.

The static stiffness and dynamic stiffness of the stator casing are shown in Fig.3, the bearing 1 horizontal direction stiffness of the stator casing is \(K_{1sy}\), the bearing 1 vertical direction stiffness of the stator casing is \(K_{1sz}\), the bearing 2 horizontal direction stiffness of the stator casing is \(K_{2sy}\), the bearing 2 vertical direction stiffness of the stator casing is \(K_{2sz}\), the bearing 5 horizontal direction stiffness of the stator casing is \(K_{5sy}\), the bearing 5 vertical direction stiffness of the stator casing is \(K_{5sz}\). We can see that the dynamic stiffness changes severely with the stator resonant frequencies. The bearing 1 horizontal direction dynamic stiffness and bearing 2 horizontal direction dynamic stiffness are similar, with the peaks or troughs near the 30 Hz, 40 Hz, 50 Hz. The bearing 1 vertical direction dynamic stiffness and bearing 2 vertical direction dynamic stiffness have similar characteristics, with the peaks or troughs near the 25 Hz, 40 Hz, 75 Hz, 110 Hz. The bearing 5 horizontal direction stiffness have peaks or troughs near the 30 Hz, 35 Hz, 60 Hz, 70 Hz, 95 Hz. The bearing 5 vertical direction stiffness have peaks or troughs near the 85 Hz, 95 Hz, 120 Hz. If a mode analysis is applied, these frequencies are the resonant modal frequencies of the stator casing.

![Dynamic stiffness](image)

**stiffness \(K_{1sy}\)**

![Dynamic stiffness](image)

**stiffness \(K_{1sz}\)**
The support of the rotor can be treated as two springs in series, one spring is stator casing, the other is elastic support between rotor and stator, such as squirrel-cage support. The total support stiffness of the rotor can be obtained by the follow formula:

\[
\frac{1}{K} = \frac{1}{K_c} + \frac{1}{K_s}
\]

(1)

Where \(K\) is total support stiffness of the rotor, \(K_c\) is the stiffness of the elastic support, \(K_s\) is the stiffness of the stator casing.

### 4.2 Campbell diagram and critical speed with static stiffness

The Campbell diagram based on the static stiffness is shown in Fig.4. With the static stiffness, only the stiffness but no the frequency of the stator is considered, which means only rotor modes are represented and no mode coupling between rotor and casing is considered. The Campbell diagram is simple and clear, and the critical speeds of the rotor can be determined from the diagram. Here we use the speed ratio to replace the actual speed. As a result, the first critical speed (CS1) is 0.2488, the second critical speed (CS2) is 0.3065, the third critical speed (CS3) is 0.9203.
4.3 Campbell diagram and critical speed with dynamic stiffness

The preliminary Campbell diagram based on the dynamic stiffness is shown in Fig.5, in which mode-sorting errors are indicated. With dynamic stiffness method, both stiffness and resonant frequency of casing are considered. Therefore, the rotor support stiffness changes with the rotor speed, especially at the resonant mode. As a result, the frequencies in Campbell diagram change severely with the rotor speed, which leads to significant errors in mode sorting.

Since obtaining the critical speeds for the Campbell diagram with static stiffness is relatively easy, we propose a method which is based on the comparing the two Campbell diagrams. Comparing the preliminary Campbell diagram based on the dynamic stiffness (Fig.5) with the one with the static stiffness (Fig.4), and also considering the mode shapes, we can use the information from the Fig.4 to obtain the corrected Campbell diagram based on the dynamic stiffness by correcting the mode-sorting errors, which is shown in Fig.6, and the critical speeds are: CS1=0.2529, CS2=0.3208, CS3=0.9010.

5. Modified Campbell diagram and critical speed with whole engine finite element model

Using the above mentioned approach, we can future obtain the corrected Campbell diagram based on the whole engine model, which is shown in Fig.7. The correct critical speeds are determined based on the corrected diagram, which are: CS1=0.2490, CS2=0.3116, CS3=0.8854.

The critical speeds predicted by the above three methods are shown in Tab.1. We can see that the first two critical speeds with the static stiffness method are closer to the results based on the whole engine model than the result with dynamic stiffness, but the third critical speed with the dynamic stiffness should be closer than the result with the static stiffness. Ideally, the critical speeds with the dynamic stiffness are closer to the results with whole engine model than static stiffness results, but the support dynamic stiffness influenced by the damping, which is difficult to set correctly. As a result, the critical speed analysis is influenced accordingly.
Table 1. The critical speeds predicted by the three methods

<table>
<thead>
<tr>
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<th>Static stiffness</th>
<th>Dynamic stiffness</th>
<th>Whole engine model</th>
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<tbody>
<tr>
<td>CS1</td>
<td>0.2488</td>
<td>0.2529</td>
<td>0.2490</td>
</tr>
<tr>
<td>CS2</td>
<td>0.3065</td>
<td>0.3208</td>
<td>0.3116</td>
</tr>
<tr>
<td>CS3</td>
<td>0.9203</td>
<td>0.9010</td>
<td>0.8854</td>
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6. **Unbalance response analysis with 3D whole engine finite element model**

An additional unbalance response analysis is performed to validate the critical speeds prediction based on the whole engine model. The unbalance load is applied at the fan disk, and a harmonic response analysis is performed. The displacement response amplitude is shown in Fig.8. The frequencies corresponding to the unbalance response peaks and the critical speeds predicted by the Campbell diagram based on the whole engine model are shown in Tab.2. The second to the sixth response peaks correlate the critical speeds well, with the errors less than 6%. There are two or more local peaks existing near the critical speeds, which may be induced by the coupling effect between the stator and the rotor and can not be determined by rotordynamic analysis in the rotor-support system. Other low magnitude peaks may be induced by local modes.

![Figure 8](image)

**Figure 8.** The displacement response amplitude

Table 2. The unbalance response peaks and the critical speeds predicted by the Campbell diagram

<table>
<thead>
<tr>
<th>Unbalance response peaks (The critical speeds, errors relative to the critical speeds)</th>
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<tbody>
<tr>
<td>1</td>
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7. **Summary**

In this paper, a method is proposed to eliminate the mode-sorting error and provide the correct Campbell diagram and critical speeds prediction based on whole engine model. This method is applied to the critical speed analysis of a commercial aero-engine low pressure rotor. The critical speeds predicted through this method are validated by an additional unbalance response analysis. The study shows that the proposed method is an effective approach to provide correct critical speeds prediction based on whole engine modelling.
REFERENCES


