Technical Note

VIBRATION SIGNAL ANALYSIS OF MAIN COOLANT PUMP FLYWHEEL BASED ON HILBERT–HUANG TRANSFORM

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Abstract

In this paper, a three-dimensional model for the dynamic analysis of a flywheel based on the finite element method is presented. The static structure analysis for the model provides stress and strain distribution cloud charts. The modal analysis provides the basis of dynamic analysis due to its ability to obtain the natural frequencies and the vibration–made vectors of the first 10 orders. The results show the main faults are attrition and cracks, while also indicating the locations and patterns of faults. The harmonic response simulation was performed to gain the vibration response of the flywheel under operation.

In this paper, we present a Hilbert–Huang transform (HHT) algorithm for flywheel vibration analysis. The simulation indicated that the proposed flywheel vibration signal analysis method performs well, which means that the method can lay the foundation for the detection and diagnosis in a reactor main coolant pump.

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1. Introduction

Vibration monitoring is an important issue for the maintenance and safety of main coolant pumps. Operation data show that the flywheel is mostly prone to fault in actual operation, which affects the safety of the whole nuclear power plant. The critical process involved in vibration monitoring is to extract reliable features representative of the vibration signal of the flywheel.

2. Flywheel modeling

2.1. Modeling environment selection

Before establishing a flywheel finite element model, we need to get a three-dimensional solid model first. For an uncomplicated model, the CAD modeling method is generally applied. The model is then imported into the ANSYS geometry module through the software interface [1]. In this study, the solid flywheel model was established using...
Pro/E, then imported into ANSYS. In order to clarify the entire process, Fig. 1 shows the flow chart of the modeling and analysis.

2.2. **Flywheel structural parameters**

The flywheel is composed of six parts: hub, heavy tungsten segments (HTS), shrink ring, end ring, sleeve, and thrust plate. The sectional view is shown in Fig. 2.

The axisymmetric model can be established by stretching, rotating, and arraying procedure. Fig. 3 depicts the exploded solid model established by Pro/E according to the structural parameters of the flywheel.

2.3. **The finite element model of the flywheel**

The required material parameters are listed in Table 1. Based on the above data, the flywheel finite element model is shown in Fig. 4. In order to improve the accuracy of finite element analysis, we chose tetrahedrons, hex dominant, and sweep methods to mesh the model. There are 75,052 elements and 225,351 nodes in total. In addition, the minimum edge length is $6 \times 10^{-3}$ m. According to the mesh metric data shown in Table 2, we can conclude that the mesh quality is good enough to meet the requirements.

3. **Static structure analysis of flywheel model**

The fixed remote displacement is imposed at the upper and lower thrust plates to act as constraints. The rotational load shown in Fig. 5 is applied to the center of the hub. The displacement and stress distribution cloud charts of the calculation results are shown in Fig. 6, 7. These results indicate that the minimum displacement is $5.02 \times 10^{-4}$ m, which occurs in the center, and the maximum displacement is $1.88 \times 10^{-3}$ m, which occurs in the periphery. At the same time, the minimum stress is 1236.8 Pa located in the center of the lower thrust plate and the maximum stress is $1.78 \times 10^{7}$ Pa located in the keyway of the hub.

There are three obvious faults. One of them is a crack being extended through the whole sleeve (Fig. 7A), while another is identified as the attrition existing in the keyway of the hub.
and finally the third fault is assumed to happen at the angle of heavy tungsten segments which contact the hub.

4. Modal analysis of flywheel model

4.1. The theory of modal analysis

Based on mechanical vibration theory, the motion differential equation for any multiple degree of freedom system is shown in Equation 1:

\[ M \ddot{q} + C \dot{q} + Kq = Q \]  

Where \( M \), \( C \), and \( K \) are mass matrix, damping matrix, and stiffness matrix, respectively [2].

When the system has no external excitation \( (Q = 0) \), the system vibration is free. Then the equation is:

\[ M \ddot{q} + Kq = 0 \]  

If the damping effect is ignored, the system can be considered an undamped free system for analysis. Thus the equation is:

\[ M \ddot{q} + Kq = 0 \]  

Solving the flywheel natural frequencies and mode shapes is equivalent to solving the generalized eigenvalues and eigenvectors of Eq. (3).

4.2. Modal analysis of flywheel model

Structural vibration can be expressed as the linear combination of each order of natural vibration modes, with the lower vibration modes having a greater influence on structure vibration [3]. As a result, the vibration characteristics of structural analysis usually consider the first 5–10 modes. This study extracts the first 10 natural frequencies which are shown in Table 3.

The value of vibration type is a relative value (relative displacement value), which suggests the relative ratio of vibration magnitude of each point at one natural frequency. It
reflects the transportation of the natural vibration frequency, not the actual vibration value [4].

Figs. 8–12 show the vibration mode corresponding to the first to fifth orders of frequency. The first order exhibits shrink vibration, the second and third orders show swinging vibration, while the fourth and fifth orders show thrust plate expansion (see Figs. 9–11).

5. Harmonic response analysis of flywheel model

5.1. The basic theory of harmonic response analysis

Harmonic analysis is used for rotating equipment to determine a structure response under the known sinusoidal (harmonic) load [5]. When the spindle undergoes forced vibration, the main consequence will be vibrations in the spindle drive hub and HTS, and cracks will occur on the contact surface where stress is concentrated (as explained in Section 4).

At this point, the general motion Equation 1 turns into Equation 4 in the harmonic response:

\[
\begin{align*}
\mathbf{u}(t) &= \mathbf{C}_0^{-1} \left[ \mathbf{M} \ddot{q}_1 + \mathbf{C} \dot{q}_1 + \mathbf{K} q_1 + \mathbf{Q}_1 + \mathbf{Q}_2 \right] \\
&= \mathbf{C}_1 q_1 + \mathbf{C}_2 \dot{q}_1 + \mathbf{Q}_1 + \mathbf{Q}_2
\end{align*}
\]

Subscript 1 and 2 express the real and imaginary components, respectively.

5.2. The harmonic response signals

The vibration was measured by accelerometers mounted on the flywheel. Fig. 13 shows one of the amplitude spectra of vibration signals. However, due to the strong noise, the spectrum is quite complex and consists of many peaks. It is difficult to find any strong indication regarding the flywheel condition.

5.3. The HHT of vibration signals analysis

The Hilbert–Huang transform (HHT) consists of two parts: empirical mode decomposition (EMD) and Hilbert spectral analysis (HSA). It is potentially viable for TFR of nonlinear and non-stationary data. In addition, it has produced sharper results and has revealed true physical meaning compared with most other existing techniques in all the cases analyzed [6]. Since EMD decomposes signals into intrinsic mode functions (IMF) according to the order from high to low frequency [7], the applied EMD–threshold approach

<table>
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<tr>
<th>Order</th>
<th>Frequency (Hz)</th>
<th>Critical speed (r/min)</th>
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<tbody>
<tr>
<td>1</td>
<td>1.08</td>
<td>65</td>
</tr>
<tr>
<td>2</td>
<td>197.2</td>
<td>11,832</td>
</tr>
<tr>
<td>3</td>
<td>197.66</td>
<td>11,860</td>
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<td>4</td>
<td>326.23</td>
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<td>5</td>
<td>596.47</td>
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</tr>
<tr>
<td>6</td>
<td>610.49</td>
<td>36,629</td>
</tr>
<tr>
<td>7</td>
<td>612.66</td>
<td>36,760</td>
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<td>8</td>
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<tr>
<td>10</td>
<td>720.47</td>
<td>43,228</td>
</tr>
</tbody>
</table>

![Stress distribution cloud chart.](image)
Fig. 8 – Mode of vibration under the first natural frequency.

Fig. 9 – Mode of vibration under the second natural frequency.

Fig. 10 – Mode of vibration under the third natural frequency.

Fig. 11 – Mode of vibration under the fourth natural frequency.
provides a parameter to control which IMFs shall be considered. The spectra of the 1st to the 10th IMFs of the signal are shown in Fig. 14.

The HHT solution for the response spectra is shown in Fig. 15, which displays the frequency of the highest amplitude. The figure indicates that at 200 Hz, 320 Hz, and 610 Hz the maximum acceleration occurs when the corresponding speed is 12000 r/min, 19,200 r/min, and 36,600 r/min, respectively. The vibration of the thrust plate is the most intense while the central hub is the weakest, but they are basically the same frequency when they reach the peak accelerations.
6. Conclusion

In the end, we came up with three conclusions. Firstly, this paper has obtained the natural frequencies of the flywheel and the first 10 orders of vibration modes. Secondly, the prospect of using spectrum analysis for vibration monitoring of flywheels is emphasized. Finally, the peaks in the spectrum along the harmonic frequencies can be explained by the modulation phenomena related to the local defect of a main coolant pump flywheel.

Conflicts of interest

The author declares no conflicts of interest.

References