Analysis of Unbalance Effect for Super High-speed Spindle using ADAMS®-based Flexible Body Modeling Technique and Spindle Error Analyzer® Experiment

Ki Beom Park
Ji Hun Joo
Jae Min Kim
Mechanical Design & Manufacturing Engineering, Changwon National University
Changwon, 641-773, South Korea
Tel: +82-55-267-1138, Fax: +82-55-263-5221
Email: jojogun@hanmail.net

And

Won Jee Chung
School of Mechatronics, Changwon National University
Changwon, 641-773, South Korea

ABSTRACT

The rotating accuracy of a super high-speed spindle depends on the centrifugal force which can be affected by an unbalance mass. Consequently, rotational balance problem becomes more important in super high-speed spindle. This paper develops a simulation using ADAMS®-based flexible body modeling technique with full configuration of spindle without any simplification. The proposed technique aims at figuring out the unbalance effect accurately, which will be confirmed by experimental results using spindle error analyzer®. The proposed technique can be used for simulating rotational accuracy according to an added unbalance mass and the expecting the experimental result of an actual balancing activity in industrial field.

Keywords: Super high-speed spindle, Unbalance, Vibration, Flexible body analysis, ADAMS®, Rotational accuracy performance.

1. INTRODUCTION

It is well known that a vibration system has an input of dynamic force such as harmonic or impulsive force while its output is exposed to vibration in the form of displacement, velocity or acceleration, as shown in Fig. 1. For a vibration system, resonance phenomenon happens in case an input frequency of the dynamic force comes across a natural frequency of system [1]. Response of the input can be amplified in the vicinity of the resonance frequency, which in turn can increase vibration amplitude. This resonance phenomenon is dangerous and thus should be avoided.

![Fig. 1 Vibration System](image)

In general, a critical speed indicates a speed of a rotation system in the range of resonance frequency. It means that the operating speed of a rotation system happens to be the same as its natural frequency. Several investigators have performed in order to deal with the critical speed by altering natural frequency, adding viscous damping or adding another vibration system.

In the meanwhile, the causes of vibration for a rotation system come from unbalance, misalignment, bent shaft, shaft crack, mechanical looseness, bearing faults, cavitations, rotor rub, electrical motor problems, and so on. [2] Among them, unbalance inevitably exists to some extent for the most of rotation system because of non-uniform density, machining
accuracy, asymmetric shape, impure material, etc. An unbalance mass, \( m \), in Fig. 2 generates the centrifugal force, \( F \), that is proportional to the square of the operating speed, \( \omega \), given by

\[
F = mr\omega^2
\]

where \( r \) denotes the distance of the unbalance mass from the axis of rotation. Consequently the centrifugal force can result in large vibration.

\[\text{Fig. 2 Centrifugal force with unbalance mass}\]

In the industrial field of a rotation system such as a spindle of machine tool, mass redistribution (empirical) method is commonly used to solve balancing problem of spindle. The mass redistribution method is to simply add mass or remove mass to a rotating shaft of spindle. Even this balancing work of mass redistribution method cannot completely rule out unbalance mass. In a recent work [3], active balancing method using experiments was proposed. Chung et al. [4] proposed a simulation technique for rotational balance of a high-speed spindle, in which a spindle was simplified by using ANSYS® and ADAMS® so as to figure out the longitudinal distance of an unbalancing mass on a spindle, based on mass-addition approach and DOE (Design Of Experiment). But this technique has the inconvenience that even a rough model should be made by using ANSYS® and then imported into ADAMS®. [5]

In this paper, we propose an ADAMS®-based flexible body modeling technique with full configuration of spindle without any simplification. This aims at figuring out the unbalance effect accurately, which will be compared with experimental results using spindle error analyzer®. Therefore the proposed technique will be used for simulating rotational accuracy according to an added unbalance mass and the expecting the experimental result of an actual balancing activity in industrial field.

2. ADAMS®-based Flexible Body Modeling Technique of Super High-speed Spindle

The schematic diagram of a super high-speed spindle (40,000 rpm) including a rotating-shaft (encircled by red lines) and a rotor is shown in Fig 3, in which the rotating-shaft with its length of 310mm is supported by 3 front and rear bearings (2 for front and 1 for rear).

\[\text{Fig. 3 The schematic of 40,000 rpm spindle}\]

The super high-speed spindle using ADAMS®, i.e., famous special software for dynamic analysis, has been modeled as shown in Fig. 4. Especially the front and rear bearings are modeled with reality of a designed drawing. The bearing parts are fixed will full constraints. The spindle part can be made to rotate by assigning revolute joint on it. Then giving a motion of function form to the revolute joint can result in the control of rotational speed of spindle.

\[\text{Fig. 4 Modeling of super high speed spindle}\]
In the proposed ADAMS®–based flexible body modeling, it is very important to make the markers of bearings be in coincidence with those of a spindle in order to constrain the bearings to the spindle. Because there are so many node points at the flexible body of spindle, it is not easy to designate markers at exact positions. The procedure of making coincidence of marker position between bearings and a spindle is as follows. First, the positions of bearings should be fixed with a ground part (a reference fixed part in ADAMS®) at the designed drawing point. Second, markers should be assigned on the flexible body of spindle. Last, after we confirm bearing center positions, the marker positions of flexible body should be moved to bearing center positions. This completes the ADAMS®–based flexible body modeling.

The bearing support part is fixed with full constraints. The spindle part can be made to rotate by assigning revolute joint on it. Then giving a motion of function form to the revolute joint can result in the control of rotational speed of spindle. In particular, the spindle should be modeled in a flexible body. If the spindle would be modeled as a rigid body, the effect of rotation cannot be figured out even though any force or torque can be exerted on it. An unbalance mass can be located spindle end position, fixed with spindle.

Table 1 shows the simulations parameters which will be used in the same manner as the experiment conditions in Section 3. The unbalance mass of 0.5g, 1g and 1.5g will be added to the end position of spindle. The rotational speed (unit: rpm (revolutions per minute)) of a spindle changes consecutively by increasing 2,000rpm to 2,000 ~ 28,000rpm.

<table>
<thead>
<tr>
<th>Item</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unbalance mass</td>
<td>0g, 0.5g, 1g, 1.5g</td>
</tr>
<tr>
<td>Rpm</td>
<td>2,000rpm - 28,000rpm (increasing 2,000rpm)</td>
</tr>
<tr>
<td>Factors</td>
<td>Displacement of spindle center</td>
</tr>
</tbody>
</table>

Based on the flexible modeling technique, simulations in ADAMS® have been performed for the super high-speed spindle with 40,000 rpm as shown in Fig. 5.

Figure 6 shows the graphs that illustrate the result of simulation using ADAMS®–based flexible body modeling technique. These simulation results are obtained at the rotational speed of 18,000 rpm. It can be noticed that the displacement of spindle center changes according as unbalance mass increases. In case there is no unbalance mass, the displacement of spindle center is $\pm 4.4 \mu m$. In case there is 0.5g unbalance mass, displacement of spindle center is $\pm 4.8 \mu m$. When there is 1g unbalance mass, displacement of spindle center is $\pm 6.65 \mu m$. In case of 1.5g unbalance mass, displacement of spindle center is $\pm 9.55 \mu m$. Hence it has confirmed that rotational accuracy (the absolute error between the nominal spindle center and the simulated (or measured) spindle center) can be increased as unbalance mass increases. The proposed ADAMS®–based flexible body modeling technique will be shown to be effective in Section 3, compared with the actual measurement of rotational accuracy using SEA (Spindle Error Analyzer®).
In order to confirm the effectiveness of ADAMS®-based flexible body modeling technique, the actual measurement of rotational accuracy using SEA (Spindle Error Analyzer®) has been performed. As shown in Fig. 7, the SEA measures the end position of tool clamped with a spindle by combining its displacement sensors into X- and Y-direction fixed jigs, respectively. In the same manner to the simulation, the spindle rpm changes 2,000 rpm to 28,000 rpm by consecutively increasing 2,000 rpm. The unbalance mass of 0.5g, 1g and 1.5g are attached to the end of spindle. In addition, Table 2 illustrates the experiment parameters for the measurement of rotational accuracy.

### Table 2 Measurement parameters

<table>
<thead>
<tr>
<th>Item</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unbalance mass</td>
<td>0g, 0.5g, 1g, 1.5g</td>
</tr>
<tr>
<td>Rpm</td>
<td>2,000rpm - 28,000rpm(increasing 2000rpm )</td>
</tr>
<tr>
<td>Factors</td>
<td>TIR X, TIR Y, Synchronous, Asynchronous, Total Error Motion</td>
</tr>
</tbody>
</table>

Measurement contents include 5 factors; TIR X, TIR Y, Synchronous Error Motion, Asynchronous Error Motion, Total Error Motion. TIR X and TIR Y indicate the width of each channel signal, showing the whole movement of a spindle. Synchronous Error Motion is the error synchronized with the spindle rotation. Asynchronous Error Motion is the error which is not synchronized with the spindle rotation. Total Error Motion indicates the whole error combined with TIR X and TIR Y.

The rotational accuracy of the spindle, measured by using SEA, is shown in Fig. 8. This figure illustrates TIR X, TIR Y, Synchronous Error Motion, Asynchronous Error Motion, and Total Error Motion as the unbalance mass increases from 0g to 1.5g. The minimum and maximum values of each measurement factor are listed in Table 3.

### Table 3 Result of Rotational accuracy performance

<table>
<thead>
<tr>
<th></th>
<th>0g</th>
<th>0.5g</th>
<th>1g</th>
<th>1.5g</th>
</tr>
</thead>
<tbody>
<tr>
<td>TIR X</td>
<td>10.1-15.38</td>
<td>7.75-20.02</td>
<td>9.69-21.02</td>
<td>22.66-30.77</td>
</tr>
<tr>
<td>Synch</td>
<td>1.34-5.73</td>
<td>2.06-8.47</td>
<td>1.48-7.41</td>
<td>2.9-11.28</td>
</tr>
<tr>
<td>Asynch</td>
<td>4.2-9.56</td>
<td>5.4-10.5</td>
<td>4.66-10.68</td>
<td>10.87-16.52</td>
</tr>
<tr>
<td>Total</td>
<td>6.14-10.44</td>
<td>5.48-11.35</td>
<td>5.65-10.97</td>
<td>11.98-18.15</td>
</tr>
</tbody>
</table>
Figure 9 illustrates the comparison of TIR X and TIR Y with the rotational accuracy of ADAMS®-based flexible body modeling technique at the rotational speed of 18,000 rpm. It is worth noticing that the proposed ADAMS®-based flexible body modeling technique can trace the actual effect of unbalance within the bound of 7 μm, as shown in Fig. 9. Therefore it can be concluded that our proposed modeling technique can be used for analyzing the effect of unbalance a priori.

4. CONCLUSION

In this paper, we have proposed an ADAMS®-based flexible body modeling technique with full configuration of spindle without any simplification. This aims at figuring out the unbalance effect accurately because unbalance inevitably exists to some extent for the most of rotation system. The comparison of actual measurement (TIR X and TIR Y) using spindle error analyzer® with the rotational accuracy of ADAMS®-based flexible body modeling technique has confirmed that our proposed technique can trace the actual effect of unbalance. Therefore the proposed technique can be used for simulating rotational accuracy according to an added unbalance mass and the expecting the experimental result of an actual balancing activity in industrial field.

5. ACKNOWLEDGMENT
6. REFERENCES