Multi-body Dynamic Analysis of Wind Power Gearbox

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ABSTRACT

Using multi-body dynamics theory, we establish a multi-body system model of a 1.5WM wind power gearbox. Based on Hertz contact theory, The meshing force calculation method of gear meshing transmission is presented. This paper makes a simulation study about the time domain characteristics and frequency spectrum characteristics of meshing forces in planetary transmission, analysis the variation curves with time of dynamic load coefficients of multistage planetary transmission. The results show that the contact force in time domain has obvious dynamic characteristics when planetary gear transmission and its frequency domain showed modulation phenomenon. The carrier is the gear meshing frequency and modulation wave frequency is gear rotation frequency.

INTRODUCTION

The wind power increasing gearbox is the key component of the wind turbine which is between the impeller and the generator. The power generated by the rotation of the impeller by the wind force is transmitted to the generator to generate electricity. As the wind power gearbox must meet the larger transmission ratio to meet the rated speed demand of the generator. So, the wind power gear box is multi-stage planetary gear arrangement. Because of the compact structure,
strong bearing capacity and low bearing load of star gear drive, it is widely used in aviation, ship, automobile, military, machinery, metallurgy and other fields. At present, foreign and domestic scholars have studied the dynamics of planetary gear transmission with simple single-stage transmission as the object. Parker et al. think that the lumped parameter model is far from the actual situation, and they used the the semi analytical finite element method to study the dynamic characteristics of the single stage planetary gear mechanism. Sun Tao, Shen Yunwen and so on used the single freedom analytic harmonic balance method to study the nonlinear problem of the backlash in single stage planetary transmission. Considering comprehensive transmission error, manufacturing error, backlash and so on, Kahraman A. studied planetary gear dynamic load characteristics. However, most of the applications are multistage planetary gear drive and planetary gear and other transmission (such as star drive) combination drive. Compared with the single stage planetary gear drive, the dynamic performance of the multistage planetary gear transmission is more complex, and there are many nonlinear factors, such as the gear clearance, the time-varying stiffness and the friction of the gear surface. These factors make it difficult to solve the gear system. Therefore, the dynamics of the multi-stage planetary transmission system has becoming a research focus of the scientific community. But, the research data about multistage planetary gear is very deficient. So, this paper taking a multi class 1.5MW wind gear box as a carrier study its multi-body dynamics characteristics.

**BODY DYNAMIC MODEL OF WIND POWER GEARBOX**

![Figure 1. Model of a 1.5MW class wind gear box.](image)
Figure 2. Schematic diagram of 1.5M wind gear box transmission system.

**MAIN GEAR PARAMETERS OF WINDGEAR BOX**

**TABLE 1. MAIN GEAR PARAMETERS OF GEAR BOX PLANETARY GEAR PARAMETERS OF PRIMARY PLANETARY GEAR.**

<table>
<thead>
<tr>
<th>Gear parameter</th>
<th>Ring gear</th>
<th>Sun wheel</th>
<th>Planetary gear</th>
</tr>
</thead>
<tbody>
<tr>
<td>modulus (m)</td>
<td>13</td>
<td>13</td>
<td>13</td>
</tr>
<tr>
<td>gear number (z)</td>
<td>102</td>
<td>38</td>
<td>24</td>
</tr>
<tr>
<td>pressure angle (a)</td>
<td>20</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>helix angle (β)</td>
<td>4</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>addendum coefficient (ha)</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>clearance coefficient (c)</td>
<td>0.25</td>
<td>0.25</td>
<td>0.25</td>
</tr>
<tr>
<td>gear width (b)</td>
<td>345</td>
<td>345</td>
<td>345</td>
</tr>
</tbody>
</table>

**TWO STAGE PLANETARY GEAR PARAMETERS**

<table>
<thead>
<tr>
<th>Gear parameter</th>
<th>Ring gear</th>
<th>Sun wheel</th>
<th>Planetary gear</th>
</tr>
</thead>
<tbody>
<tr>
<td>modulus (m)</td>
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<td>10</td>
<td>10</td>
</tr>
<tr>
<td>gear number (z)</td>
<td>107</td>
<td>25</td>
<td>41</td>
</tr>
<tr>
<td>pressure angle (a)</td>
<td>20</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>helix angle (β)</td>
<td>4</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>addendum coefficient (ha)</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>clearance coefficient (c)</td>
<td>0.25</td>
<td>0.25</td>
<td>0.25</td>
</tr>
<tr>
<td>gear width (b)</td>
<td>175</td>
<td>175</td>
<td>175</td>
</tr>
</tbody>
</table>
MULTI-BODY DYNAMICS THEORY

In the multi rigid body system used the Descartes algebraic coordinate system to establish the differential algebraic equations:

\[ M(q,t) \ddot{q} + \Phi_q^T(q,t)\lambda - Q(q,\dot{q},t) = 0 \]  

(1)

Among, \( q, \dot{q}, \ddot{q} \in R^n \) are system displacement, velocity and acceleration vector, \( \lambda \in R^m \) is Lagrange multipliers, \( t \in R \) is time, \( M \in R^{mn} \) is inertia matrix of mechanical system, \( \Phi_q \in R^{mn} \) is bound Jacobi matrix, \( Q \in R^n \) is external force vector, \( \Phi \in R^m \) is position constraint equation.

CALCULATION METHOD OF CONTACT FORCE OF MULTI RIGID BODY

Suppose that the object in contact with each other is a rigid body, ignoring the elastic deformation of the object and the clearance of the moving pair, and the contact force can be defined as:

\[
\begin{align*}
Kx^e + F_s(x,0,0,d_e,C_{\text{max}})X & \quad x < 0 \\
0 & \quad x \geq 0
\end{align*}
\]

(2)

\( K \) is Stiffness coefficient, \( X \) is distance variable between two objects, \( e \) is nonlinear index, \( F_s \) is step function, \( C_{\text{max}} \) is damping coefficient, \( d_e \) is breakdown depth at maximum damping.

When \( x \geq 0 \), two objects did not come into contact, and the contact force was zero. When \( x < 0 \), the contact force of two objects is related to stiffness coefficient, deformation quantity, nonlinear index, damping coefficient and breakdown depth. It can be seen that the contact force is divided into two parts:

1. Elastic component \( Kx^e \), Similar to a nonlinear spring.
2. damping component \( F_s(x,0,0,d_e,C_{\text{max}})x \), its direction is opposite to its direction of motion. The analysis shows that the parameters of stiffness, nonlinear index, damping coefficient and the breakdown depth of the maximum damping should be determined first. The stiffness coefficient is related to the material and shape of parts, and is the key to the study of contact force.
STIFFNESS COEFFICIENT OF INVOLUTE HELICAL GEAR

The meshing deformation mainly occurs near the contact zone, ignoring the elastic wave in the object, regardless of friction, and the contact between the rigid bodies can be obtained directly from the Hertz contact theory. For two simple rotating rigid bodies:

\[
\delta = \frac{a^2}{R} = \left( \frac{9P^2}{16RE^2} \right)^{\frac{1}{3}} \frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} \quad \frac{1}{E^*} = \frac{1}{E_1} + \frac{1}{E_2}
\]  

(3)

\( \delta \) is the distance between two objects in contact with each other. \( P \) is load on an object. \( R \) is composite radius of curvature. \( E^* \) is composite elastic modulus. \( \nu_1 \) and \( \nu_2 \) are poisson ratio of two object materials. \( E_1 \) and \( E_2 \) are elastic modulus of material for two objects. \( R_1 \) and \( R_2 \) are the equivalent radius of curvature for the contact point of two objects.

The relation between contact force \( P \) and deformation \( \delta \) can be deduced from the above three formulas.

\[
P = \frac{4}{3} \rho^2 E^* \delta^2
\]  

(4)

\( K \) is depended on the material and shape of the contact object,

\[
K = \frac{4}{3} \rho^2 E^*
\]  

(5)

NUMERICAL SIMULATION AND DISCUSSION OF DYNAMICS SIMULATION

SIMULATION CONDITIONS

Input of gear box is fed with a rotational speed function STEP (time, 0, 0, 0.8, 120d). That is, the speed is within 0.8s, from zero to 120 degrees/s, and remains the same. A constant resistance moment 5000N. M is applied at the output of the gearbox, simulation time 3 seconds. Gear elastic modulus:
\[ E_1 = E_2 = 2 \times 10^5 \text{ N / mm}^2 \]  

(6)

Poisson ratio:

\[ \nu_1 = \nu_2 = 0.29 \]  

(7)

According to formula (5), the stiffness coefficients of gears at various levels are calculated. By the literature collision force index \( e = 1.5 \), damping coefficient \( C = 10 \), \( \text{N} \cdot \text{s/mm} \).

**RESULT ANALYSIS AND DISCUSSION**

As shown in Figure 3, the input and output speed of the wind gear box can be seen that the input speed is 120 degrees / sec, and the output speed fluctuates at an average of 12082 degrees / sec. Therefore, the simulation results show that the transmission ratio is 100.683, and the theoretical transmission ratio is 100.63, and the error is less than 1 per thousand.
The changes shown in Figure 4 the contact force is wind power gear box drive a planetary wheel, seen from the figure in the initial stage due to the speed of movement of the customer is very small, the load torque, the contact force is, with input speed increase and keep unchanged, the contact force curve will maintain stability.

Figure 5 shows the spectrum for Figure 4 contact force curve, from the two high amplitude in the figure: a planetary wheel rotation frequency frequency corresponding to 0.89HZ; the other is a planetary gear meshing frequency frequency corresponding to the 34HZ.
Thus, the contact force of the first planetary gear is the frequency of the planetary gear, and the modulation frequency is the rotation frequency of the planetary gear. Fig. 6 is the dynamic load coefficient curve of the first planetary gear. The variation trend of dynamic load coefficient curve is consistent with the variation trend of contact force.

Figure 7. Contact force at the end of the output varies with time.

Figure 8. Output contact force spectrum.
Figure 7 shows the output of steady wind power gear box when the contact force changes with time, can be seen from the figure in the steady state, the meshing force of gear components containing obvious dynamic contact force, and the size of Rao a certain value fluctuation, show obvious impact vibration of involute gear in meshing process.

Figure 8 shows the spectrum for Figure 7 contact force curve, from the graph spectrum shows that the rotation frequency of the output frequency corresponding to the 33.8HZ end of the small gear; the other one is the output end of the gear meshing frequency corresponding to the frequency of 1004HZ, and also appeared 2 times frequency meshing frequency in 2007HZ at.

Thus, the contact force of the output gear is that the carrier frequency is the meshing frequency of the gear, and the modulation frequency is the rotation frequency of the output gear. Fig. 9 is the dynamic load coefficient curve of the output end. It can be seen from the diagram that with the increase of speed and stability, the meshing frequency of the output end increases as well, and the frequency and amplitude of the gear meshing and impact loading also increase rapidly.

CONCLUSIONS

(1) Based on Hertz contact theory, we established the model of wind turbine gearbox multistage planetary transmission load conditions, gives a method to calculate the dynamic gear meshing force, provides a basis for further analysis of stiffness and fatigue strength.

(2) Gear has obvious dynamic phenomenon in the process of meshing, the contact force at a mean fluctuated, show obvious impact vibration of involute
gears in the meshing process, and the spectrum is in the form of a certain modulation phenomenon.

(3) A new calculation method of dynamic load coefficient of multistage planetary transmission is given, which provides a new way for further research of multistage planetary transmission.

REFERENCES