Dynamics numerical simulation of planetary gear system for wind turbine gearbox

Zhao Yang*a Mutellip Ahmatb & Mamtimin Gena

aSchool of Mechanical Engineering, Xinjiang University, Urumqi, Xinjiang, 830047, China
bSchool of Electrical Engineering, Xinjiang University, Urumqi, Xinjiang, 830047, China

Received 18 July 2013; accepted 30 July 2014

In this paper, the precise 3D geometric models of helical planetary gears system in the wind turbine gearbox are built up. The contact stress curves of planetary gears are analyzed by the nonlinear dynamics and the numerical method. The deformation of the gear teeth and the time-varying mesh stiffness curve of planetary gears meshing inside and outside are got by numerical analysis, the impact stress of the carrier of the helical planetary gears system is analyzed. The results provide a theoretical basis for the structure analysis and optimization of the type system.

Keywords: Helical planetary gear, Nonlinear dynamics, Contact stress, Mesh stiffness

The planetary gear systems are widely used in wind turbines, cars, aircrafts and other areas because of their compact structure, small size, higher transmission ratio and load capacity. Due to a series of nonlinear factors such as assembly errors, tooth flank and bearing clearance, varying mesh stiffness of the gear, tooth surface friction and variable windy load external, when wind turbine planetary gears are meshing, the internal and external excitation such as stiffness excitation, errors excitation and the meshing impact excitation will be produced. According to change of the meshing cycle phase, when the different gears are meshing internal and external, the repeated dynamic impact load will affect the life and reliability of the gear system, therefore the nonlinear dynamic analysis of the planetary gear system should be studied.

The dynamic characteristics of the planetary gear system have been studied extensively, however, most of them are the straight planetary gear, and the input power of the planetary gear carrier is rarely considered. In this paper, the precise 3D model of the planetary helical gear and carrier were established with parametric modeling method, and a multi-gap system was built up. The nonlinear dynamic characteristics of the system were analyzed by considering the tooth surface friction. The force and the deformation of the helical gears system were got, the time-varying mesh stiffness when the gears meshed internal and external was calculated, at the same time, the impact stress of the carrier was analyzed.

FEM Mixed Solution to Impact Contact Dynamics

In reality, the teeth pitches are unequal when the gears are meshing, so actual meshing points will deviate from the theoretical meshing line, while there is a gap resulting in shock and vibration caused by different reasons. The tooth side gap will cause the lag in the impact process; the impact force with the change of the gap with the engagement of the tooth side is concerned1.

Two elastomeric contacted by the dynamic equations2, it can be deduced the tooth impact-dynamic contact equation through the Hamilton variation principle:

$$M_i u_i(t) + C_i v_i(t) + K_i a_i(t) = P_i(t) + F_i(t) \quad (i \in p, g) \quad \ldots (1)$$

Where $M_i$, $C_i$, $K_i$ are the respectively mass matrix, the damping and the stiffness matrix of the active and the driven gear; $P_i(t)$ and $F_i(t)$ are the respectively the active and the driven gear outer load vector, and the contact force vector; $u_i(t)$, $v_i(t)$, $a_i(t)$ are respectively the displacement vector, velocity vector and acceleration vector of the active and the driven gear. The Eq. (1) uses the Newmark-β law, assume that the acceleration of a certain point where it is in the time interval $t \sim t + \Delta t$ looks as the actual acceleration, while the dynamic contact FEM equation with only containing displacement vector $u_i(t+\Delta t)$

$$\vec{K}_i u_i(t+\Delta t) = \vec{p}_i(t+\Delta t) + R_i(t+\Delta t) \quad \ldots (2)$$

Where $\vec{K}_i$ is the effective stiffness matrix; $\vec{p}_i(t+\Delta t)$ is the payload vector.

*Corresponding author (E-mail: zhy86222@163.com)
\[ \tilde{K}_i = K_i + \frac{1}{\beta \Delta^2} M_i + \frac{\gamma}{\beta \Delta} C_i \quad \ldots \quad (3) \]

\[ \tilde{P}_{i(t)} = P_{i(t)} + M_i \left[ \frac{\gamma}{\beta \Delta} u_{i(t)} + \frac{1}{\beta} v_{i(t)} + \frac{1}{\beta} \alpha_{i(t)} \right] + C_i \left[ \frac{\gamma}{\beta} u_{i(t)} + \left( \frac{\gamma - 1}{\beta} \right) v_{i(t)} + \frac{\gamma - 1}{\beta} \alpha_{i(t)} \right] \quad \ldots \quad (4) \]

Where \( \gamma \) and \( \beta \) are the adjusted parameters, the stable of the Newmark-\( \beta \) law is unconditionally guaranteed with \( \gamma=0.5 \) and \( \beta=0.25 \).

Although the two objects may not be enough constraints, but for this type dynamic contact problem, the equation of dynamic contact surface can be directly derived by Eq. (2):

\[ \tilde{f} F_{i(t)} = -\tilde{S}_{p(t)} - \epsilon_0 \quad \ldots \quad (5) \]

Where \( \tilde{f} \) is the effective flexibility matrix of the contact surface; \( \tilde{S}_{p(t)} \) is the ground contact point relative spacing generated by the effective external load \( \tilde{P}_{i(t)} \), \( \epsilon_0 \) is the original clearance for the contact point.

However, LS-DYNA software is an explicit algorithm that use the Newmark recursive formula (order \( a = 0 \), \( \delta=0.5 \)) central difference method, a small error though through software calculations and numerical calculation\(^3\), if numerical calculation was used here will be more complicated, so the finite element software LS-DYNA was used to calculate and analyze.

**Explicit Dynamics Finite Element Solution**

**FEM modeling**

The helical planetary gear system in the gearbox of the 1.5 MW wind turbines was taken as the research object; the geometric model established by the method of parametric modeling in PRO/E, corresponding geometric model entities were assembled in it, leaving a certain gap between the gears and planet carrier, which is shown in Fig. 1. The basic parameters of planetary helical gear system are given in Table 1. ANSYS software was used as pretreatment. The finite element model of system is shown in Fig. 2. The tooth flank clearance is 0.8 mm, which is shown in Fig. 3.

**Table 1—Basic parameters of planetary helical gear system**

<table>
<thead>
<tr>
<th></th>
<th>Number of teeth</th>
<th>Normal module</th>
<th>Tooth width</th>
<th>Pressure angle</th>
<th>Helix angle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ring gear</td>
<td>125</td>
<td>12 mm</td>
<td>300 mm</td>
<td>22.5°</td>
<td>8°</td>
</tr>
<tr>
<td>Planet gear</td>
<td>47</td>
<td>12 mm</td>
<td>300 mm</td>
<td>22.5°</td>
<td>8°</td>
</tr>
<tr>
<td>Sun gear</td>
<td>31</td>
<td>12 mm</td>
<td>300 mm</td>
<td>2 2.5°</td>
<td>8°</td>
</tr>
</tbody>
</table>

Fig. 1—Geometric model of helical planetary gear system

Fig. 2—Finite element model of helical planetary gear system

Fig. 3—Engaging portion of helical planetary gear
The outer surface of the ring gear and inner surface of the planetary gears were defined as rigid body, and the whole helical gears defined as elastic bodies. Due to the small deformation of the carrier, it was defined as rigid body, forming rigid-flexible mixed model. The grid entity was divided by Solid164 unit, but the unit does not have a rotational degree of freedom, the grid of the sun gear of the inner surface and the ring outer surface were shell163 unit, the thickness of the outer surface of the ring gear and the inner surface of the sun gear are 0.1 mm, different parts were defined. Density is \(7.83 \times 10^{-9}\) T/mm\(^3\), and elastic modulus is \(2.07 \times 10^5\) MPa, Poisson ratio is 0.3.

**Loads and boundary conditions**

The planetary gear carrier input the energy for the wind turbine, through planetary gear, the energy is output by the sun gear. All of the displacement of the sun gear were constrained and its rotation in the xy-plane was set; planet gear was set to the torsion of the z-direction, and the outer surface of the planetary gears is applied in the z-direction displacement constraints; outside surface of the ring gear was fixed; the planet carrier was set the rotation of the xy-direction. In order to avoid the impact generated when the planet carrier started suddenly, the initial rotational speed of the planet carrier is loaded 19 rad/s. The rotational speed of the planet gear is 19 rpm, the drag torque of the sun gear is \(6.4 \times 10^2\) kNm.

**Contact set**

In the nonlinear dynamic analysis process, the motion process is complex. If the setting type of contact is correct, the penetration phenomenon can be eliminated. In the contact process, the dynamic friction coefficient is 0.02, and the static friction coefficient is 0.028. Here, two type contacts were used, which contain automatic surface-to-surface contact and automatic nodes-to-surface contact, while distinguishing between the contact surface and the target surface can be eliminated.

**Results and Discussion**

**Contact stress**

The overall impact and contact stress distribution of the planet gear system with the planet carrier driven can be seen from Figs 4 and 5.

In Figs 6 and 7, the variation of contact stress in the planetary helical gear meshing process can be seen, for the gap between the gears, the planet gear cannot
be meshed and separated simultaneously. When the tooth is just into meshing region, the tooth surface contact stress is small. With planetary gear continues to rotate, the front teeth leave the meshing area gradually, the load which is bear by the meshing gear will increase, the tooth surface contact stress is also increase, the role of lag and stress impact which the teeth flank clearance will be seen in Figs 6 and 7. Since the middle tooth carrying the load, the stress will reach the maximum. When it mesh with a post tooth, the load will distributed in the region which is both the intermediate tooth and post tooth, the contact stress values will declined. The meshing teeth are about to exit, the contact stress will be smaller. The simulation result is the same with the actual, owing to contact ratio is bigger between the helical gears, when they move smooth the contact stress will change small.

In order to verify the reliability of the results, the calculated results were compared with the curve which is the static stress curve in the engagement process. These were shown in Figs 6 and 7. In these figures, The static is static stress curve in static meshing process. Through comparing, the same basic trend can be found, but the static stress curve does not have a zero-stress part. Gears dynamic meshing stress can reflect the stress distribution in the whole process better, while the transmission errors and meshing impact effects will be taken into account fully.

**Tooth root stress**

During meshing of a gear pair of the driving, driven gear, the maximum Von-Mises stress was recorded, it can be seen from Figs 8 and 9, the driving teeth root stress begins to increase, and this is because the meshing position changes from the tooth root to the top. The bent and stress teeth roots suffered increases gradually. The teeth surface force during meshing got maximum; root stress also reaches the maximum value. When the driven gear begins to mesh, the meshing force is not great, but the meshing point is in the top of the driven tooth, the arm from the tooth root to top of the tooth is long, therefore the bent the tooth root suffered is higher. Due to meshing force increasing, the stress begins to rise. The role the tooth side gap plays can be seen throughout the all meshing process. Currently, the material has reached a certain level, which can lead the contact strength of the gear in general meet requirement. But it is still necessary to study the teeth root bending stress, by comparing variation between the dynamic stress and static stress. The curves keep the same change trend.

**Meshing stiffness**

The composite stiffness of the teeth meshing means to participate in the combined effect of the meshing of each pair of tooth in the meshing zone, which is relate with elastic deformation of the teeth. The comprehensive elastic deformation of the tooth, the contact ratio of the gear. The elastic deformation of the teeth is the elastic deformation of the tooth meshing of the tooth surfaces under load. The comprehensive elastic deformation of the tooth is integrated to the elastic deformation of the gear teeth meshing process. The gear contact ratio is generally not an integer multiple, there will be more than one tooth in meshing at the same time. Provided the meshing teeth pair is \( n \), the deformation of the drive and mainly gear is respectively \( \delta_{pi}(i=1,2,\ldots,n) \) and \( \delta_{gi}(i=1,2,\ldots,n) \), the contact force of each tooth is \( F_i(i=1,2,\ldots,n) \), the tooth composite mesh stiffness as follow:

\[
K = \frac{\sum_{i=1}^{n} F_i}{\sum_{i=1}^{n} \delta_{pi} + \delta_{gi}} \tag{6}
\]

Through the 3D finite element contacting analysis, the load distribution between the teeth and the deformation may be got, and the teeth mesh stiffness throughout can also be calculated. Firstly the teeth \( n+1 \) is in meshing, with it exiting slowly, the teeth \( n \) gradually gets into
mesh, when the teeth $n$ is meshing, the teeth $n-1$ will be brought into engagement. Although undulation of the helical gear mesh stiffness is also apparent, there will be no mutation. Helical gear mesh stiffness is relatively steady, while the meshing stiffness of teeth $n-1$ is basically a straight line, which can be seen in Fig. 10, the width of the gear and the contact ratio will influence the mesh stiffness, which is because the width of the gear is big, the contact ratio between ring gear and planetary gear.

Meshing stiffness was got by adding the each engaging teeth varying meshing stiffness in the process, which will compare with theoretical curve. The shapes of them are similar. At the same time, there is something between the meshing stiffness and contact ratio, if the contact ratio is bigger, the meshing stiffness will become smooth, which can be shown in Figs 11-13. Meshing stiffness directly affects the gear system dynamics, it will also affect the strength of the gear, thereby it is important to get accurate calculation.

Contact stress of the planet carrier and planet wheels

The gearbox of the wind generator is used in arms planet carrier, its structural rigidity is good, the bearing is usually installed in the planet gear, now it is simplified in order to facilitate of calculation and observation, owing to the rotation of the carrier, there should be a impact between the carrier and planet gear, so studying this has a certain significance for the stability of planet gear system (Figs 14 and 15). When
the speed of the spindle has a mutation, such shocks will reach a big value in an instant, it will become zero when they rotate smoothly. When rotational of the carrier is 19 rad/s, with initial rotation speed 19 rad/s, the impact action will not exist. But the the rotational speed of the planet carrier is 40 rad/s, the initial speed is 0 rad/s, when it is 0.14E-4 s, the impact stress turn to 500.704 MPa.

Conclusions

By the calculation of the LS-DYNA finite element software can be more intuitive, the dynamic functioning of the planetary helical gear and the stress wave propagation process can be observed during the whole meshing, the impact is caused by the factors such as the elastic deformation of the gear, the engaging force is the fluctuation, the mesh stiffness can be obtained by the contact ratio, the width of the gear and other factors. When torque is a constant, the amount of change of impact force of the planet carrier and the rotational speed is a linear relationship, when the blower starts and the rotational speed changes, the impact action of the planetary carrier should be taken into consideration.

Acknowledgements

This paper is supported by the University Scientific Research Projects Fund of Xinjiang Autonomous Region (XJEDU2010I15) and the National Natural Science Foundation of China (51165043).

References