INTRODUCTION

The wind energy, as one of the most important sustainable energy sources for the future world, has some challenges, such as uneven distribution and low energy density of the wind. The wind energy is suitable to develop and use in the nearby areas. China possesses rich low-wind speed resources close to high-energy consumption areas. Therefore, the development of the wind turbine gearbox for capturing the low-wind speed resources is the research focus of the wind energy.

Many researchers focused on the design and the transmission characteristics of the conventional wind turbine gearbox for the double-fed type wind turbine. The conventional wind turbine gearbox of the double-fed type wind turbine has different transmission structures, such as two planetary gear transmission stages plus one parallel gear transmission stage, one planetary gear transmission stage plus two parallel gear transmission stages and flexible-pins type transmission structure. Some wind turbine gearboxes have only two planetary gear transmission stages. Generally, the flexible-pins type transmission structure provides the best load-sharing performance. In order to investigate the transmission characteristics of the conventional wind turbine gearbox, many factors, such as uncertainties, the flexibility of the internal ring gear, and assembly errors of the pin shafts are reviewed in the literature. Some researchers...
investigated the effect of variable input loads on the wind turbine gearbox. A huge wave was existed in the dynamic load-sharing coefficient curve because of the gravity of the parts. The parameter sensitivity of the planetary gear was also investigated. For investigating the nonlinear dynamic and the exact dynamic response, the flexible multibody modeling and the backlash were considered in the dynamic modeling of the wind turbine gearbox. Meanwhile, for capturing the wind energy of the low-wind speed areas, numerous papers investigated the structure of the blades and many researchers adopted the traditional P-v curve to control the operation of the wind turbine. Larwood et al designed a swept wind turbine blade, and developed a dynamic analysis tool for the blade. Pourrajabian et al developed an aerodynamic design and optimization of the blades for the low-wind speed wind turbine. Asl et al designed a feedback control for the wind turbine gearbox by establishing a classic dynamic model for the wind turbine. Song et al adapted the adaptive algorithms to design a control strategy for a variable-speed wind turbine. Wang et al proposed a linear feedback control strategy by building a two-degree freedom dynamic model of the wind turbine. Fan et al presented an optimized wind turbine power-wind speed curve for capturing more wind energy in the low-wind speed areas.

The structure design and the transmission characteristic of the traditional wind turbine gearbox are investigated. For capturing the wind energy of the low-wind speed areas, the design of the blades and the control of the wind turbine are discussed extensively, based on the traditional P-v curve. Few studies are found to discuss the structure design and the transmission characteristic of the improved wind turbine gearbox, based on the optimized P-v curve. In the present study, the optimized P-v curve is adopted for capturing the wind energy of low-wind speed areas and it is investigated, compared with the conventional P-v curve. An improved transmission structure of the wind turbine gearbox is presented for the low-wind speed areas, based on the optimized P-v curve of the variable-speed double-fed wind turbine.

Transmission characteristics of the improved transmission system are analyzed.

2 | THE OPERATION PRINCIPLE OF THE TRANSMISSION STRUCTURE FOR THE IMPROVED WIND TURBINE GEARBOX

In order to maximize the efficiency and ensure the safe operating of the wind turbine, it is operated according to the wind turbine power-wind speed curve (P-v curve) also called the operation mode of the wind turbine. Figure 1 shows the power-wind speed curves of conventional and optimized wind turbines.

Figure 2 illustrates the improved transmission structure for the wind turbine, based on the optimized P-v curve. Figure 2 shows that the first, second, and third transmission stages of the conventional wind turbine gearbox are the low-speed, medium-speed, and high-speed planetary transmission stages. Moreover, the fourth transmission stage is the torque-implement parallel transmission stage. The transmission system of the improved wind turbine
gearbox consists of two parts, the speed-implement, and the torque-implement transmission systems. The speed-implement transmission system is formed by two planetary transmission stages and a parallel transmission stage, while the torque-implement transmission system is constituted by two parallel stages, the high-speed stage and the torque-implement stage. The pinion gear of the torque-implement parallel stage is connected to the motor by an electromagnetic clutch. It should be indicated that the wheel gear of the torque-implement stage is identical with the wheel gear of the high-speed stage.

If the wind speed is equal or greater than the original cut-in wind speed $v_{in0}$, the wind turbine operates in the region 3 and the clutch is disconnected. On the other hand, if the wind speed is less than the original cut-in wind speed, the wind turbine operates in the region 2 and the clutch is connected.

### 3 | Dynamics Model and Parameters of the Transmission System

The wind turbine in this research is a variable-speed fixed-pitch wind turbine. The pitch angle is zero. The optimized tip-speed ratio $\lambda_{\text{opt}}$ is 6.3. The blade radius is 63 m. The original cut-in wind speed $v_{in0}$ is 5 m/s. The new cut-in wind speed $v_{in1}$ is 2.86 m/s. Table 1 shows specific parameters of the transmission system for the wind turbine gearbox.

| Stage            | Item                        | Ring gear | Sun gear | Planet gear |
|------------------|-----------------------------|-----------|----------|-------------|
| Low speed        | Number of teeth             | 91        | 29       | 31          |
|                  | Module (mm)                 | 23        | 23       | 23          |
|                  | Helix angle (°)             | 5         | 5        | 5           |
|                  | Pressure angle (°)          | 25        | 25       | 25          |
| Medium speed     | Number of teeth             | 109       | 23       | 43          |
|                  | Module (mm)                 | 16        | 16       | 16          |
|                  | Helix angle (°)             | 10        | 10       | 10          |
|                  | Pressure angle (°)          | 20        | 20       | 20          |
| High speed       | Number of teeth             | 121       | 27       |              |
|                  | Module (mm)                 | 12        | 12       |              |
|                  | Pressure angle (°)          | 20        | 20       |              |
|                  | Helix angle (°)             | 9         | 9        |              |
| Torque- implement| Number of teeth             | 121       | 19       |              |
|                  | Module (mm)                 | 12        | 12       |              |
|                  | Pressure angle (°)          | 20        | 20       |              |
|                  | Helix angle (°)             | 9         | 9        |              |

**TABLE 1** Specific parameters

![Dynamic model of the speed-implement transmission system](image-url)
3.1 Dynamic model of the speed-implement transmission system

Moreover, Figure 3 illustrates the dynamic model of the speed-implement transmission system for the wind turbine gearbox. The dynamic model of the planetary and parallel stage is shown in the left picture and the right picture, respectively. Zhai et al. derived and analyzed dynamic equations of the speed-implement transmission system. Dynamic equations of the wheel and pinion gears of the speed-implement transmission system are expressed in Equations (1) and (2), respectively.

\[
\begin{align*}
 m_w \ddot{x}_w + K_w x_w + K_{wpp} \cos(\varphi_{pp} - \frac{\pi}{2} - \alpha_{wt}) \cos \beta_w \delta_{wpp} + K_{wppT} \cos(\varphi_{ppT} - \alpha_{wt}) \cos \beta_w \delta_{wppT} &= 0 \\
m_w \ddot{y}_w + K_w y_w + K_{wpp} \sin(\varphi_{pp} - \frac{\pi}{2} - \alpha_{wt}) \cos \beta_w \delta_{wpp} + K_{wppT} \cos(\varphi_{ppT} - \alpha_{wt}) \cos \beta_w \delta_{wppT} &= 0 \\
m_w \ddot{z}_w + K_{wz} - K_{wpp} \sin \beta_w \delta_{wpp} - K_{wppT} \cos \beta_w \delta_{wppT} &= 0 \\
I_{ww} \ddot{\vartheta}_w + K_{ww} \vartheta_w &= 0 \\
I_{wp} \ddot{\vartheta}_p + K_{wp} \vartheta_p - K_{wpp} \sin \beta_w \delta_{wpp} + K_{wppT} \sin \beta_w \delta_{wppT} &= 0 \\
I_{wp} \ddot{\vartheta}_p + K_{wp} \vartheta_p - K_{wpp} \cos \beta_w \delta_{wpp} + K_{wppT} \sin \beta_w \delta_{wppT} &= 0,
\end{align*}
\]

And the mesh displacement between the wheel gear and the pinion gear of the speed-implement system is expressed as

\[
\delta_{wpp} = (x_w - x_{pp}) \cos \beta_w \cos(\varphi_{pp} - \frac{\pi}{2} - \alpha_{wt}) + (y_w - y_{pp}) \cos \beta_w \sin(\varphi_{pp} - \frac{\pi}{2} - \alpha_{wt}) + (z_{pp} - z_w) \sin \beta_w - (\theta_w^r \ r_{wj} + \theta_p^r \ r_{ppje}) \sin \beta_w - (\theta_w^r \ r_{w} + \theta_p^r \ r_{pp}) \cos \beta_w - e_{wpp}.
\]

3.2 Dynamic model of the torque-implement stage

Figure 4 shows the dynamic model of the torque-implement stage. It should be indicated that the wheel gear of the torque-implement stage is identical with the wheel gear of the high-speed stage. The corner mark of the pinion gear for the torque-implement stage is \(ppT\), reference 1 listed the meaning of other corner marks of the parallel stage. Damping terms are neglected. The dynamic equation of the torque-implement stage is shown in the Equation (3).

\[
\begin{align*}
m_{ppT} \ddot{x}_{ppT} + K_{ppT} x_{ppT} - K_{wppT} \cos(\varphi_{ppT} - \alpha_{wt}) \cos \beta_w \delta_{wppT} &= 0 \\
m_{ppT} \ddot{y}_{ppT} + K_{ppT} y_{ppT} - K_{wppT} \cos(\varphi_{ppT} - \alpha_{wt}) \cos \beta_w \delta_{wppT} &= 0 \\
m_{ppT} \ddot{z}_{ppT} + K_{ppT} z_{ppT} + K_{wppT} \cos \beta_w \delta_{wppT} &= 0 \\
I_{ppT} \ddot{\vartheta}_{ppT} + K_{ppT} \vartheta_{ppT} &= 0 \\
I_{ppT} \ddot{\vartheta}_{ppT} + K_{ppT} \vartheta_{ppT} - K_{wppT} \sin \beta_w r_{ppT} \delta_{wppT} &= 0 \\
I_{ppT} \ddot{\vartheta}_{ppT} + K_{ppT} \vartheta_{ppT} - K_{wppT} \sin \beta_w r_{ppT} \delta_{wppT} &= T_{ppT}.
\end{align*}
\]

3.3 Dynamic model of the overall transmission system

The dynamic equation of the transmission system for the wind turbine gearbox is shown as

\[
M \ddot{X} + C_X \dot{X} + K_X X = F.
\]

The format of the damping matrix and the stiffness matrix was shown in Figure 5. \(C, R, P, S\) represent the carrier, the ring gear, the planet gear, and the sun gear of the planetary transmission stage for the wind turbine gearbox, respectively. \(CP, RP, SP\) represent the coupling terms. \(W, PP, ppT\) represent the wheel gear and the pinion gear of the high-speed stage and the pinion gear of the torque-implement stage, respectively. \(WPP\) and \(WppT\) are the coupling terms.
4 | TRANSMISSION CHARACTERISTIC FOR THE TRANSMISSION SYSTEM

Figure 6 shows the power coefficient of the wind turbine and describes the capacity of the wind turbine to obtain the wind energy. The coefficient is a function of the blade pitch angle (β) and the tip-speed ratio (λ). From the viewpoint of the Betz limit, the maximum of wind turbine power coefficient can be achieved in the region b and the region 3.

The generator power is expressed as

\[ P_b(v_{in0}) = \frac{1}{2} \rho \pi R^2 v_{in0}^3 C_p(\lambda_{in0}, \beta). \] (5)

The wind power at the new cut-in wind speed is expressed as

\[ P_b(v_{in1}) = \frac{1}{2} \rho \pi R^2 v_{in1}^3 C_p(\lambda_{in1}, \beta). \] (6)

The max motor power is expressed as

\[ P_M(\text{max}) = P_b(v_{in0}) - P_b(v_{in1}). \] (7)

The max motor torque is expressed as

\[ T_M(\text{max}) = \frac{P_M(\text{max})}{i_{1st} i_{2nd} i_{4th} \omega_{in0}}. \] (8)

The lowest blade speed and the tip-speed ratio are expressed as

\[ \omega_{in0} = \frac{\lambda_{opt}}{R} v_{in0}, \quad \lambda_{in0} = \frac{R \omega_{in0}}{v_{in0}}, \quad \lambda_{in1} = \frac{R \omega_{in0}}{v_{in1}}. \] (9)

4.1 | Speed ratio characteristic of the transmission system

The transmission ratios of the low-speed stage, medium-speed stage, and high-speed stage are expressed as

\[
\begin{align*}
i_{1st} &= \frac{Z_R}{Z_S} + 1, \\
i_{2nd} &= \frac{Z_{Med}}{Z_S} + 1, \\
i_{3rd} &= \frac{Z_W}{Z_{pp}}.
\end{align*}
\] (10)

The transmission ratio of the torque-implement stage is expressed as

\[ i_{4th} = \frac{Z_W}{Z_{ppt}}. \] (11)
The transmission ratio of the speed-implement system is expressed as

\[ i_{\text{SIS}} = i_{1t}i_{2nt}i_{3mt}. \]  \hspace{1cm} (12)

The transmission ratio of the torque-implement system is expressed as

\[ i_{\text{TIS}} = i_{3mt}i_{4nt}. \]  \hspace{1cm} (13)

\( Z_R \) is the tooth number of the ring gear for the low-wind stage. \( Z_S \) is the tooth number of the sun gear for the low-wind stage. \( Z_{R_{\text{Med}}} \) is the tooth number of the ring gear for the medium-wind stage. \( Z_{S_{\text{Med}}} \) is the tooth number of the sun gear for the medium-wind stage. \( Z_W \) is the tooth number of the wheel gear for the high-speed and the torque-implement stages. \( Z_{pp} \) is the tooth number of the pinion gear for the high-speed stage. \( Z_{ppt} \) is the tooth number of the pinion gear for the torque-implement stage. Based on the Table 1, the transmission ratio of the speed-implement transmission system is 106.427 and the transmission ratio of the torque-implement transmission system is 0.704.

4.2 Dynamic characteristic of the transmission system

Figure 7 illustrates the motor power of the various wind speeds.

100% represent the motor power was 100% \( P_M(\text{max}) \), 50% represent the motor power is 50% \( P_M(\text{max}) \), 0% represent the motor power is 0% \( P_M(\text{max}) \). Moreover, Figure 8A,B show effects of the various motor powers on the peak values of the transverse vibration for the planetary and parallel transmission stages, respectively. Furthermore, Figure 8C,D illustrate vibration tracks of the sun gear for the low-speed and medium-speed planetary stages at 0% \( P_M(\text{max}) \), respectively.

It is observed that for a steady state system, the peak value of the transverse vibration for the planetary and parallel stages is the biggest value of the gear vibration at the \( x \) and \( y \) directions. The sun gear is chosen for the planetary stages. For the track diagram, the horizontal ordinate presents the vibration of the sun gear along the \( x \) direction in the time domain, while the vertical ordinate presents the vibration of the sun gear along the \( y \) direction in the time domain.

Maximal peak values of the transverse vibration for the sun gears of the planetary transmission stages are appeared at the 0% \( P_M(\text{max}) \). Maximal peak values of the transverse vibration for sun gears of the low-speed planetary stage and the medium-speed planetary stage are 0.31 and 0.34 \( \mu \text{m} \), respectively. When the motor power increases and the wind power decreases, the acting torque on the planetary stages decreases and the peak value of the transverse vibration for the sun gears of the planetary stages decreases. It is observed that the maximal peak value of the transverse vibration for the parallel transmission stages appeared at the 100% \( P_M(\text{max}) \). Maximal peak values of the transverse vibration for the pinion gear of the high-speed planetary stage, the wheel gear and the pinion gear of the torque-implement stage are 0.064, 0.105, and 0.107 \( \mu \text{m} \), respectively. When the motor power increases, the acting torque on the torque implement stage increases and the peak value of the transverse vibration for gears of the torque implement stage increases. The peak value of the transverse vibration for the pinion gear of the high-speed stage increases because of the vibration of the torque implement stage. It should be indicated that the scope of the vibration track for the sun gear of the low-speed planetary stage is the same with that for the sun gear of the medium-speed planetary stage. Track diagrams of sun gears for two planetary stages are smooth and regular and the transmission system is steady.

For verifying the accuracy and the validity of the numerical results, the results from Masta software are added. Meanwhile, load-sharing coefficients, natural frequency, and the dynamic mesh forces are calculated and compared with the results from the Masta software. Figure 9 shows the effect of the various motor powers on load-sharing coefficients of the low-speed and medium-speed planetary stages for the wind turbine gearbox.

The equation of the load-sharing coefficient \( K_f \) is expressed as

\[ K_f = \max\left\{ \frac{\max(F_{\text{rp}})}{\text{mean}(F_{\text{rp}})}, \frac{\max(F_{\text{sp}})}{\text{mean}(F_{\text{sp}})} \right\}. \]  \hspace{1cm} (14)

Figure 9 indicates that the minimal load-sharing coefficient for the speed-implement planetary stage appears at the 0% \( P_M(\text{max}) \). Moreover, the minimal load-sharing coefficients of the low-speed and medium-speed planetary stages are 1.0158 and 1.003, respectively. It is observed that the load-sharing coefficients of the speed-implement planetary stage are heightened, as the motor power increases. When the
FIGURE 8 The effect of the various motor power on the peak values of the transverse vibration and the vibration tracks for the planetary transmission stages. (a) Transverse vibration of the planetary stages; (b) Transverse vibration of the parallel stages; (c) Track of the low-speed planetary stage; (d) Track of the medium-speed planetary stage

FIGURE 9 The effect of the various motor power on the load-sharing coefficients of the low-speed planetary stage and the medium-speed planetary stage. (a) Low-speed planetary stage; (b) Medium-speed planetary stage
motor power exceeds 80% $P_{M(max)}$, slopes of the load-sharing coefficient curves are aggrandized. The calculated load-sharing coefficients from the Masta software are similar with the ones from the numerical method.

The natural frequency of the transmission system for the wind turbine gearbox is shown in the Table 2.

Five natural frequency orders are between 0 and 100 Hz. Seven orders are between 100 and 200 Hz. Seven orders are between 200 and 300 Hz. Five orders are between 300 and 400 Hz. Four orders are between 400 and 500 Hz. The Campbell chart of the transmission system for the wind turbine gearbox is shown in the Figure 10.

The left picture is the Campbell chart between 0 and 250 Hz and the right picture is the Campbell chart between 250 and 500 Hz. The blue dotted line is the mesh frequency of the low-speed planetary stage (1st). The black dotted line is the mesh frequency of the medium-speed planetary stage (2nd). The red dotted line is the mesh frequency of the high-speed parallel stage (3rd) and the torque-implement parallel stage (4th).

Moreover, when the motor operates at the 100% $P_{M(max)}$, the Campbell chart of the transmission system for the wind turbine gearbox is shown in the Figure 11. The largest numbers of the cross points between the natural frequency lines and the mesh frequency line of the high-speed parallel stage (3rd) and the torque-implement parallel stage (4th) are existed. Figure 11 shows the effects of the high-speed mesh frequency and the various motor powers on meshing forces for the low- and medium-speed planetary stages.

When the motor operates at the 0% $P_{M(max)}$, meshing forces of the low-speed planetary stage vary between 229.15 and 229.63 kN, while meshing forces of the medium-speed planetary stage vary between 70.96 and 72.60 kN. On the other hand, when the motor operates at the 50% $P_{M(max)}$, meshing forces of the low-speed planetary stage are between 122.50 and 122.59 kN, while meshing forces of the medium-speed planetary stage vary between 38.11 and 38.35 kN. Moreover, when the motor operates at the 100% $P_{M(max)}$,
Meshing forces of the low-speed planetary stage are between 15.14 and 17.58 kN, while meshing forces of the medium-speed planetary stage are between 2.61 and 7.51 kN. When the wind turbine operates at the region 2, the mesh frequency at the high-speed stage is 228.65 Hz. Meshing forces of the low- and medium-speed planetary stages vanish as the motor power increase. Mesh forces from the Masta software have excellent agreement with the numerical results. Figure 12 shows effects of the high-speed mesh frequency and various motor powers on meshing forces for the high-speed and torque-implement stages.

When the motor operates at 0% $P_M(max)$, meshing forces of the high-speed parallel stage are between 53.29 and 54.77 kN. On the other hand, when the motor operates at 50% $P_M(max)$, meshing forces of the high-speed parallel stage vary from 53.56 to 54.93 kN, while meshing forces of the torque-implement parallel stage vary from 25.12 to 25.23 kN. Moreover, when the motor operates at 100% $P_M(max)$, meshing forces of the high-speed parallel stage are between 53.44 and 55.22 kN, while meshing forces of the torque-implement parallel stage are between 49.58 and 51.88 kN. It is found that meshing forces of the high-speed and torque-implement parallel stages increase as the motor power increases.

5 | CONCLUSION

An improved transmission structure of the wind turbine gearbox for low-wind speed areas is presented in this study. The effect of the various motor powers on transmission characteristics of the transmission system for the wind turbine gearbox is investigated. The numerical results are compared with the ones from a professional software. The transverse vibration for sun gears of the planetary stages gradually declines as the motor power increases. It is concluded that the transverse vibration for the pinion gear of the high-speed stage and gears of the torque-implement stage are enlarged with the enhancement of the motor power. Moreover, it is found that as the motor power increases, load-sharing coefficients of the speed-implement planetary stage are heightened. Meshing forces of the low- and medium-speed planetary stages decrease as the motor power increases. Meshing forces of the high-speed and torque-implement parallel stages increase with the augmentation of the motor power.

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NOMENCLATURE

- M mass matrix
- C damping matrix
- K stiffness matrix
- X displacement matrix
| Symbol | Description                      |
|--------|----------------------------------|
| \(\alpha\) | pressure angle                  |
| \(\beta_{w,pp, ppT}\) | helical angle                  |
| \(e\) | static transmission error         |
| \(\delta\) | mesh displacement                |
| \(m\) | mass                             |
| \(I\) | moment of inertia                |
| \(c\) | damping                          |
| \(K\) | stiffness                        |
| \(x\) | vibration displacement at \(X\) direction |
| \(y\) | vibration displacement at \(Y\) direction |
| \(z\) | vibration displacement at \(Z\) direction |
| \(\theta\) | torsional displacement           |
| \(r\) | radius                           |
| \(\varphi\) | position angle                   |
| \(\lambda\) | tip-speed ratio                  |
| \(\beta\) | pitch angle                      |
| \(P\) | power                            |
| \(\rho\) | air density                      |
| \(R\) | blade radius                     |
| \(v\) | wind speed                       |
| \(C_p\) | wind turbine power coefficient   |
| \(T\) | torque                           |
| \(\omega\) | rotation                         |
| \(i\) | transmission ratio               |
| \(Z\) | tooth number                     |

**SUBSCRIPTS**

- \(w\): wheel gear
- \(pp\): pinion gear of speed-implement transmission system
- \(ppT\): pinion gear of torque-implement transmission system
- \(R\): ring gear
- \(S\): sun gear
- \(wt\): transverse pressure angle
- \(je\): pitch circle
- \(ji\): base circle
- \(b\): generator power
- \(M\): motor power
- \(w\): wind power
- \(\text{in0}\): original cut-in wind speed
- \(\text{in1}\): wind speed in the region 2
- \(1\text{st}\): low-speed stage
- \(2\text{nd}\): medium-speed stage
- \(3\text{rd}\): high-speed stage
- \(4\text{th}\): torque-implement stage
- \(\text{SIS}\): speed-implement system
- \(\text{TIS}\): torque-implement system
- \(\theta_x\): around \(X\) axis
- \(\theta_y\): around \(Y\) axis
- \(\theta_z\): around \(Z\) axis

**SUPERSCRIPTS**

- \(x\): around \(X\) axis
- \(y\): around \(Y\) axis
- \(z\): around \(Z\) axis

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