Optimization design of wind turbine drive train based on Matlab genetic algorithm toolbox

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Abstract. In order to ensure the high efficiency of the whole flexible drive train of the front-end speed adjusting wind turbine, the working principle of the main part of the drive train is analyzed. As critical parameters, rotating speed ratios of three planetary gear trains are selected as the research subject. The mathematical model of the torque converter speed ratio is established based on these three critical variable quantity, and the effect of key parameters on the efficiency of hydraulic mechanical transmission is analyzed. Based on the torque balance and the energy balance, refer to hydraulic mechanical transmission characteristics, the transmission efficiency expression of the whole drive train is established. The fitness function and constraint functions are established respectively based on the drive train transmission efficiency and the torque converter rotating speed ratio range. And the optimization calculation is carried out by using MATLAB genetic algorithm toolbox. The optimization method and results provide an optimization program for exact match of wind turbine rotor, gearbox, hydraulic mechanical transmission, hydraulic torque converter and synchronous generator, ensure that the drive train work with a high efficiency, and give a reference for the selection of the torque converter and hydraulic mechanical transmission.

1. Introduction
Intermittent and random of the wind resource will bring about a series of problems to the utilization of wind [1]. In order to enhance the electricity generation stability of the wind turbine, the hydraulic mechanical speed control device had introduced into the wind turbine drive train [2, 3].

In this paper, in order to get the key parameters which can ensure the torque converter and the drive train working with a high efficiency, the working principle of the main part of the drive train is analyzed, the mathematical model is established, and then the optimization calculation about the efficiency of the drive train is carried out based on the MATLAB genetic algorithm toolbox. The model based on those key parameters provides a basis for the prototype test in the next step.

2. The structure of the drive train of the front-end speed adjusting wind turbine
The whole drive train of the front-end speed adjusting wind turbine is composed of wind turbine rotor, gearbox, hydraulic mechanical transmission, hydraulic torque converter and synchronous generator [4].

As shown in Figure 1, after the gear box, a planetary frame is connected at the planet wheel of the hydraulic mechanical transmission. This planet wheel drives the sun wheel. And the sun wheel drives
the spindle. In order to adapt to the wind, a part of energy of the spindle is diverted to the sun gear by the hydraulic transmission. By this way, this drive train can enhance the stability of the spindle.

3. Mathematical Model and Key Parameters

As shown in Figure 1, from left to right, there are three planetary gears in the drive train. And the mechanical transmission satisfies the following relationship:

\[
a = -\frac{n_{j2}}{n_q}
\]  
\[
i_{\text{m2}}^{j2} = \frac{n_t - n_{j2}}{n_q - n_{j2}}
\]
\[
i_{\text{tq}}^{j3} = \frac{n_t - n_{j3}}{n_q - n_{j3}}
\]

Here \(n_t\) is the rotating speed of the wind turbine rotor, \(n_{j1}, n_{j2}\) and \(n_{j3}\) are respectively corresponding to the rotating speed of the first, the second and the third planet carrier, \(a\) is the speed increase ratio of the gearbox, \(n_t\) is the rotating speed of the sun gear, \(n_q\) is the rotating speed of the outer ring gear of the hydraulic transmission, \(n_t\) is the rotating speed of the turbo of the torque converter, \(i_{\text{m2}}^{j2}\) is the ratio of the sun gear and the fixed outer ring gear of the second planet carrier, and \(i_{\text{tq}}^{j3}\) is the ratio of the turbo of the torque converter and the fixed outer ring gear of the third planet carrier. The key parameters are defined as follow: \(b = -i_{\text{m2}}^{j2}\), \(c = -i_{\text{tq}}^{j3}\). Then, the Eq. (2) and (3) can be expressed as shown below:

\[
n_t = (1 + b)n_{j2} - bn_q
\]  
\[
n_t = (1 + c)n_{j3} - cn_q
\]

The rotating speed of third planet carrier is zero. Based on Eq. (5), the following equation is obtained:

\[
c = -\frac{n_t}{n_q}
\]

Note that the sun gear, the pump wheel and the generator are connected at the spindle, then:

\[
n_t = n_{\text{w}} = n_{\text{i3}}
\]
Here \( n_B \) is the rotating speed of the pump wheel of the torque converter, and \( n_G \) is the rotating speed of the generator. Based on Eq.(1), (4), (6) and (7), the following equation can be derived:

\[
n_i = \frac{c n_B + a(1 + b) c n_G}{b}
\]

(8)

Then the rotating speed ratio of the torque converter is obtained as follow:

\[
i_{TB} = \frac{n_i}{n_B} = \frac{c n_B + a(1 + b) c n_G}{b n_B}
\]

(9)

When \( n_f \), \( n_B \) and \( n_G \) are given, based on Eq. (9), \( i_{TB} \) is only relate to the parameters of \( a \), \( b \) and \( c \). With the energy diversion effect on the spindle, the hydraulic torque converter must work in high state to ensure the efficient operation of the entire system. To ensure the efficient, the torque converter must work within a certain range [5].

4. Objective function

Based on the drive train system shown in Figure 1, the torque balance equation and the energy balance equation can be obtained as follows:

\[
M_i = M_q + M_{j2}
\]

(10)

\[
M_i n_i = M_q n_q + M_{j2} n_{j2}
\]

(11)

The proportional relationship between the torques can be obtained based on Eq. (4), (10) and (11):

\[
M_i : M_q : M_{j2} = -1 : b : -(1 + b)
\]

(12)

Combining with Eq. (1) and (6), the following equation can be derived:

\[
M_i : M_f : M_{j} = 1 : \frac{b}{c} : -a(1 + b)
\]

(13)

Hydraulic torque converter has the following basic relationships:

\[
K = \frac{M_f}{M_B}
\]

(14)

\[
\eta_{TB} = K i_{TB}
\]

(15)

Here \( K \) is torque ratio, and \( \eta_{TB} \) is the efficiency of hydraulic torque converter. Then combining with Eq.(13), (14) and (15), the following equation can be derived:

\[
\frac{M_B}{M_f} = -\frac{b i_{TB}}{a(1 + b) c \eta_{TB}}
\]

(16)

By analysing key parts of the drive train, power equations can be obtained as follows.

\[
M_q \omega_q = M_i \omega_i \eta_{Tq} = M_B \omega_B \eta_{TB} \eta_{Tq}
\]

(17)

\[
M_f \omega_f = M_i \omega_i \eta_{Tj}
\]

(18)

\[
M_i \omega_i = (M_B + M_G) \omega_B
\]

(19)

Combining with Eq. (11), (17), (18) and (19), the following equation can be derived:

\[
M_i \omega_i = M_B \omega_B \eta_{TB} \eta_{Tq} + M_f \omega_f \eta_{Tj} - M_B \omega_B
\]

(20)
Now, based on Eq.(20), the efficiency of the whole drive train can be derived as shown below:

$$\eta = \frac{M_a \omega_b}{M_f \omega_f} = \frac{M_a \omega_b \eta_{TB} \eta_{Tq}}{M_f \omega_f} + \frac{M_e \omega_e \eta_f}{M_f \omega_f} - \frac{M_a \omega_b}{M_f \omega_f}$$

Furthermore, combining with Eq. (16), the efficiency of the whole drive train can be expressed as shown below:

$$\eta = \eta_f + (1 - \eta_{TB} \eta_{Tq}) \frac{n_b}{n_e} \frac{b_i_{TB}}{a(1+b)c \eta_{TB}}$$

(21)

The fitness function can be obtained based on Eq.(21):

$$F = 1/[\eta_f + (1 - \eta_{TB} \eta_{Tq}) \frac{n_b}{n_e} \frac{b_i_{TB}}{a(1+b)c \eta_{TB}}]$$

(22)

It is shown in the Eq. (21) that the efficiency of the whole drive train has a close relationship with key parameters of $a$, $b$ and $c$. In order to get the highest efficiency of the system, the choosing of key parameters should make sure that $F$ in the Eq. (22) reach the minimum value.

5. Constraint functions

According to the range $\eta_{f_{\text{max}}}$ to $\eta_{f_{\text{min}}}$, $i_{TB_{\text{min}}}$ and $i_{TB_{\text{max}}}$ is confirmed. And the design rotating speed ratio of the hydraulic torque converter, $i_{TB}$, is corresponding to the common low rotating speed of the wind turbine rotor, $n_e$. Then, the following equations can be obtained:

$$0 \leq i_{TB_{\text{min}}} < i_{TB1} < i_{TB} < i_{TB2} < i_{TB_{\text{max}}} \leq 1$$

(23)

$$i_{TB} = \frac{cn_b + a(1+b)c n_f}{bn_b} = i_{\eta_{\text{max}}}$$

(24)

6. Examples and results analysis

In this paper, technical data of the wind turbine DeWind 8.2 and experimental data of the hydraulic torque converter NY5 are used to carry out the matching calculation. The detail is shown in Table 1, Table 2.

Seen from Table 1, the rotating speed range of the wind turbine rotor is $11.1 \sim 20.7 \text{r/min}$. The common low rotating speed of the wind turbine rotor is defined as $16 \text{r/min}$. The gear ratio is $1:25$, so the parameter, $a = 25$. And the rotating speed of the generator is $-1500\text{r/min}$.

| Parameters                | Value                  |
|---------------------------|------------------------|
| Rated power               | 2000 kW                |
| Rotor diameter            | 80 m                   |
| Swept area                | 5027 m$^2$             |
| Rotor speed range         | 11.1 to 20.7 min$^{-1}$|
| Gear ratio                | 1:25                   |
| Windrive ratio            | 1:3 to 1:5.5           |
| Generator speed           | 1500 min$^{-1}$ at 50Hz; 1800 min$^{-1}$ at 60Hz |
| Output voltage            | 4.16 bis 13.8 kV       |
Seen from Table 2, the maximum efficiency of the torque converter, $\eta_{TB} = 0.766$. And the rotating speed ratio, $i = 0.3$. Then, Transmission efficiency of mechanical parts is defined as $\eta_i = \eta_{TB} = 0.985$. So, based on Eq. (22), (23) and (24), the fitness function and the constraint function can be derived as shown below:

$$F = 1/\left[0.985 - \left(1 - 0.766 \times 0.985\right) \frac{1500}{16} \times \frac{0.3 \times b}{0.766 \times 25 \times (1 + b)c}\right]$$ \hspace{1cm} (25)

$$0 \leq i_{TB_{\text{min}}} = \frac{-1500c + 20.7 \times 25 \times (1 + b)c}{-1500 \times b} < 0.3$$ \hspace{1cm} (26)

$$0.3 < i_{TB_{\text{max}}} = \frac{-1500c + 11.1 \times 25 \times (1 + b)c}{-1500 \times b} \leq 1$$ \hspace{1cm} (27)

$$i_{TB} = \frac{-1500c + 16 \times 25 \times (1 + b)c}{-1500 \times b} = 0.3$$ \hspace{1cm} (28)

Based on Eq. (25) to (28), fitness function file FitFun.m and constraint function file NonCon.m are established in MATLAB. The results of the calculation that aim at maximum efficiency can be obtained as follows:

Based on table 3, the best key parameters can be obtained. $a = 25$, $b = 1.9$, $c = 2.5$. According to the key parameters of this group, this flexible drive train of the front-end speed adjusting wind turbine can get the maximum efficiency of 89% when the rotating speed ratio of the hydraulic torque is 0.3.

| $i$  | $\eta_i \times 10^6$ | $K$   | $\eta_{TB}$ |
|------|----------------------|-------|-------------|
| 0    | 0.48                 | 7.85  | 0           |
| 0.1  | 0.538                | 5.18  | 0.52        |
| 0.2  | 0.546                | 3.47  | 0.696       |
| 0.3  | 0.538                | 2.54  | 0.766       |
| 0.4  | 0.536                | 1.87  | 0.756       |
| 0.5  | 0.534                | 1.6   | 0.652       |

Table 2. Experimental data of NY5 Hydraulic torque converter [7].

| $a$  | $b$ | $c$ | $i_{TB}$ | $\eta$  |
|------|-----|-----|----------|---------|
| 25   | 1.898 | 2.508 | 0.30022  | 0.89085 |
| 25   | 1.899 | 2.509 | 0.29983  | 0.89087 |
| 25   | 1.895 | 2.492 | 0.29983  | 0.8903  |
| 25   | 1.833 | 2.248 | 0.2999   | 0.88123 |
| 25   | 1.868 | 2.384 | 0.30017  | 0.8865  |
| 25   | 1.277 | 0.975 | 0.29991  | 0.77761 |
| 25   | 1.229 | 0.909 | 0.29999  | 0.76631 |
| 25   | 1.602 | 1.571 | 0.30021  | 0.8437  |
| 25   | 1.537 | 1.425 | 0.2999   | 0.83172 |
| 25   | 1.333 | 1.058 | 0.29991  | 0.79029 |
7. Conclusion
In this paper, based on the analysis and mathematical modelling of the drive chain, objective function and constraint function are established. Furthermore, based on the application of the MATLAB genetic algorithm toolbox, optimization analysis about the key parameters is carried out. As the result, reasonable key parameters provide a basis to ensure the efficient operation of the front-end speed wind turbine drive train. The results show that, when the torque converter rotating speed ratio is 0.3 and the rotating speed of the wind turbine rotor is 13r/min, the optimized three key parameters of $a = 25$, $b = 1.9$, $c = 2.5$ will make the transmission efficiency of the drive train reached 89%. The optimization method and results provide an optimization program for exact match of wind turbine rotor, gearbox, hydraulic mechanical transmission, hydraulic torque converter and synchronous generator, ensure that the drive train work with a high efficiency, give a reference for the selection of the torque converter and hydraulic mechanical transmission, and provide a necessary foundation to the optimization match design of the large power wind turbine drive chain.

However, in practice, before the rated wind, the rotor speed of the wind turbine will change with the random wind speed change. Different rotor speed will lead different optimal parameters. This means that the rotating speed ratios of three planetary gear trains are variable. In order to deal with this situation, a new variable speed control mechanism should be introduced. Different from the traditional gear transmission mechanism, this new transmission mechanism should have continuity, stability and accuracy at the same time, just like the CVT (Continuously Variable Transmission) system in the car. Further study should be carried out in the future, to ensure the wind turbine to adapt the variable wind, and work with the highest efficiency.

References
[1] Sun H and Cheng Z M 2012 Science & Technology Information 2012(12) 127
[2] de Vries E 2007 Renewable Energy World 10(2) 55
[3] Thomsen K E, Dahlhaug O G, Niss M O K and Haugset S K 2012 Energy Procedia 24 76 -82.
[4] Shao J H, He Y L, Jin X and Yang X M 2007 Modern Manufacturing Engineering 2007(6) 112-5
[5] Dong Y, Wang H J and Zhou X Q 2008 Journal Of Engineering For Thermal Energy And Power 23(6) 670-5
[6] Detlef K 2009 Sun & Wind Energy 2009(5) 216-8
[7] Northern Jiaotong University 1980 Hydraulic Transmission of Diesel locomotive (Beijing: China Railway Publishing House) pp 78-89