Active Load Reduction Control of Floating Wind Turbine
Based on Double Damper

Zhu-li DENG¹,*, Lei WANG¹ and Li-ping DENG²
¹School of Automation College, Chongqing University, Chongqing, China
²College of Foreign Languages, Chongqing University of Posts and Telecommunications, Chongqing, China
*Corresponding author

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Abstract. Due to the lack of rigid support structure of floating wind turbine (WT) at sea, the external load affected by wind-wave-flow coupling is much larger than that of fixed-pile WT. How to make the floating WT safe, reliable and stable operation has become a research hotspot in the current far-reaching sea breeze energy utilization. In order to reduce the load of offshore floating WT subjected to external environment during operation, this paper studies a double damper load reduction structure with confined mass dampers installed in the nacelle and floating platform. Active control method to reduce the load of floating WT at sea. Through the simulation verification analysis of NREL 5MW wind turbine, it is found that the load impact and output power stability are effectively improved, which provides a useful method for the development of deep sea wind power resources in China.

Introduction

With the development of floating wind power at sea, the requirements for load reduction control are getting higher and higher [1]. At present, in the control of active and passive load reduction structures, some control methods have been successfully applied in recent years [2]. Among them, the vibration energy dissipation method of active damping, semi-active damping and passive damping is used to reduce the system load of floating WT [3]. Although passive control techniques can play a good role in fine-tuning damper parameters in specific environments, they may lose their effectiveness and produce dissonant effects due to the environment or system change, especially in harsh environments in the deep sea [4]. Different from passive control, semi-active control can adjust the characteristics of the system in real time in time domain according to the nature of vibrational forces such as frequency content and amplitude [5], while active control can ensure that it is effective in different working environments [6]. Brodersen et al. [7] proposed an active tuned mass damper(ATMD) to carry out structural vibration control of a wind motor, in which the additional active control is rendered by the motion of the wind motor tower itself and the motion of the damper.

Most floating wind motor active damper load reduction structure control methods only consider the case where the damper is installed in the nacelle. Because the size of the nacelle itself is refrained, the damper can only move in a small range. Therefore, this method has a large limitation on the range of the mass and the damper [8]. In this paper, in the case that it is difficult for a single damper to provide sufficient load reduction control under heavy wind loads, a load reduction control structure with active dampers installed in the nacelle and floating platform is proposed, using FAST-SC [9] wind power. The machine simulation platform was simulated under a variety of different loads. It was found that the installation of active dampers in the nacelle and floating platform simultaneously improved the load reduction control effect, comparing to the installation of a single damper.
Double Damper Load Reduction Structure Modeling

In order to effectively reduce the load of the floating WT, this paper proposes a method of installing the tuned mass damper in the nacelle and the floating platform at the same time. The two dampers jointly reduce the load on the entire floating WT. A typical active damper is shown in Fig. 1.

![Figure 1. Actively tuned mass damper.](image1)

![Figure 2. Schematic Diagram of Damper Installation.](image2)

For the floating WT shown in Fig. 2, the motion constraint equation can be derived from the Euler-Lagrangian equations:

\[
\begin{align*}
-M \omega_{wind} + f_{T} R_{n} \pm k_{\text{lim}} \left( \theta_{p} - \theta_{i} \right) - m_{R} g_{R} \theta_{p} - (k_{\text{lim}} R_{n} + m_{n} g_{n} + k_{\text{lim}} R_{n}) ( x_{n} - R_{n} \theta_{i} ) \\
+ k_{\text{lim}} x_{\text{lim}} R_{n} + d_{p} \left( \dot{\theta}_{p} - \dot{\theta}_{i} \right) + (d_{\text{lim}} R_{n} + d_{p} R_{n}) (R_{n} \dot{\theta}_{i} - \dot{x}_{n}) + I_{i} \ddot{\theta}_{i} = 0 \\
-M \omega_{wave} - f_{T} R_{p} + (k_{p} + m_{p} g_{p}) \theta_{p} - k_{\text{lim}} x_{\text{lim}} R_{p} \left( \theta_{p} - \theta_{i} \right) + (k_{\text{lim}} R_{p} - m_{p} g_{p} + k_{\text{lim}} R_{p}) ( x_{p} + R_{p} \theta_{p} ) \\
- k_{\text{lim}} x_{\text{lim}} R_{p} + d_{p} \dot{\theta}_{p} - d_{i} \left( \dot{\theta}_{i} - \dot{\theta}_{p} \right) + (d_{\text{lim}} R_{p} + d_{p} R_{p}) (R_{p} \dot{\theta}_{p} + \dot{x}_{p}) + I_{p} \ddot{\theta}_{p} = 0 \\
-f_{T} + (k_{n} + k_{\text{lim}}) ( x_{n} - R_{n} \theta_{i} ) - k_{\text{lim}} x_{\text{lim}} R_{n} m_{n} g_{n} \theta_{i} + (d_{\text{lim}} + d_{n}) ( \dot{x}_{n} - R_{n} \dot{\theta}_{i} ) + m_{n} \ddot{x}_{n} = 0 \\
-f_{T} + (k_{p} + k_{\text{lim}}) ( x_{p} + R_{p} \theta_{i} ) - k_{\text{lim}} x_{\text{lim}} R_{p} m_{p} g_{p} \theta_{p} + (d_{\text{lim}} + d_{p}) ( \dot{x}_{p} + R_{p} \dot{\theta}_{p} ) + m_{p} \ddot{x}_{p} = 0.
\end{align*}
\]

(1)

In order to obtain the state space expression of the system, the constraint equation of the system is described as a matrix form:

\[
M \dot{X} + D \dot{X} + KX = F_{T} f_{T} + F_{d} f_{d} + Cst.
\]

(2)

In the state space representation of the system, the angle of the tower and pontoon swing are used, the travel of the two active constrained damper masses and their first derivative \([X \ \dot{X}]\) as the state of the system \(x\), thus the state space expression is acquired, as follow:

\[
\begin{align*}
\dot{x} &= Ax + B_{T} f_{T} + B_{d} f_{d} \\
y &= Cx
\end{align*}
\]

(3)

where the state matrix \(A\), the control input matrix \(B_{T}\) and the wind-wave joint load disturbance input matrix \(B_{d}\) are:

\[
A = \begin{bmatrix}
0 & I \\
-M^{-1}K & -M^{-1}D
\end{bmatrix},
B_{T} = \begin{bmatrix}
0 \\
M^{-1}F_{T}
\end{bmatrix},
B_{d} = \begin{bmatrix}
0 \\
M^{-1}F_{d}
\end{bmatrix}
\]

(4)

Let the output \(y\) of the state space expression be:

\[
y = \begin{bmatrix}
R_{i} \left( \theta_{i} - \theta_{p} \right) & \theta_{p} & x_{n} & R_{n} \theta_{i} & x_{p} & R_{p} \theta_{p}
\end{bmatrix}
\]

(5)

the items in the above formula are indicated in turn: the horizontal movement offset distance of the spire, the pitch angle of the pontoon swing, the horizontal displacement of the restricted damper mass installed in the nacelle, and the displacement of the restricted damper mass installed in the pontoon.
When designing the controller, try to make $y$ close to zero. We can express the output matrix of the system state space expression as:

$$
C = \begin{bmatrix}
-R_t & R_t & 0 & 0 & 0 & 0 & 0 & 0 \\
1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & -R_{tn} & 1 & 0 & 0 & 0 & 0 & 0 \\
-R_{tp} & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\
\end{bmatrix}.
$$

**Active Load Reduction Controller Design**

Linear quadratic regulator (LQR) can obtain the optimal control law of state linear feedback. As mentioned above for the controller-based target, we get the optimal control objective function as follows:

$$
J = \frac{1}{2} \int_0^\infty \left[ q_1 \left( R_t \left( \theta_t - \theta_p \right) \right)^2 + q_2 \theta_p^2 + q_3 \left( x_{tn} - R_{tn} \theta_t \right)^2 + q_4 \left( x_{tp} + R_{tp} \theta_t \right)^2 + r_1 f_{tn}^2 + r_2 f_{tp}^2 \right] dt,
$$

where $q_1$, $q_2$, $q_3$ and $q_4$ are the tower-top fore-aft deflection (TTDspFA), the platform pitch rotational displacement (PtfmPitch), the horizontal movement distance of the damper in the nacelle (HmdXDxn), and the horizontal movement distance of the damper in the floating platform (HmdYDyn). In order to obtain the optimal TMD weight coefficient, this paper adopts a genetic algorithm (GA) based parameter optimization genetic algorithm [10]. $r_1$ and $r_2$ are the weighting factors for the forces acting on the nacelle and pontoon damper, respectively. To facilitate the solution, the above objective function is written in a matrix form as follows:

$$
J = \frac{1}{2} \int_0^\infty \left( x^\top C^\top Q C x + f_{r}^\top R_{r} f_{r} \right) dt,
$$

where $Q$ is the diagonal matrix consisting of $q_1$, $q_2$, $q_3$ and $q_4$, $R$ is the diagonal matrix consisting of $r_1$ and $r_2$, and $C$ is the output matrix. In order to get the optimal control law and minimize the objective function, we need to solve the following Riccati equation:

$$
P A + A^\top P - P B_t R_t^{-1} B_t^\top P + C^\top Q C = 0,
$$

where $P$ is the unknown matrix to be solved, from which the control law can be obtained. If the resulting matrix $P$ is positive definite, the system is asymptotically stable. The final control law is as follows:

$$
f_r(t) = R_t^{-1} B_t^\top P x(t).
$$

**Simulation Analysis**

In this paper, the 5MW barge type floating wind motor designed by NREL is used for simulation. In order to compare the load reduction performance differences between different load reduction structures (with or without damper, single or multiple damper load reduction control structures), it is necessary to select different load reduction control structures under different load conditions. Perform a simulation comparison of active load reduction control. According to the IEC61400-3 offshore WT design standard [12], we choose the Kalman spectrum as the wind spectrum model. For wave loads, the JONSWAP spectrum as the spectral model is used for the simulation. The two typical conditions for simulation are shown in Table 1.
Table 1. Two typical load cases.

<table>
<thead>
<tr>
<th>case</th>
<th>case one</th>
<th>case two</th>
</tr>
</thead>
<tbody>
<tr>
<td>average wind speed (ms(^{-1}))</td>
<td>10</td>
<td>18</td>
</tr>
<tr>
<td>effective wave height (m)</td>
<td>5.6</td>
<td>9.2</td>
</tr>
<tr>
<td>wave load peak period (s)</td>
<td></td>
<td>11.8</td>
</tr>
</tbody>
</table>

The simulation results under different loads by using different load reduction control structures are shown in Fig. 3 and Fig. 4.

During our simulation, it is found that the reduction in pontoon vibration significantly affected other performance metrics. It can be seen from the time domain curve of each performance indicator in the figure that these three active control methods are superior to the baseline undamper control method under different load conditions. The fluctuation amplitude of various performance indicators of the double damper optimal control method is obviously lower than other control structures.

Conclusions

In this paper, a double damper load reduction control structure is proposed and the linear quadratic regulator controller is designed to actively control it to reduce the load of the floating WT. By comparing with the simulation experiment without damper and single damping active control method, the following results can be obtained: a) Double damping optimization control method considering multiple performance indicators as a comprehensive multi-input and multi-output control method, it can effectively reduce the load of WT and improve the stability of output power. At the same time, it can also consider the stroke limit and the control force of the damper. b) The double damping optimal control of the load reduction effect is better than the single damping active control. In addition to greatly reducing the load on the WT, optimal control of multiple dampers stabilizes the output power of the WT. c) When the wind speed exceeds the rated wind speed, the structural stability of the WT can increase the total power generation. This also means that the active control of multiple
dampers not only reduces the load on the WT, stabilizes the power, but also ensures the output power when the rated wind speed is exceeded.

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