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Vibration Analysis of the Low Speed Shaft and Hub of a Wind Turbine Using Sub Structuring Techniques

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Abstract

The wind turbine vibration causes dynamic instability of components of the transmission system desired to generate electricity. The vibration motion of the hub relative to the shaft need to be studied in order to improve durability of components part and ensure that a good percentage of wind energy captured is transmitted to electricity. This paper attempts to do that. A sub-structuring technique was adopted in the analysis of the vibration motion. Firstly, the finite element equation was derived for the hub and shaft structures before coupling. After coupling, the appropriate boundary condition was used to minimize the system complexities. The solution to the eigenvalue problem was then developed using MATLAB. This results obtained show that sub-structuring is a suitable technique for the dynamic analysis of the transmission system of a wind turbine as compared with other approaches.

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Keywords: Vibration, Sub structural Techniques, Low Speed Shaft and Hub, Wind Turbine

1. Introduction

Vibration is a mechanical phenomenon whereby oscillations occur about an equilibrium point. The oscillations may be periodic such as the motion of a pendulum or random such as the movement of a tire on a gravel road. The physical explanation of the phenomena of vibration concerns the interplay between potential energy and kinetic energy. The fundamental Kinematic quantities used to describe the motion of a particle are displacement, velocity and acceleration vectors. The structures designed to support the high speed engines and turbines are subjected to vibration. [1] The vibration causes rapid wear of machine parts such as bearings and gears. Unwanted vibrations may cause loosening of parts from the machine because of improper design or material distribution, the wheels of locomotive can leave the truck due to excessive vibration which results in accident or heavy loss. Sometimes because of heavy vibrations, proper readings of instrument cannot be taken.[2,3] Thus keeping in view all these devastating effects, the study of vibration is essential for a mechanical engineer to minimize the vibrational effects over the me-

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chanical components by designing them suitably. Vibrations can be used for useful purposes such as vibration testing equipment, vibratory conveyors, hoppers, sieves and compactors. Vibration is found very fruitful in mechanical workshops such as in improving the efficiency of machining, casting, forging and welding techniques, musical instruments and earthquakes for geological research. It is useful for propagation of sound. However, undesirable vibrations should be eliminated or reduced up to certain extent by the following methods. (Singh, 1997): Removing external excitations, Using shock absorbers, Dynamic absorbers, and Resting the system on proper vibration isolators. [4,5]

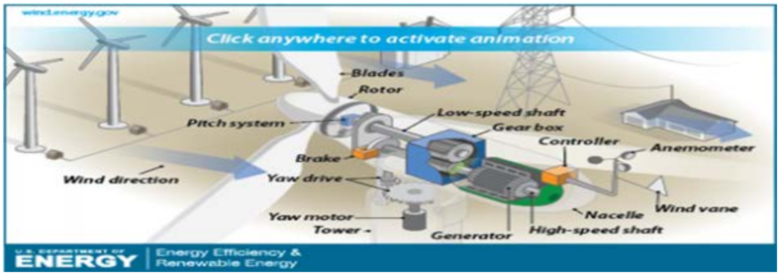

Fig. 1: Wind turbine components

The wind energy industry still encounters untimely component failures, which increases operation and maintenance costs, and in this manner, the cost of energy for wind power. As turbines sizes are increased and introduced offshore, these failures will turn out to be much all the more excessive. To make wind power more focused, there is a requirement for the industry to enhance turbine reliability and accessibility.[6,7,8]

Wind turbine vibrations decrease the life-cycle of wind turbines and in this manner it merits further studies. Reducing the vibrations of a wind turbine can possibly decrease number of part failures and improve the usability of the components. This results in greater turbine accessibility and diminished maintenance costs. [9,10,11]

This paper majors on the vibration analysis on the low speed shaft linking the hub to the main transmission. The hub is a rigid centre piece with hollow portions for proper assembly of the rotor blades. The rotating blades which captures the wind energy transmits motion from the centre hub coupled to a low speed rotating shaft. The speed is then magnified through transmission gears into the high speed shaft that is attached to a generator which then generates electricity.[12] This analysis considers the free vibration only without considering the time dependent wind loading. The solution of the eigenvalue problem was determined only without considering modal testing. The shaft being considered is circular with uniform cross section and there would be no experiment carried out. [13,14,15,16]

2. Problem Formulation

2.1. Assumptions

The governing equation is based on the following assumptions.

- The material of the shaft is uniform throughout.
- The shaft circular in sections remains circular after loading.
- A plane section of shaft normal to its axis before loading remains plane after the torques have been applied
- The twist along the length of the shaft is uniform throughout.
- The shaft is a hollow type of thick walled tube

- The distance between any two normal cross-sections remains the same after the application of torque.
- The effect of spherical roller bearings on the shaft is neglected
- The shape of the hub is perfectly circular although not practically so

2.2 Governing equations

According to the Euler-Bernoulli theory, the forced lateral vibration w(x,t) of a uniform beam is governed by this equation (Rao, 2007)

$$\mathrm{EI}\frac{\partial^4 \mathrm{w}}{\partial \mathrm{x}^4}(\mathrm{x},\mathrm{t}) + \rho\mathrm{A}\frac{\partial^2 \mathrm{w}}{\partial \mathrm{t}^2}(\mathrm{x},\mathrm{t}) = \mathrm{f}(\mathrm{x},\mathrm{t})(3.2) \qquad (1)$$

Where f (x, t) denotes the time-varying distributed force.

I — axial moment of inertia of the cross section of the beam;

A — cross-sectional area of the beam;

ρ — mass density of the beam.

3. Problem Solution

3.1 Finite element representation.

Using Garlekin's approach to minimize Eq. (3.2). This is done by the integral merging of the shape function equivalent to the weight function of each element

The stiffness matrix for each finite element can be expressed symmetrically as follows:

$$K = \frac{EI}{l^3}\begin{bmatrix} 12 & 6l & -12 & 6l \\ & & 4\boldsymbol{l}^2 - 6l & 2\boldsymbol{l}^2 \\ & & 12 & -6l \\ sym & & & 4\boldsymbol{l}^2 \end{bmatrix} \qquad (2)$$

The mass matrix for each finite element can also be expressed symmetrically as

$$M = \frac{\rho A l}{420}\begin{bmatrix} 156 & 22l & 54 & -13l \\ & & 4\boldsymbol{l}^2 - 13l & -3\boldsymbol{l}^2 \\ & & 156 & -22l \\ sym & & & 4\boldsymbol{l}^2 \end{bmatrix} \qquad (3)$$

The finite element equation for element 1 i.e the connecting center hub can be written as

$$363.53\begin{bmatrix} 156 & 66 & 54 & -39 \\ & 36 & -39 & -27 \\ & & 156 & -66 \\ sym & & & 36 \end{bmatrix}\begin{Bmatrix} \ddot{W}_1 \\ \ddot{W'}_1 \\ \ddot{W}_2 \\ \ddot{W'}_2 \end{Bmatrix} + 5.3x10^{10}\begin{bmatrix} 12 & 18 & -12 & 18 \\ & 36 & -18 & 18 \\ & & 12 & -18 \\ sym & & & 36 \end{bmatrix}\begin{Bmatrix} W_1 \\ W'_1 \\ W_2 \\ W'_2 \end{Bmatrix}$$

$$= \begin{Bmatrix} F_1 \\ F'_1 \\ F_2 \\ F'_2 \end{Bmatrix} \qquad (3)$$

The finite element equation for element 2 i.e the first shaft sub-element is given as

$$22.02\begin{bmatrix}156 & 44 & 54 & -26\\ & 16 & -26 & -12\\ & & 156 & -44\\ sym & & & 16\end{bmatrix}\begin{Bmatrix}\ddot{W}_2\\ \ddot{W'}_2\\ \ddot{W}_3\\ \ddot{W'}_3\end{Bmatrix}+0.242x10^{10}\begin{bmatrix}12 & 12 & -12 & 12\\ & 16 & -12 & 8\\ & & 12 & -12\\ sym & & & 16\end{bmatrix}\begin{Bmatrix}W_2\\ W'_2\\ W_3\\ W'_3\end{Bmatrix}$$

$$=\begin{Bmatrix}F_2\\ F'_2\\ F_3\\ F'_3\end{Bmatrix} \qquad (5)$$

The finite element equation for element 3 i.e the second shaft sub-element is given as

$$22.02\begin{bmatrix}156 & 44 & 54 & -26\\ & 16 & -26 & -12\\ & & 156 & -44\\ sym & & & 16\end{bmatrix}\begin{Bmatrix}\ddot{W}_3\\ \ddot{W'}_3\\ \ddot{W}_4\\ \ddot{W'}_4\end{Bmatrix}+0.242x10^{10}\begin{bmatrix}12 & 12 & -12 & 12\\ & 16 & -12 & 8\\ & & 12 & -12\\ sym & & & 16\end{bmatrix}\begin{Bmatrix}W_3\\ W'_3\\ W_4\\ W'_4\end{Bmatrix}=$$

$$\begin{Bmatrix}F_3\\ F'_3\\ F_4\\ F'_4\end{Bmatrix} \qquad (6)$$

Applying boundary conditions for a supported end shaft, the displacement W_1 and $W_4 = 0$.

We applied Gaussian elimination technique to eliminate the first and seventh row and column of the coupled mass and stiffness matrices. We obtained reduced matrices.

4. Results and Discussion

The results from solving the eigenvalue problem shows that the maximum and minimum linear displacement of the shaft about it equilibrium position is within the range of (-0.2 to 0.3) m. The mode shapes for this element is gotten from the eigenvectors and displayed in the following figures:

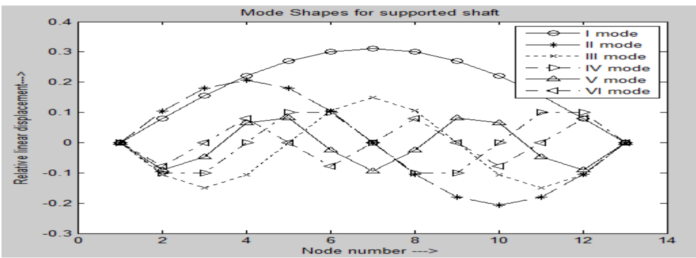

Fig 2: Mode shape for a simply supported shaft

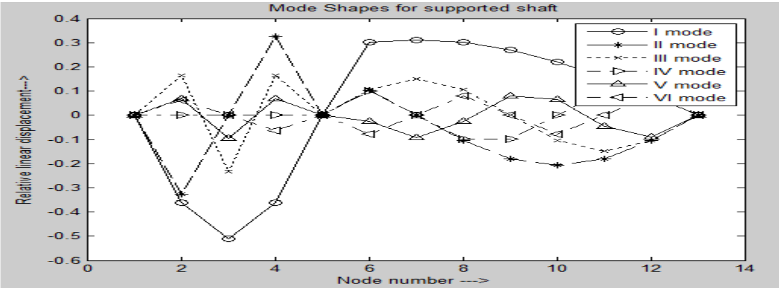

Fig 3: Mode shapes of shaft using four elements

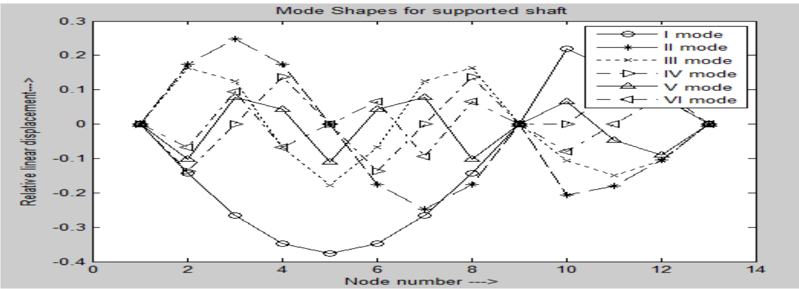

Fig 4: Mode shape of shaft using eight elements

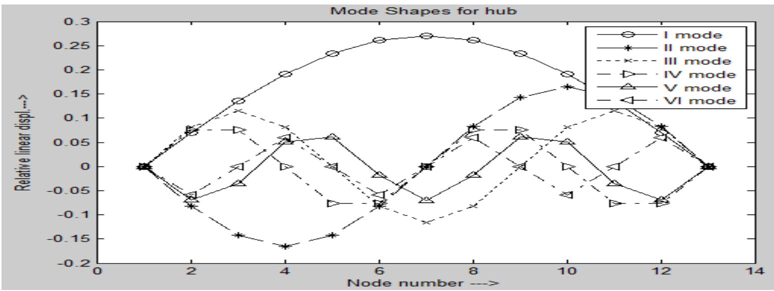

Fig 5: Mode shapes of the hub

Therefore, the number of mode shapes to be plotted depends on the number of natural frequency and degree of freedom. In this case, we have six mode shapes for the shaft. The mode shapes is determined by the Eigen-vectors of the corresponding Eigen frequency. Fig. 1 explains that the linear transformation from a vector space points farthest to the positive x-direction at the first mode and lowest to the negative x-direction at the second mode

We considered normal operating conditions; a without considering the unusual nature of wind gust that would make the wind speed change instantaneously. The material property of the low speed shaft given its geometric features would never get to it critical speed. Degree of freedom W_1 and W_4 were eliminated cause the coupled elements were assumed to be simply supported at its ends. These results compared with other approaches shows greater accuracy and efficiency in describing the structures response at each node compared with other approach. Both linear estimation technique and neural networks used in the analysis of the vibration motion failed in this wise.

Therefore, sub-structuring is an effective method for the dynamic analyses of coupled structures.

Fig. 2 shows the mode shape of the shaft using four sub-elements while Fig. 3 otherwise shows the mode shape when we consider eight sub-elements. This means the greater number of finite elements used increases the accuracy of the results to be determined. This implies that the higher number of elements used increases the compactness in a state space representation. Also, the relative linear displacement is found to decrease as the number of elements to be analyzed increases. It has been shown that through sub-structuring and application of the right boundary conditions, the eigenvalue analysis can be determined for a structural body. This means that the maximum and minimum displacement of the hub is within a range of (-0.15 to 0.4m). This means the hub is more rigid and less susceptible to failure unlike the shaft element. The hub material property has a greater young modulus compared to the shaft material.

5. Conclusion

The results show a good correlation between the dynamic behaviour of the coupled hub and low speed shaft using the MATLAB simulation. The hub and the shaft due to their material and geometric properties would not fail if the rotational speed of the blades do not exceed the lowest Eigen frequency. This is practically so as the rotational speed of the wind turbine blades would normally not exceed 70rad/s.(Siemens wind turbine, 2011). Although, the amount of wind energy captured by the blades is proportional to the height of the tower according to Beaufort scale

Sub-structuring technique is therefore proven to be an effective numerical method for analysing the vibration of a wind turbine compared with other approaches. The reduction of the degree of freedom from eight to six minimized the complexities of matrix computations. It has also increased efficiency and reduced the time for the stiffness and mass matrix computations

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