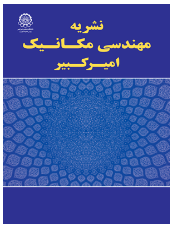

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Out-of-Plane Vibration Mitigation of Wind Turbine Blade Using Highly Efficient Nonlinear Energy Sink

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ABSTRACT: Nowadays, the use of wind as one of the main sources of low carbon and renewable energy is expanding rapidly all around the world. Recently, with the development of wind farms and the increase in the size of wind turbines, the wind loads on them have increased, and as a result, they have become more difficult and expensive to maintain. Therefore, researchers have deeply focused on the analysis and the control of their vibration. In this study, a wind turbine blade with a type of nonlinear absorber, called highly efficient nonlinear energy sink is analyzed, furthermore the interaction between the heavy and long blade and the nonlinear energy sink, under the influence of gravity in the vertical plane and time-dependent wind force, which is due to its height dependency is examined. For this purpose, the equations of motion of the system are obtained by the energy method and solved numerically. The blade-nonlinear energy sink system behavior is compared to that of the blade and linear absorber system. Also, the sensitivity of the parameters affecting the performance of the nonlinear energy sink is analyzed and the vibration of the system with optimized nonlinear energy sink is compared with the alone blade and the blade with the optimal linear absorber behaviors.

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1. INTRODUCTION

Due to the increase in demand for electricity and higher productivity expectations, modern wind turbines are being built in larger dimensions in order to produce more electricity, which may generate higher loads and significant vibrations.

In the literature, considerable attention has been paid to the structural control of wind turbines and to reduce wind vibrations and improve the dynamic response of the structure, various control methods including the use of tuned mass dampers, Tuned liquid column damper and Active vibration control devices have been provided, while nonlinear energy sink has not been employed yet.

The present research has studied the control of out-of-plane vibration of a wind turbine blade, by using a kind of nonliner absorber called highly efficient NonLinear Energy Sink (NES). For this purpose, the turbine blade is considered as an elastic beam rotate in the vertical plane. The model generated in [1] is applied for wind turbine blade structure and the highly efficient NES introduced in Ref. [2] is utilized as an absorber. Accordingly, equations of blade and blade with absorber are extracted using the Euler-Lagrange equation and the effect of NES is compared with the effect of optimal linear absorber [3].

2. EQUATIONS

A wind turbine blade is considered as an elastic beam so that one end is fixed perpendicular to the horizontal main shaft, and the other end is free. The wind blows in the horizontal direction, and the Wind load acts on the rotating blade in out of plane direction, while its force (Q_y) varies depending on the height from the ground.

By deriving kinematic energy, potential energy, and non-conservative forces and applying Lagrange's equation, the equations of motion of a blade with an NES are obtained as:

$$
\begin{aligned}
&(1+4(\alpha_1+\alpha(\delta)^2 fn_{11})z_l^2)\ddot{z}_l+c_l\dot{z}_l-c_{nes}\frac{h\delta}{hl}(\dot{z}_{nes}-\\
&\frac{h\delta}{hl}\dot{z}_l)+[4(\alpha_1+\alpha(\delta)^2 fn_{12})\dot{z}_l^2+2w^2(\alpha_2-\alpha_1 z_l^2)]z_l\\
&+[2w^2(\alpha(\delta)fn_{13}-\alpha(\delta)^2 z_l^2 fn_{14})+1-g(1+fn_{15})\\
&\sin wt]z_l+k_{nes1}\frac{h\delta}{hl}(z_{nes}-\frac{h\delta}{hl}z_l-z_c)-k_{nes2}\frac{h\delta}{hl}\\
&(z_{nes}-\frac{h\delta}{hl}z_l-z_c)^3+k_{nes3}\frac{h\delta}{hl}(z_{nes}-\frac{h\delta}{hl}z_l-z_c)^5\\
&-k_{nes4}\frac{h\delta}{hl}(z_{nes}-\frac{h\delta}{hl}z_l-z_c)^7=Q_c+\Delta Q\sin wt
\end{aligned}
\tag{1}
$$

$$
\begin{aligned}
&fn_{21}\ddot{z}_{nes}+c_{nes}(\dot{z}_{nes}-\frac{h_\delta}{h_l}\dot{z}_l)-k_{nes1}(z_{nes}-\frac{h_\delta}{h_l}z_l-z_c)^7+\\
&k_{nes2}(z_{nes}-\frac{h_\delta}{h_l}z_l-z_c)^3-k_{nes3}(z_{nes}-\frac{h_\delta}{h_l}z_l-z_c)^5+\\
&k_{nes4}(z_{nes}-\frac{h_\delta}{h_l}z_l-z_c)^7=0
\end{aligned}
\tag{2}
$$

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Table 1. Blade Parameters

Parameter	Qty.	Unit
Density	7870	kg/m^3
Module of elasticity	200	GPa
Length	1.0	m
Width	0.04	m
Thickness	0.0025	m
c_l	0.001	N.s/m
Q_c	0.3	N
ΔQ	0.1	N

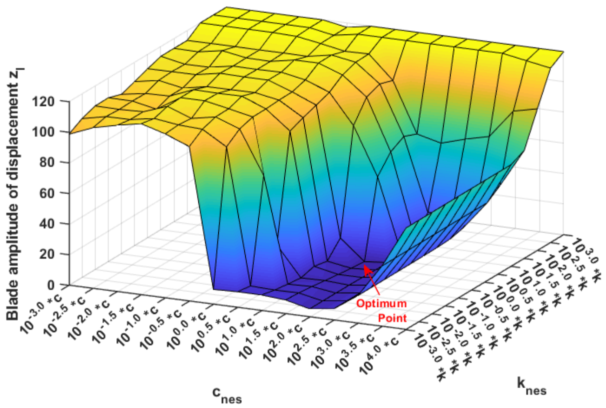

Fig. 1. Blade Vibration amplitude versus stiffness and damping coefficients of NES

While:

$$fn_{11} = m_{nes}\alpha \Big/ \rho A h^2 \tag{3}$$

$$fn_{12} = fn_{14} = fn_{11} \tag{4}$$

$$fn_{13} = m_{nes}\delta / \rho A h \tag{5}$$

$$fn_5 = m_{nes}\alpha(\delta)\alpha_3 / \rho A \beta_5 h^2 \tag{6}$$

$$fn_{21} = m_{nes} / \rho A \alpha_3 \tag{7}$$

Moreover, the equations of motion of the blade and linear absorber are as Eqs. (8) and (9):

$$(1+4(\alpha_1+\alpha(\delta)^2 fn_{11})z_l^2)\ddot{z}_l + c_l\dot{z}_l - c_{abs}\frac{h_\delta}{h_l}(\dot{z}_{abs} - \frac{h_\delta}{h_l}\dot{z}_l) + [4(\alpha_1+\alpha(\delta)^2 fn_{12})\dot{z}_l^2 + 2w^2(\alpha_2 - \alpha_1 z_l^2)]z_l + [2w^2(\alpha(\delta) fn_{13} - \alpha(\delta)^2 z_l^2 fn_{14}) + 1 - g(1+fn_{15})\sin wt]z_l - k_{abs}\frac{h_\delta}{h_l}(z_{abs} - \frac{h_\delta}{h_l}z_l) = Q_c + \Delta Q \sin wt \tag{8}$$

$$fn_{21}\ddot{z}_{abs} + c_{abs}(\dot{z}_{abs} - \frac{h_\delta}{h_l}\dot{z}_l) + k_{abs}(z_{abs} - \frac{h_\delta}{h_l}z_l) = 0 \tag{9}$$

3. SOLVING THE EQUATIONS

The above equations are solved numerically using the Runge Kutta method by considering the following values for the parameters:

Sensitivity analysis of the effect of stiffness and damping coefficients of the NES on the blade displacement is performed to extract the optimal system as shown in Fig. 1. The initial spring stiffness and damping coefficient for the NES are such that the NES linear frequency is equal to the blade frequency and the damping is 10 times the blade damping coefficient.

The frequency response diagrams of the wind turbine blade with NES, the blade with the optimal linear absorber, and the alone blade around the second super-harmonic resonance region are shown in Fig. 2 as:

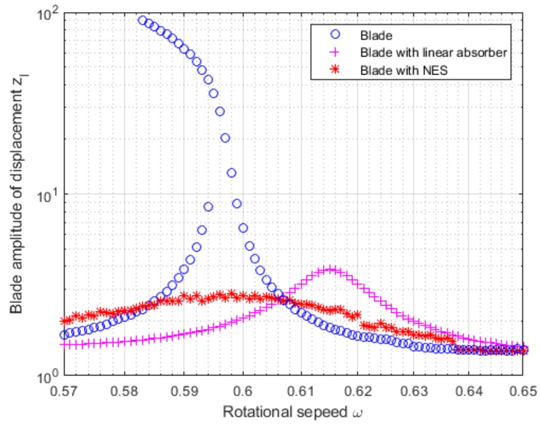

Fig. 2. Frequency response of the blade with NES, optimal linear absorber and without them

Fig. 2 shows that both linear and nonlinear absorbers have significant effects in reducing the amplitude of displacement of wind turbine blade vibration so that its value from 90.01 in the alone blade reduces to 2.81 with NES and 3.85 with the optimal linear absorber. In this way, the NES, due to the use of non-linear springs, in addition to the ability to further reduce in vibration amplitude, causes robust changes.

In the literature, it is shown that the best connection point of the NES is at the location of the maximum amplitude of the structure. However, since the primary system was considered linear in these papers, and the nonlinearity of blade structure as a primary system in this study sensitivity analysis for installing position of NES is performed for more assurance. In this way, the NES is connected to a one-meter-long blade at 0.1 m intervals. As shown in Fig. 3, the maximum absorbent is obtained at a point of 0.9 m from the blade root. However, it can be seen that it is almost similar in the range of 0.7 m to 0.9 m.

4. CONCLUSION

Based on the presented results, the following main points can be concluded as:

A. The use of highly efficient NES could reduce the amplitude of blade vibrations in the resonance conditions by 96% from 90.01 to 2.81.

B. The reduction of blade vibration with NES is 31% more

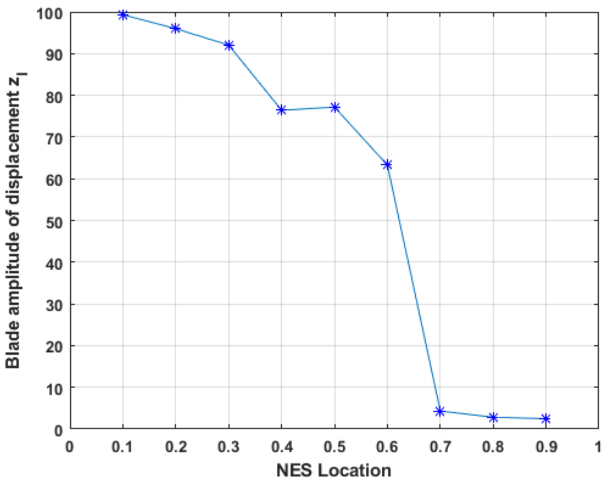

Fig. 3. Maximum vibrations amplitude of blade around the second super-harmonic resonance depending on the location of the absorber on the blade

than when using an optimal linear absorber, which indicates better performance of the NES.

C. The effect of spring stiffness and damping coefficients of NES on its performance is so effective, and therefore, by considering inappropriate parameters, the amplitude of vibration could decrease.

D. The best performance of NES is achieved when it is installed at the farthest possible distance from the root of the blade.

REFERENCES

[1] T. Inoue, Y. Ishida, T. Kiyohara, Nonlinear vibration analysis of the wind turbine blade (occurrence of the superharmonic resonance in the out of plane vibration of the elastic blade), Journal of vibration and acoustics, 134(3) (2012).

[2] M.A. Al-Shudeifat, Highly efficient nonlinear energy sink, Nonlinear Dynamics, 76(4) (2014) 1905-1920.

[3] J.P. Den Hartog, Mechanical vibrations, Courier Corporation, 1985.

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