Vibration suppression effects on rotating wind turbine blade using a particle damping method

Santosh R. Sandanshiv[1]**, Umesh S. Chavan**[2]
[1]Genba Sopanrao Moze College of Engineering, Pune, India
[1, 2]Vishwakarma Institute of Technology, Pune, India
[1]Corresponding author
E-mail: [1]*santoshsandanshiv@yahoo.com*, [2]*umesh.chavan@vit.edu*

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Abstract. Due to the vibration of the wind turbine blade, the rate of electricity generation gets reduced. In this research, the focus is on reduction of wind turbine blade vibration. The important point is that in the attempt of vibration suppression, a new method of damping named particle damping has been tried. The novelty of this study is that this method is adopted for the first time in wind turbine blade for rotating condition. In this method, containers filled with spherical particles are mounted at four different positions on each blade alternatively. Taking tests at different rpm and container positions gives a different vibration suppression effect as compared to without damping and finding optimum positions for mounting of damper.

Keywords: wind turbine blade, vibration suppression, damper.

1. Introduction

Control of vibration in blade is necessary as it adversely affects electricity generation. According to Thomsen [1], the main modes of vibration in the blade are edgewise and flap wise. According to Dapeng [2], edgewise vibration is the main problem in most of the blades. Giguere [3] gives a characteristic which provides important findings of dynamic characteristics. Mainly load acting on the blade is wind load and many scientists have already worked by blade element momentum method (BEM) for calculating aerodynamic load of a blade. Extreme wind turbine load investigation, using different methods, are studied by Saranyasoontorn [5]. Typhoon winds are critically analyzed for the investigation of turbulent conditions, Ishizaki [4]. The damage of structural parts like nacelle cover and blades of wind turbine are investigated by Maalami [6] and Duquette [7]. Passive damper is inserted in wind turbine tower for minimizing vibration induced by wind loads, Murtagh [8]. Krenk [11] introduces active struts mounted near the root of each blade for reducing blade vibrations. For mitigating edgewise vibrations active tuned mass damper is investigated by Fitzgerald [10]. A roller damper and a tuned liquid column damper (TLCD) inside a rotating blade are introduced by Box and Khan [11, 12]. For multi mode vibration reduction of offshore wind turbine under seismic excitation, Hussan [13] introduces a multiple tuned mass damper (MTMD) technique.

The use of particle damping in wind turbine blade is not much explored in as found through the relevant literature. Therefore, in this research the focus is on the same.

2. Particle damping method

The use of a particle damping method is based on the ability of contact interactions using a small number of parameters that capture the most important contact properties. Forces between cavity walls and individual particles are calculated based on force-displacement relations.

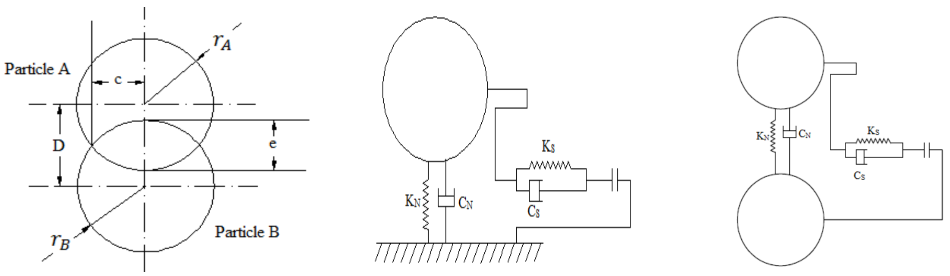
Forces created due to particle-cavity and particle-particle impacts are the main aspect for modeling. Spherical particles, A and B with radii r_A and r_B with particle centres separated by distance D is shown in Fig. 1(a). At approach e is positive, at that time two particles interact with each other. The approach can be defined as:

$$e = (r_A + r_B) - D, \tag{1}$$

contact forces between two colliding balls become:

$$\vec{f} = f^n \vec{N}^n + f^s \vec{N}^s, \tag{2}$$

where, f^n – normal force, f^s – shear force, $\vec{N}^n$ – unit vector in normal direction and $\vec{N}^s$ – unit vector in shear direction. C_N, C_S, K_N, K_S damp constants and stiffness respectively.



a) b) c)

Fig. 1. a) Particle-particle impact parameters, b) ball-wall spring mass diagram, c) ball-ball spring mass diagram

3. The experimental setup

Fig. 2 shows a block diagram of the experimental set-up. Three blades having 1525 mm length each are mounted on flange connected to a shaft. The shaft is supported with a bearing, considering that all vibrations of the blade are transmitted through the flange and shaft to the bearing. Three-directional accelerometer is mounted on the bearing and it transmits signals to FFT analyzer (Bruel and Kjaer) and further from the analyzer to the display unit. Fig. 3 shows 9 mm spherical ball, particle damper container. Fig. 4 shows the experimental set-up of 1 kW wind turbine (a) FEA Model (b) test set-up. In all tests, considering the size of spherical particles as 9 mm having material containing chemical compositions of 0.010 % Mo, 0.050 % Ni ,0.98 % C, 0.33 % Mn, 0.25 % Si, 0.010 % S, 0.012 % P, 1.40 % Cr. Container diameter is 48 mm, and height is 28 mm having material of Poly-propylene (PP).

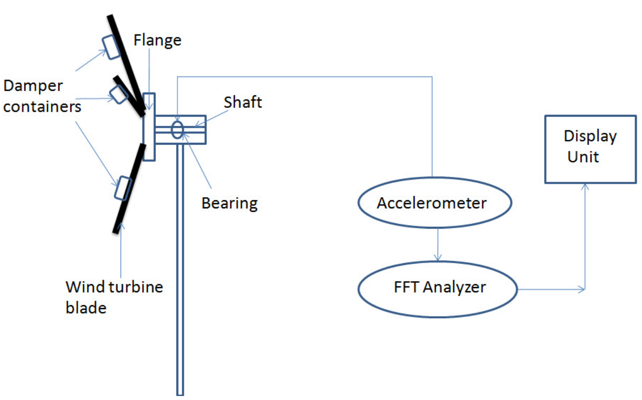


Fig. 2. Block diagram of the experiment set-up

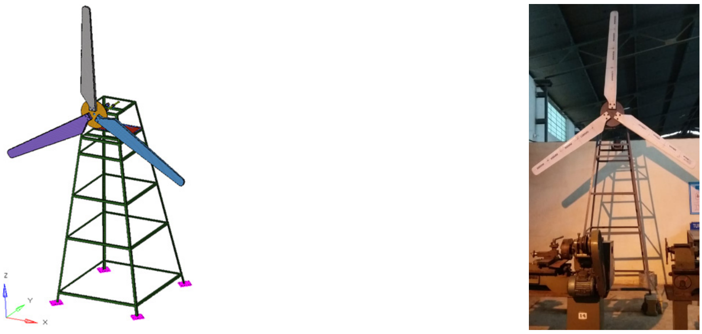
Fig. 3. Container with particle damper

3.1. Parameters for testing

Position of Damper: Fig. 5(a) to (d) show a particle damper mounted on four positions on each blade at the difference of 300 mm from the tip of the blade as 300 mm, 600 mm, 900 mm and

1200 mm respectively. At each position, a container is mounted externally on all the three blades at the same time.

RPM of Blade: In this research, three different rpm i.e. 60 rpm, 70 rpm and 80 rpm are kept. Therefore, the first takes the readings on without damping conditions and compares results with with-damping conditions.



a) b)

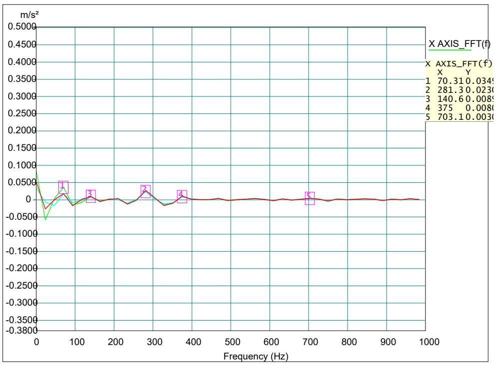
Fig. 4. Experimental set-up of 1 kW wind turbine a) FEA model and b) actual experimental set-up

a) b) c) d)

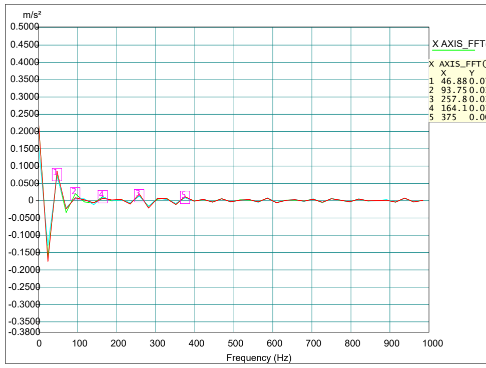
Fig. 5. a) Damper at 300 mm, b) damper at 600 mm, c) damper at 900 mm, d) damper at 1200 mm

4. Results and discussion

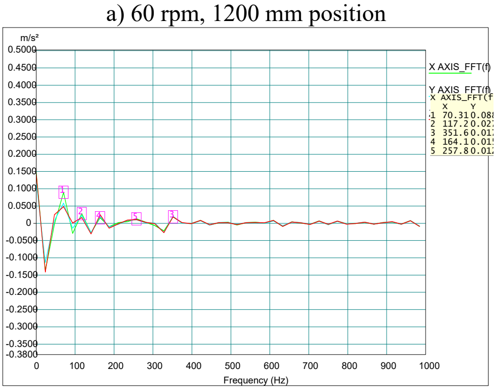
FEA model of the test set-up is prepared using hypermesh software by taking 2D mixed mesh and 3 D tetra mesh elements having 161073 nodes, 65761 elements. First, a test was conducted without damping using FEA results were compared with another similar test conducted through experimental set up. For without damping conditions taking reading at 90 rpm randomly gives result as 1.57 m/s^2 at 50.96 Hz from CAE and acceleration of 1.66 m/s^2 at 46.87 Hz from test set-up, which is close to 90% CAE results. Table 1 shows all the testing results with the first two modes for different rpm and position. Fig 6. shows the acceleration versus frequency with frequency range up-to 1000 Hz. Fig. 5(a) to (d) show damper at 300 mm, 600 mm, 900 mm and 1200 mm respectively. Fig. 6 shows testing results of damping with different positions and at different rpms. In all the results, the five peak modes are shown, and their magnitudes are also shown. Fig. 7 shows all results of undamped and damped conditions together for the first two modes.



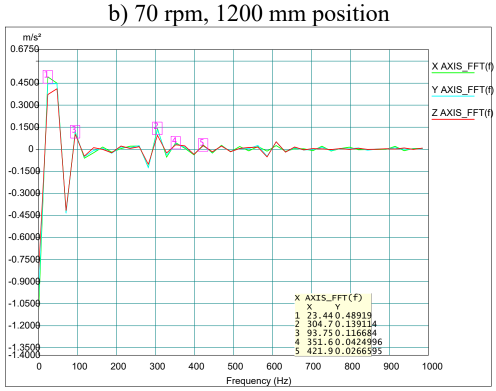
a) 60 rpm, 1200 mm position



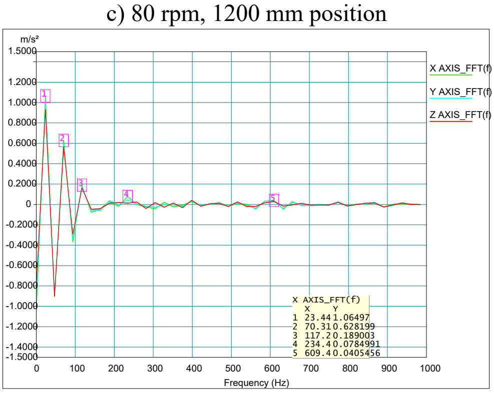
b) 70 rpm, 1200 mm position



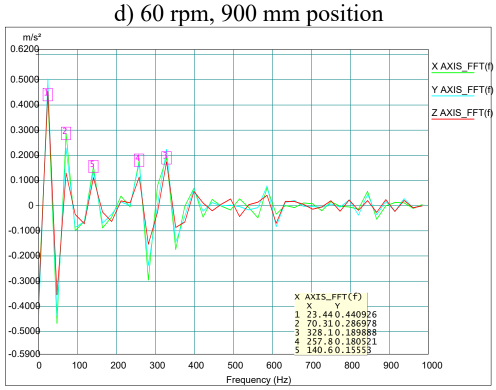
c) 80 rpm, 1200 mm position



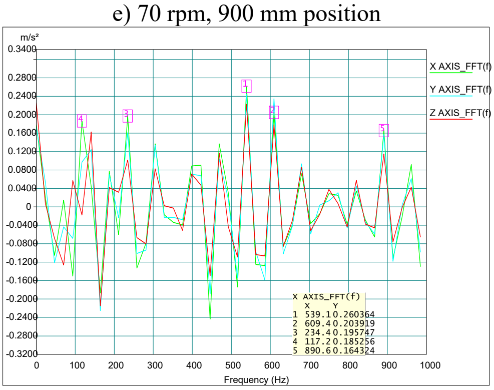
d) 60 rpm, 900 mm position



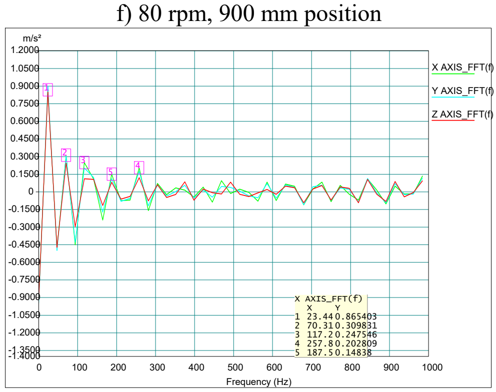
e) 70 rpm, 900 mm position



f) 80 rpm, 900 mm position



g) 60 rpm, 600 mm position



h) 70 rpm, 600 mm position

Fig. 6. Testing results of damping with different positions at different rpm

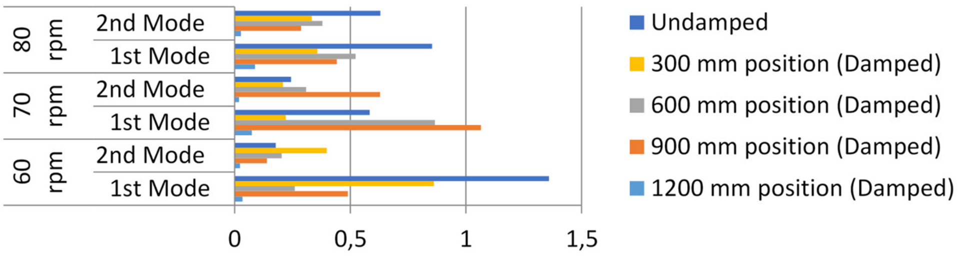


Fig. 7. Combined results of undamped and damped cases

Table 1. Testing results of acceleration (m/s^2) for different rpm and position

Position of damper	60 rpm		70 rpm		80 rpm	
	1st Mode	2nd Mode	1st Mode	2nd Mode	1st Mode	2nd Mode
1200 mm position (Damped)	0.034	0.023	0.074	0.019	0.088	0.027
900 mm position (Damped)	0.489	0.139	1.064	0.628	0.4409	0.2869
600 mm position (Damped)	0.26	0.203	0.865	0.309	0.522	0.379
300 mm position (Damped)	0.862	0.398	0.221	0.208	0.357	0.333
Undamped	1.359	0.177	0.583	0.243	0.853	0.629

5. Conclusions

Comparing dampers for all the four positions, optimum results are received at 1200 mm position location of damper, because at each 60, 70 and 80 rpm difference of acceleration value between with and without damping is greater. Comparing three different rpms, if results are compared, at 60 rpm rotation and 1200 mm position damper location gives good results.

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