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Analysis of Wind Turbine for Vibrational Control Using Single Tune Mass Damper

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ABSTRACT

Global warming and increase in costs of Non renewable energy resources, motivate to explore the new avenues for electricity generation from renewable energy which could be clean, safe and offers a pollution free environment for sustainability. One of the renewable resources of electricity is wind energy. Wind turbines convert the kinetic energy of the wind into mechanical energy. Wind turbines are large and flexible structures and are vulnerable to external vibration sources such as wind local excitations. Vibrations developed in the wind turbine causes structural failures of the turbine which are necessarily to be mitigated with the dynamic responses of wind turbines to ensure the safety of these structures.

The fundamental vibration mode of the wind mill is at the top of the turbine on the wind tower. This is because the intensity of the wind load increases as the height on the tower increases. Wind turbine tower experiences multiple loads such as uniformly varying wind load on the turbine tower and self-weight of the tower etc. Wind load on the turbine tower causes bending stress and self-weight of the turbines causes buckling stress on the turbine. The vibrations of the turbine are to be reduced to ensure the safety of the towers and to improve their service life. These vibrations can be controlled by installing the vibrational control devices.

The present work proposes the use of single turned mass damper (STM) to control Vibrations from the fundamental modes of the wind turbine tower while considering the working parameters such as wind load and the self weight of the turbine. Structural analysis, model analysis and harmonic analysis is performed to find the stresses induced, natural frequencies and displacements of the blade and tower. The effectiveness of the vibrational control is numerically validated.

Keywords:— *Windmill, NACA 63 425 Aero foil blade, Vibration damper (STMD)*

I. INTRODUCTION

Renewable energy is energy that is extracted from un-conventional energy which are naturally replenished on a human timescale, such as sun light, wind, rain, tides, waves, and geothermal heat. Renewable energy often provides energy in four important areas: electricity generation, air and water heating/cooling, transportation, and rural (off-grid energy services.

Renewable energy resources exist over wide geographical areas, in contrast to other energy source, which are concentrated in a limited number of countries. Rapid deployment of renewable energy and energy efficiency is resulting in significant energy security, climate change mitigation, and economic benefits. The results of a recent review of the literature concluded that as greenhouse gas (GHG) emitters begin to be held liable for damages resulting from GHG emissions resulting in climate change, a high value for liability mitigation would provide powerful incentives for deployment of renewable energy technologies. In international public opinion survey there is strong support for promoting renewable sources such as solar power and wind power. At the national level, at least 30 nations around the world already have renewable energy contributing more than 20 percent of energy supply. National renewable energy markets are projected to continue to grow strongly in the coming decade and beyond. Some places and at least two countries, Iceland and Norway generate all their electricity using renewable energy already, and many other countries have the set a goal to reach 100% renewable energy in the future. For example, in Denmark the government decided to switch the total energy supply (electricity, mobility and heating/cooling) to 100% renewable energy by 2050.

Renewable energy systems are rapidly becoming more efficient and cheaper. Their share of total energy consumption is increasing. Growth in consumption of coal and oil could end by 2020 due to increased uptake of renewable and natural gas.

1.1 Wind Turbines

Wind turbines switches the kinetic energy of the wind into mechanical energy. These are of different types based on various energy extraction methods. Overall the aspects of aerodynamics depend largely on the geometry. However there are some fundamentals concepts that can be applied to all turbines. Each topology has a limiting maximum power for a give flow, and certain topologies are superior to others. The method used to extract power has strong influence on this. In general all turbines can be grouped as being lift based, or drag based with the former being more efficient. The deference between these groups is the aerodynamic force that is used to extract the energy. In the wind turbines, different aero foil shapes of the blades effects the drag force and friction force of the air on the blade surface and increases the rotation of the rotor.

1.2 Types of Wind Turbine

1. Horizontal axis wind turbine
2. Vertical axis wind turbine

In this paper the horizontal axis wind turbine is considered as because it gives variable blade pitch, which gives the turbine blades the optimum angle of attack. Allowing the angle of attack to be remotely adjusted gives control, so the turbine collects the maximum amount of wind energy for the time of day and season.

1.3 Identification of the Problem

Wind turbine are large and flexible structures and vulnerable to the external vibration sources such as wind load excitations. Vibration developed in the wind turbine causes structural failures of the turbine. It is necessary ti mitigate the dynamic responses of wind turbine to ensure the safety of these structures. The fundamental vibration mode of the turbine tower in which the maximum displacement occurs is at the top of the tower. Nacelle height.

Wind turbine towers experiences multiple loads such as uniformly varying wind load on the turbine tower self-weight of the tower. Wind load on the turbine tower causes bending stress and self- weight of the the turbine causes buckling stress on the displacement of the turbine tower. The vibration of the turbine is to be reduced for minimum displacement of the tower and to ensure the safety of the tower and improve their service life.

II. Objectives of the Work

The main objective of this work is to control the vibration of the wind turbine tower under multi loading condition such as external wind load along the tower and self-weight of the wind turbine on the top of the tower. The vibrations of the tower can be controlled by installing the controlling device on the tower. The displacement of the tower is reduced in the following manner.

- To determine the natural frequencies of the wind turbine rotor blade at different mode shapes.
- To determine the stresses that are induced in the wind turbine tower under multiple loading conditions i.e. external wind load and self-weight of the turbine
- To find the natural frequencies of the turbine tower at different mode shapes.
- To find the deflection of the turbine tower under multiple loading.
- To control the vibrations of the turbine tower by installing single tuned mass damper.
- To find the effectiveness of the proposed single tuned mass damper.

III. Methodology

- The modeling of Wind Mill was carried out using CATIA V5 software of selected dimensions
- Import the geometry to the ANSYS COMPOSITE PRE-POST software and in the material data the selected material properties are given.
- Meshing of the geometry is done where the whole component is dividing into number of elements so the load is distributed evenly.
- Then the loads and boundary conditions are applied on the wind Mill
- Solving and post processing is done after the model is set up in ANSYS ACP then it can linked to type of analysis we need to solve such as static analysis, modal analysis and buckling analysis.
- Then the results of the analysis can be reviewed in post processing

IV. Design of Wind Turbine

4.1 Design of Blade

The main vibration source of the wind turbine blade is external wind load. The kinetic energy of the wind rotates the wind blade. The rotation of the blade caused vibrations in the wind turbine tower. Wind turbines designed based on the amount of electricity they produced. In this work blade is designed based on the annual power consumption in India for the year 2015-16 is 1075 KWh [15]. The power obtained from kinetic energy in air stream is determined by,

$$P_w = C_p \frac{\rho}{2} \pi R^2 V_w^3$$

Where, C_p = Power Coefficient (0.40) ρ = Density of the air (1.23 kg/m^3) $R =$ Radius of the rotor blade $V_w =$ Wind velocity (6m/s) P_w = Per capita power consumption per annum (1075 KWh)

The average annual wind velocity in Visakhapatnam region is $V_w = 6\frac{m}{s}$.

When the wind power affects in interval

Δt, the wind energy is, $E = P_w.\Delta t$
The annual electrical energy that is obtained from the wind power plant within a year by neglecting wind characteristics, capacity and availability factor.

$$E_{year} = C_p \frac{\rho}{2}\pi R^2 V_w^3 * 8760$$

$$4300 * 100(Wh) = 0.40 * \frac{1.23}{2} * \pi * R^2 * 6^2 * 8760$$

$$t_{opt}(r) = \frac{1}{3}\frac{16\pi}{1.3} r \sin^2\left[\frac{\frac{1}{3}\arctan 10}{7r}\right]$$

$$\frac{d^2}{dx^2}\left\{EI(x)\frac{d^2 Y(x)}{dx^2}\right\} = \omega^2 m(x)Y(x)$$

Boundary conditions for a cantilever beam,

$$at\ x = 0, y(x) = 0, \frac{dY(x)}{dx} = 0$$

$$at\ x = l, \frac{d^2Y(x)}{dx^2} = 0, \frac{d^3Y(x)}{dx^3} = 0$$

For a uniform beam under free vibration from equation

$$\frac{d^4Y(x)}{dx^4} - \beta^4 Y(x) = 0$$

The natural frequency, ω_n from above equation of motion and boundary conditions can be,

$$\omega_n = \beta_n^2 \sqrt{\frac{EI}{\rho A L^4}}$$

Where β_n = 1.875, 4.694, 7.885

Design wind speed

$$(V_z) = K_1 * K_2 * K_3 * [(V]_w)^2$$

Where,

$K_1 = Risk\ Co-efficient\ (1.05)$
$K_2 = Terrain\ factor$
$K_3 = The\ ground\ is\ assumed\ to\ be\ plain,$

$so\ the\ topography\ factor\ is:\ K_3 = 1 + C_s.$
Where,

$$C_s = \frac{Z}{L} = 0\ (\because mean\ surface\ height\ z\ is\ 0)$$

$$K_3 = 1$$

$$(V_z) = 1.05\ x\ 1.01\ x 1\ x\ 50 = \frac{53.025m}{sec}$$

Design wind load including carriage weight =

$$0.6\ x\ V_z^2 = 0.6\ x\ (53.025)2 = 1687N$$

w- 1687 N, Assume D/d = 1.077,

D= Base Diameter,

d- Top Diameter.

4.2 Design of Tower

$$\frac{M}{I} = \frac{\sigma}{y}$$

Bending Moment (M) =

$$W * \frac{L}{2} = 1687 * 6 = 10.12KN - m$$

Moment of inertia (I) =

$$\frac{\pi}{64}(d_o^4 - d_i^4)$$

$$= \pi/64\ [(1.0.77d)]^{\uparrow}4 - d^{\uparrow}4) = 1.718\ x\ [10]^{\uparrow}(-5)\ d^{\uparrow}4$$

Assume steel Young modulus (E) = 210 GPa, yield Strength (Sy)= 240 MPa.

$$P_{cr} = \frac{\pi^2 EI}{L_e^2}$$

Slenderness ration for long column is,

Slenderness ration = L/K

Where,

L = Effective length of the column

K = Radius of Gyration

Radius of gyration (K) =

$$\sqrt{\frac{I}{A}} \quad \sigma_y = \frac{P_{cr}}{\frac{\pi(d_o^4 - d_i^4)}{64}} \ x \ \frac{d_o}{2}$$

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\frac{[(\sigma]_x + \sigma_y)^2}{2} + \tau_{xy}^2}$$

$$\tau_{xy} = 0 \ (\because No\ Shear)$$

$$\sigma_{1,} = \sigma_x$$

$$\sigma_{2,} = \sigma_y$$

From Maximum Distortion Energy Theorem (Vonmises),

$$\sigma_1{}^2 + \sigma_1{}^2 + \sigma_1\sigma_2 \leq \left[[\frac{S_{yt}}{N}\right]]^2$$

4.3 Specification of Blade

The Wind turbine blades have tapered; twisted and asymmetric aero foil shaped cross sections. In this chapter turbine blades of NACA 63 425, DU 91W2250 wind turbine tower.

NACA 63-425 airfoil for the blade, It has a wide range of angle of attack at which it can yield high lift. It also has favorable lift coefficient for wide range of Reynolds's number which would result in steady turbine operations. shows the calculated values of the blade geometry. Aspect ratio 'AR' was chosen to be 6 since a lower value would result in a higher drag while greater values would result in a decreased chord. Decreased chord would result in a lower total lift hence the turbine would produce low power output. Since the span was initially fixed at 1.52 m, the platform area was calculated to be 0.25 m2 while chord came out to be 0.27 m. The rotor disk area was 7.52 m2.

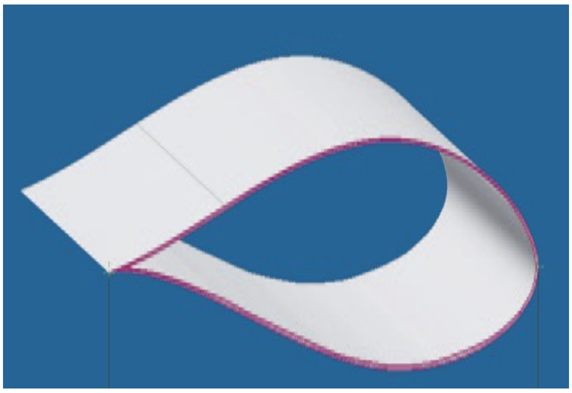

Figure 1: Isometric view of the designed blade using NACA 63-425 airfoil

Table 1: Blade Geometry Values

Blade geometry values

Parameter	Value
Airfoil	NACA 63-425
Aspect ratio—*AR*	6
Span—*b*	1.52 [m]
Planform area—S	0.25 [m^2]
Chord	0.27 [m]
Rotor area—A	7.54 [m^2]
Material	E-fibreglass-epoxy
Surface area	1.71 [m^2]

Table 2: Typical Properties of e-Fiber glass-Epoxy at Room Temperature

Typical properties of E-fiberglass-epoxy at room temperature

Property	Values
Density	1950 kg/m^3
Poisson ratio	0.22
Ultimate tensile strength—longitudinal	724 MPa
Ultimate tensile strength—transverse	70.3 MPa
Ultimate compressive strength—longitudinal	476 MPa
Ultimate compressive strength—transverse	227 MPa
Young's modulus—longitudinal	29 GPa
Young's modulus—transverse	12.6 GPa
Shear modulus	3.5 GPa

V. ANALYSIS OF WIND TURBINE

5.1 Analysis of Blade

5.1.1 Static model of NACA 63 425

Modal analysis is an important tool in vibration analysis, Identification of the natural frequencies and mode shapes of the structure is the primary and important objective of modal analysis. This chapter presents the calculation of natural frequencies of horizontal axis wind turbine blade of NACA 63 425

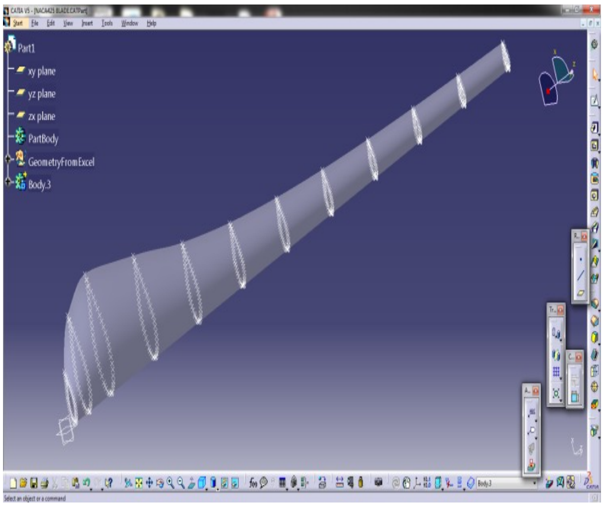

Figure 2: Static Model of NACA 63 425 blade

5.1.1.1 Mode Shapes

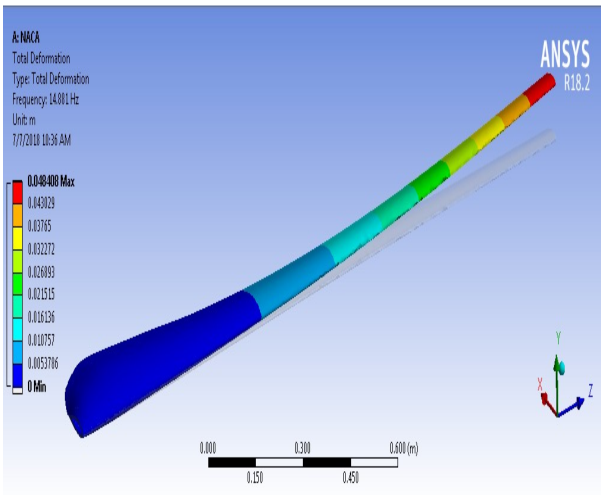

Figure 3: Numerical Natural Frequency of NACA 63 425 at 1st Mode Shape is 14.88 Hz.

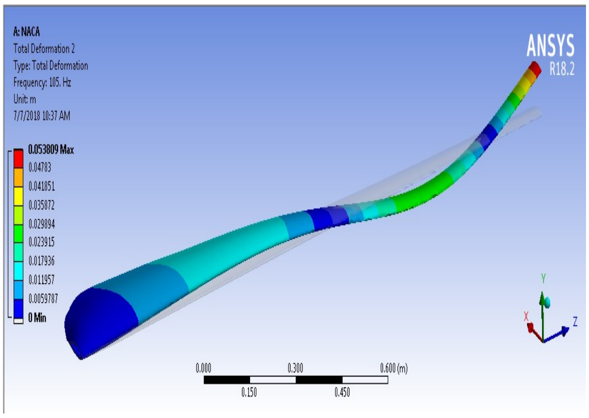

Figure 4: Numerical Natural Frequency of NACA 63 425 at 2nd Mode Shape is 105Hz.

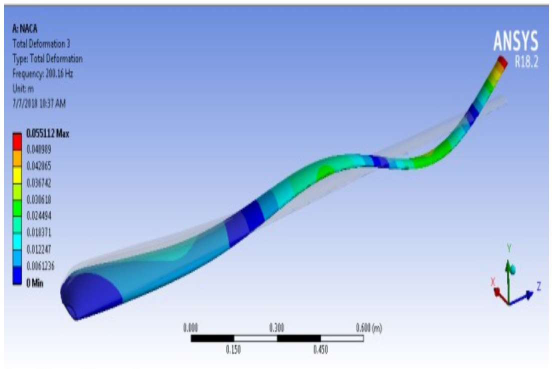

Figure 5: Numerical Natural Frequency of NACA 63 425 at 3rd Mode Shape is 200.16Hz

5.1.2 Static model of DU 91 W2 250

Modal analysis is an important tool in vibration analysis, Identification of the natural frequencies and mode shapes of the structure is the primary and important objective of modal analysis. This chapter presents the calculation of natural frequencies of horizontal axis wind turbine blade of DU 91 W2 250

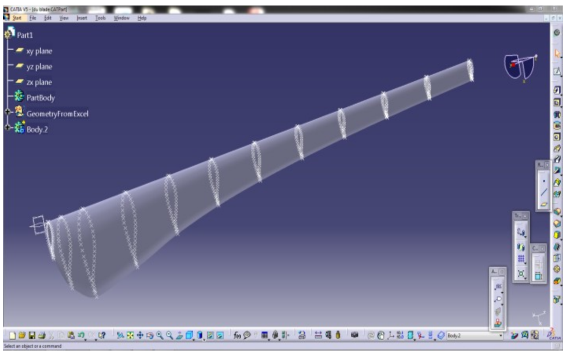

Figure 6: Static Model of DU 91 W2 250

5.1.2.1 Mode Shapes

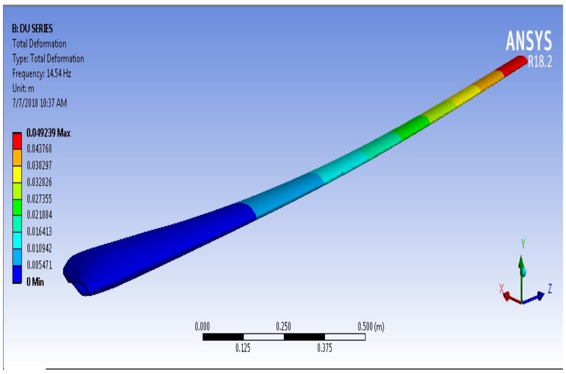

Figure 7: Numerical Natural Frequency of DU 91 W2 250 at 1st Mode Shape is 14.54 Hz

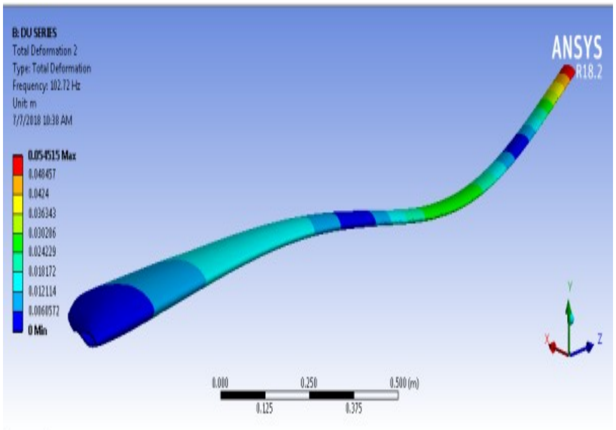

Figure 8: Numerical Natural Frequency of DU 91 W2 250 at 2nd Mode Shape is 102.72 Hz.

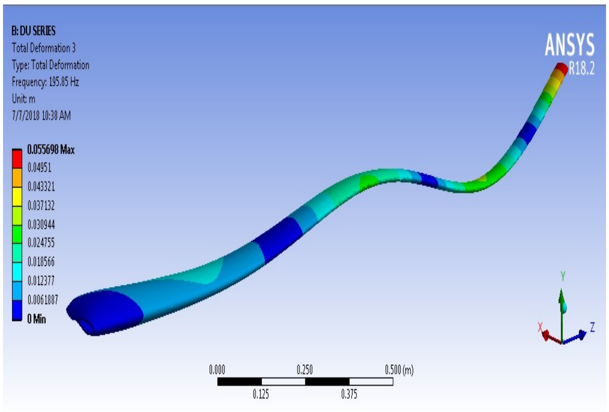

Figure 9: Numerical Natural Frequency of DU 91 W2 250 at 3rd Mode Shape is 195.85 Hz.

5.2 Analysis of Tower

Modal analysis of the turbine tower gives the frequencies of the turbine tower at which it fails due to resonance.

5.2.1 Model Shapes

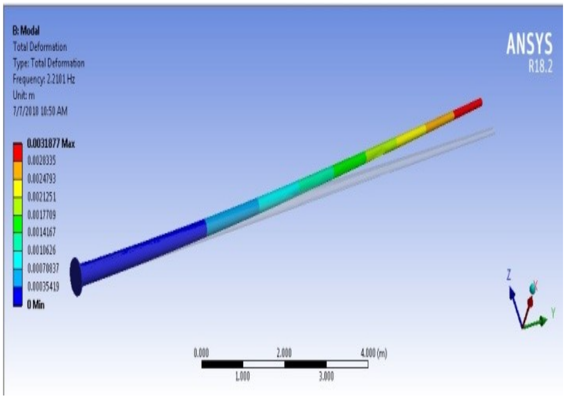

Figure 10: Natural Frequency of the Tower at 1st Mode Shape is 2.21 Hz

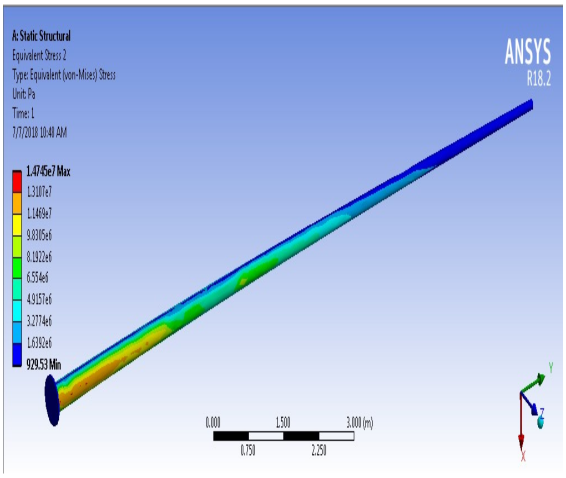

Figure 11: Natural Frequency of the Tower at 2nd Mode Shape is 9.88 Hz

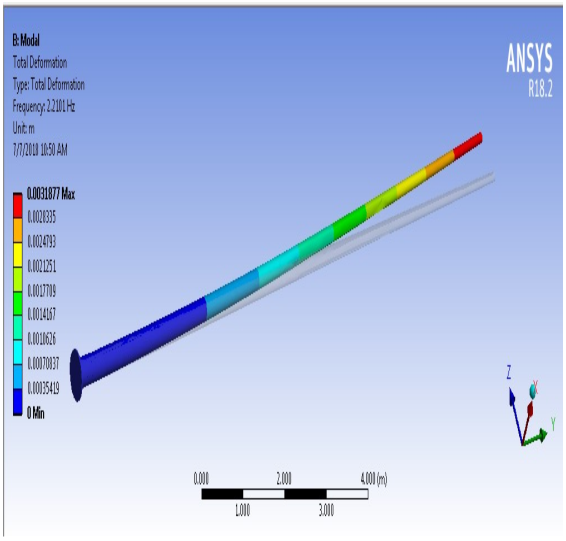

Figure 12: Natural Frequency of the Tower at 3rd Mode Shape is 10.26Hz

VI. RESULTS AND DISCUSSIONS

6.1 Theoretical Results

Table 3: Theoretical frequencies of NACA 63 425 and DU 91 W2 250 blade

Modal shape	Theoretical Frequency(Hz)	
	NACA63425	DU91 W2 250
Modal Shape1	13.85	13.45
Modal Shape2	96.86	93.18
Modal Shape3	181.30	176.79

6.2 Comparison of Theoretical and Ansys Results

Table 4: Theoretical and Ansys frequencies of NACA 63 425 BLADE

Mode Shape	Natural Frequency (Hz)	
	Theoretical	Ansys
Mode Shape 1	13.85	14.88
Mode Shape 2	96.86	105.0
Mode Shape 3	181.30	200.16

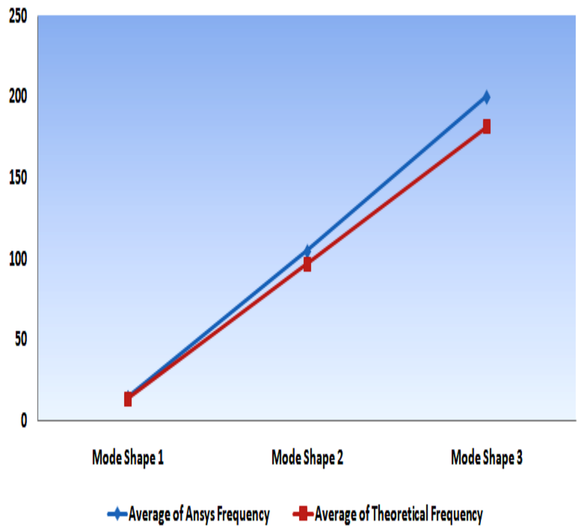

Figure 13: Frequency Variation with Different Mode Shapes of NACA 63 425

Table 5: Theoretical and Ansys Frequencies of DU 91 W2 250 Blade

Mode Shape	Natural Frequency (Hz)	
	Theoretical	Ansys
Mode Shape 1	13.43	14.54
Mode Shape 2	93.18	102.72
Mode Shape 3	176.79	195.85

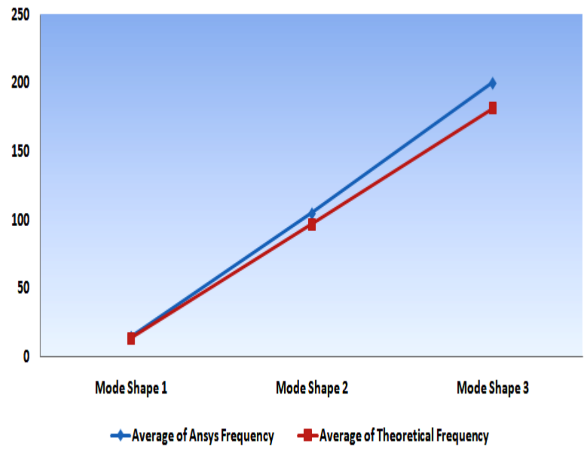

Figure 14: Frequency variation with Different Mode Shapes DU 91 W2 250 Blade

6.3 Harmonic Analysis of Tower

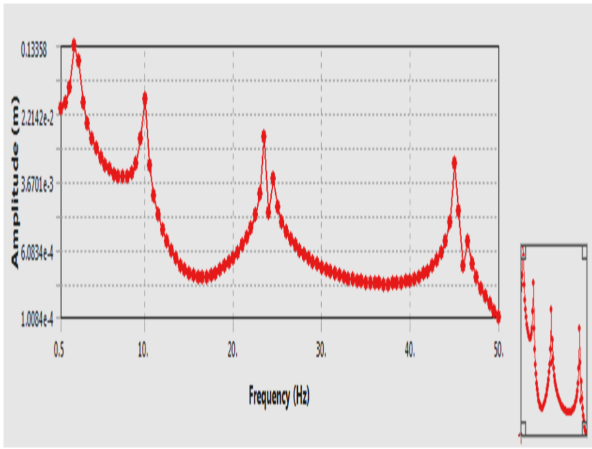

Figure 15: Frequency Response of the Tower without STMD

From the graph we observed that amplitude value at 0.6 Hz frequency is very high so that Design modification of tower done with single tune mass damper is used to suppress the particular amplitude of 0.6 Hz frequency.

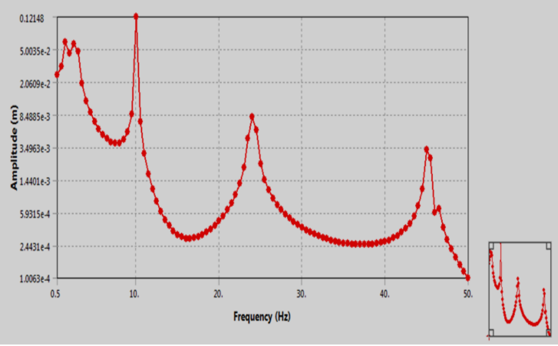

Figure 16: Frequency Response of the Tower with STMD

Table 5: Single Tuned Mass Damper Reduced Tower Displacement

Type	Without STMD	With STMD	% Reduction
Displace-ment (mm)	133	80	40.33

VII. CONCLUSION

The design and modal analysis of the wind turbine blades of NACA 63425 and DU 91 W2250 series are carried out. The use of single turned mass damper to control the vibrations of the wind turbine tower when they are subjected to the combine external wind load and the self weight of the tower. Numerical analysis is performed to examine the effectiveness of the singe tuned mass damper.

The natural frequencies of the wind turbine blades of NACA 63425 blade are 14.88Hz, 105 Hz,200.16 Hz and the DU blade under extreme under extreme wind loads.

The static structural analysis is performed on the turbine tower. The Vonmises stresses developed on the tower due to external wind load and the self-weight of the tower is 14.75 MPa.

The modal analysis of the turbine tower and its natural frequencies are ground to be 2.21 Hz at 1st mode, 9.88Hz at 2nd mode and 10.26Hz at 3rd mode.

The displacement of the turbine tower without single tuned mass damper is 133.58mm and with single tuned mass damper is 80mm. The displacement of the turbine tower is effectively reduced by the installation of the single tuned mass damper.

The effectiveness of the single turned mass damper is numerically investigated. The vibrations developed in the wind turbine tower are reduced by 40% by using single tuned mass damper. The reduction in the vibration increases the service life of the wind turbine tower.

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