STRESS AND VIBRATION ANALYSIS OF A GAS TURBINE BLADE WITH A COTTAGE-ROOF FRICTION DAMPER USING FINITE ELEMENT METHOD

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Abstract

Friction is a very complex phenomenon that occurs between two contact bodies. There are many important applications where the presence of friction is desired. Dry friction is used in such cases as a damping or isolation technique. Turbine blades, built-up structures and transportation systems use friction to enhance their performance. High-cycle fatigue caused by large resonance stresses is one of the main problems in turbine blade design. These stresses can be reduced by using a friction damper.

In this paper the simulation of the turbine blade behaviour is carried out using finite element software ANSYS 10.0 for simulating the blades without and with cottage-roof damper separately. To obtain the analysis initially, simulation of blade without the damper is carried out. In the second phase, simulation of blade with damper is carried out with the help of target elements and contact elements contact is made between the blade plate and the damper surface by creating target surface and contact surface. Since the turbine blade is fixed to the rotor, the nodes at the lower end of the blade are constrained in all DOF. The blade is subjected to a flow of hot pressurized gases which force the rotor to rotate at its required speed. With these boundary and loading conditions at different angular velocities, the displacements and stresses of blade without and with damper are investigated. Vibration behaviour of the blade is also studied by analyzing natural frequencies, mode shapes and frequency response analysis.

Keywords:
Friction damper, Resonance stress, Natural frequencies Mode shape, and Frequency response

1   Introduction

A common failure mode for turbo machinery is high-cycle fatigue of turbine blades, due to high dynamic stresses caused by blade vibration resonance within the operating range of the machinery. There are three methods of lowering the vibratory stresses, namely decreasing the force of excitation, changing the resonance frequency or increasing the damping.

The excitation force is mainly caused by variation of aerodynamic force through the guide vanes on the blades.

Reducing the force of excitation would mean changing configuration of the vanes, which in many cases is not feasible. There is also limited knowledge about how the configuration should be changed to minimize the force of excitation.

Changing resonance frequency might reduce the vibration problem if it is a question of one or a few frequencies. This is possible that too at a few times for turbines which run at constant service speed, such as power generating steam turbines. Avoiding resonance frequencies is not possible for turbines which have variable service speed, e.g. jet engines.

Reducing the vibratory resonance stress is to increase the damping by use of dry friction. Friction damping arises whenever there are two mating surfaces that are rubbing against each other. A friction damping interface may be found where blades are in contact with each other.

There are three types of damping acting on the blades namely material damping, aerodynamic damping and friction damping. Internal material damping has proved to be of less importance in controlling blade vibrations due to the low damping ratio of the materials that may be used. The aerodynamic damping is greatly affected by the aspects of the airfoil. This damping may be negative, i.e., adding energy to the motion of vibration, and causing flutter of the blade. The third mechanism of damping is friction damping.

2   Mechanism of Friction Damping

Friction is a very complex phenomenon that occurs between two contact bodies. Friction has received considerable attention by researchers for several decades. While most mechanical systems look to reduce friction and its effects, there are many important applications where the presence of friction is desired. Dry friction is used in such cases as a damping or isolation technique. Turbine blades, built-up structures and transportation systems use friction to enhance their performance. The inexpensive, environmentally robust nature of friction makes its use advantageous as a passive damping technique. Friction damping takes place wherever two contact surfaces experience relative motion. The centrifugal force arising from the rotation of the disc forces the damper against the contact surface and energy is dissipated when there is slip

motion at the contact interface. Due to its inherently complex nature, implementing friction effectively to damp the vibrations becomes a challenging task. The main reason for this is the non-linear relationship between the normal load and energy dissipation. Fig. 1 shows the typical nature of energy dissipated (E) versus the damper normal load (N) plot.
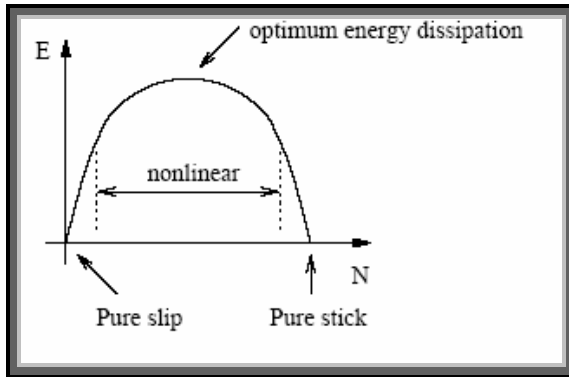

Fig. 1: Energy Vs Normal load

When the normal load is absent, the contact interface experiences pure slip only. Since no work is required to be done against friction, no energy is dissipated. On the other hand, very large normal loads cause the whole contact interface to stick. This results in no energy dissipation again since no relative motion is allowed at the interface. For normal loads that lie between these two extremities, energy is dissipated and the optimum value of energy dissipation lies within this range. However, the behaviour of the contact interface is highly non-linear in this region. This non linearity is due to the variation of friction parameters (e.g. friction force, sliding distances etc.) in space and time. The main difficulty lies in capturing the behaviour of these parameters and here in lies the difficulty of friction damper design.

3 Types of Dampers

The two most common damper types are the Cottage-Roof damper, shown   in Fig. 2 and the Flat Damper, seen in Fig. 3. The Characteristic for these dampers is that the contact surface between the blade and damper is formed as a rectangular area. The flat damper has the contact surface parallel to the vibration motion, while it is not in cottage-roof damper. Both damper types are often used in turbines, but little is known about their damping properties. Some experimental tests have shown that the cottage-roof damper might be better for vibration modes where the blades are in phase. It is believed that the damper will make a rocking motion and thus yield damping.
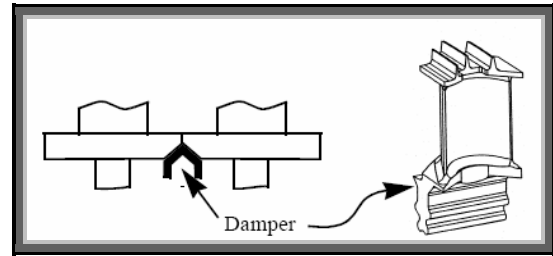

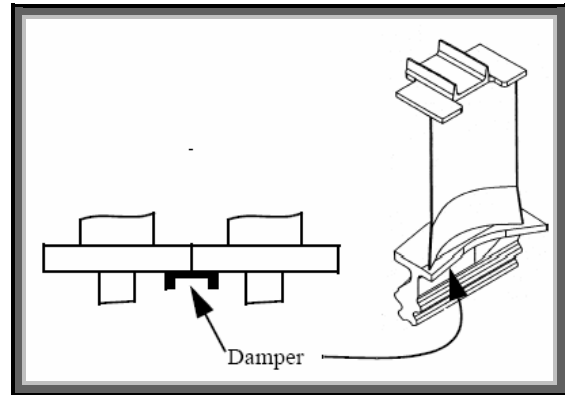Fig. 2: Cottage-roof damper attachment

Fig. 3: Flat damper attachment
Another damper type is the Dog-Bone Damper, shown in Fig. 4. This damper has less stiffness between the contact points than the previous damper types. We have here a Hertzian contact between the damper and the blade.
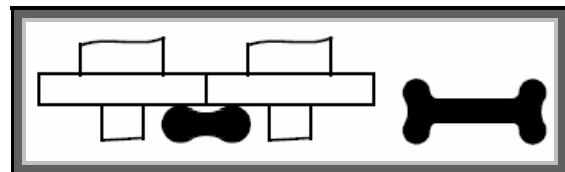

Fig. 4: The dog-bone damper

A damping device that is used mostly in steam turbines is the Lacing Wire. A hole is drilled through each blade airfoil and a wire is drawn through the holes. This wire will make the blade group stiffer, but there will also be slip between the wire and the blade, giving damping. The greatest drawback of this damper is that the wire interferes with the flow around the blade airfoil. This can be avoided if it is possible to put the wire outside the outer platform. An idea of how this is done is shown in Fig. 5.
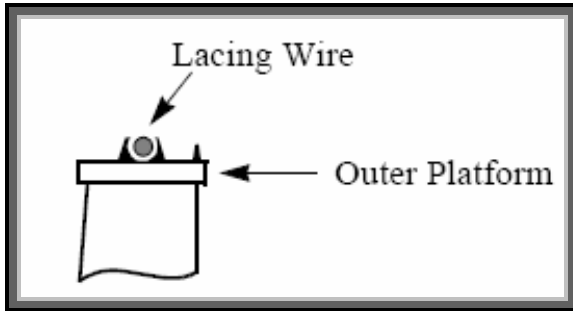
Fig. 5: A lacing wire attached to the outer platform.

4   Platform Damping

The damper, also called an under-platform friction damper, is essentially a piece of metal fitted in a slot between the disk and platforms of two adjacent blades. The damper is pressed against the platforms by centrifugal force as the turbine is rotating. Damping of blade vibrations and energy dissipation occurs when there is slip in the contact between blade and damper.

The contact surfaces where friction damping may occur are shown in Fig. 6. Inner and outer platforms of neighboring blades can come in contact due to thermal expansion or if the centrifugal load makes the blade twist. A damper device can be put under the inner platform.
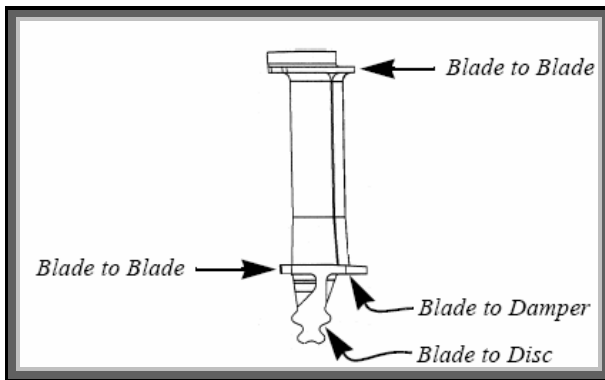

Fig. 6: Friction damping contact surfaces of a
turbine blade.

The analysis can be extremely difficult if all the details of the damper characteristics are to be included in the analysis. The difficulties arise due to many complicated factors like temperature, frequency, and surface roughness effects, the real contact locations and their variation during vibration. In spite of the physical simplicity of these dampers, the effects of these and other factors have not yet been fully understood. Accordingly, based on engineering judgment, some simplifying assumptions, listed below, have been made here in order to reduce the problem to a manageable level.

- Damper flexibility and inertia effects are negligible.
- Damper contact on each side can be represented as a point contact with three translational degrees of freedom.
- Left and right surfaces are identical, and
- The blade motion is harmonic.
- Damper and platform surfaces remain in parallel and in contact at all times.

5   Formulation of the Problem

The present work analyzes the behaviour of a cottage-roof damper present in turbine blading. Its effects on the vibrational modes, frequencies and stresses are investigated at different at different speeds of operation of the turbine rotor. The damper rests on plates projecting on both sides of the blades normal to the blade length. The initial gap between the blade plates and the damper is 0.2 mm, which gets closed progressively as the speed increases from zero value to the specified value.

Assuming turbine blade and damper are made of same material, the material properties of blade and damping plate are listed in Table. 1

Table 1: Blade and damper data

Blade size	150x27x47 mm
Material	Nickel-Chromium Alloy
Element Type	Solid 2-D plane 42, Target 169 and Contact 172
Young's Modulus	206E09 N/m^2
Poisson's ratio	0.3
Density	8900 kg/m^3
Friction coefficient	0.2
Damping co-efficient	0.1

5.1   Finite Element Modeling and Analysis

The simulation of the turbine blade behaviour is carried out using finite element software ANSYS 10.0 for simulating the blades without and with plat-form damper separately. To obtain the analysis initially, simulation of blade without the damper is carried out. In the second phase, simulation of blade with damper is carried out as shown in the Fig. 7 by using contact wizard in ANSYS. With the help of target elements and contact elements contact is made between the blade plate and the damper surface. Plates are integral parts of the turbine blade.

Since the turbine blade is fixed to the rotor, the nodes at the lower end of the blade are constrained in all DOF. The blade is subjected to a flow of hot pressurized gases which force the rotor to rotate at its required speed. The effects of aerodynamic forces are not taken into account. With these boundary and loading conditions at different angular velocities, the displacements and stresses in the cases of blade without the damper and blade with the damper are investigated. Modal analysis is carried out to study the natural frequencies and mode shapes of the blade without and with the damper.

Fig. 7: Meshed and constrained model of the blade with damper

6   Results and Discussions

Displacements, stresses and modal analysis have been carried out of the blade without and with damper.

6.1   Displacements and Stresses

Displacements of the blade without and with the damper have been analyzed for eight different values corresponding to the various operating speeds, angular velocity from 314 rad/sec (3,000 rpm) to 1047 rad/sec (10,000 rpm) and are given in Table 2.

Table-2: Displacements of blade without and with damper

Angular velocity (rad/s)	Max. displacements of blade (m)	
	without damper	with damper
314.16	2.49E-04	6.61E-05
418.88	4.44E-04	1.18E-04
523.6	6.93E-04	1.84E-04
628.32	9.98E-04	2.65E-04
733.04	1.36E-03	3.60E-04
837.76	1.77E-03	4.70E-04
942.48	2.25E-03	5.95E-04
1047.2	2.77E-03	7.35E-04

Deformed shape of the blade at an angular velocity 628 rad/sec (6,000 rpm) is shown in the Fig. 8.

Fig. 8: Deformed shape of the blade with damper at 628 rad/s

A comprehensive picture of the displacement behaviour of blade without and with the damper at different Angular velocities of 314 rad/s, 418, 523, 628, 733, 837, 942 and 1047 rad/sec is shown in Fig. 9. It can be observed that the displacements decrease in the blade with the insertion of the damper as compared to the blade with out damper.
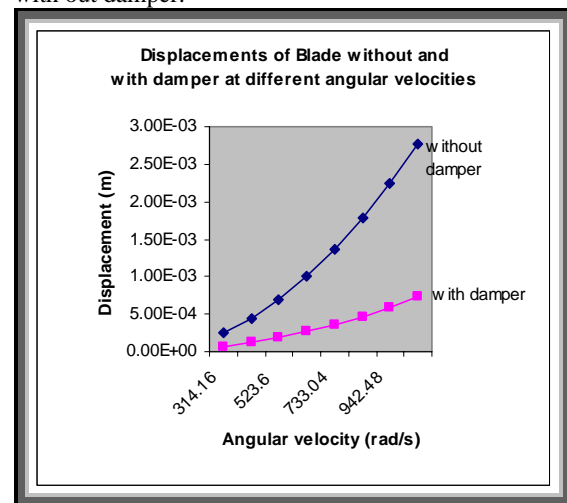

Fig. 9: Displacements of blade without and with damper

The stresses due to the rotation of the turbine blade without and with the damper have been analyzed for eight different load cases from 314 rad/s (3, 000 rpm) to 1047 rad/sec (10, 000 rpm). The maximum von Mises stresses are shown in Table 3.

Table-3: von Mises stresses of blade without and with damper

Angular velocity (rad/s)	Max. von Mises stresses of blade (N/m^2)	
	Without damper	With damper
314.16	2.68E+08	2.10E+08
418.88	4.77E+08	3.74E+08
523.6	7.45E+08	5.84E+08
628.32	1.07E+09	8.41E+08
733.04	1.46E+09	1.14E+09
837.76	1.91E+09	1.49E+09
942.48	2.41E+09	1.89E+09
1047.2	2.98E+09	2.33E+09

The von Mises stress distribution on the blade with damper at an angular velocity 628 rad/sec (6,000 rpm) is shown in the Fig. 10.

A comprehensive picture of the von Mises stress distribution of blade without and with the damper at different Angular velocities 314, 418, 523, 628, 733, 837, 942 and 1047 rad/sec is shown in Fig. 11. It can be observed that the von Mises stresses decrease in the blade with the insertion of damper as compared to the blade with out damper.
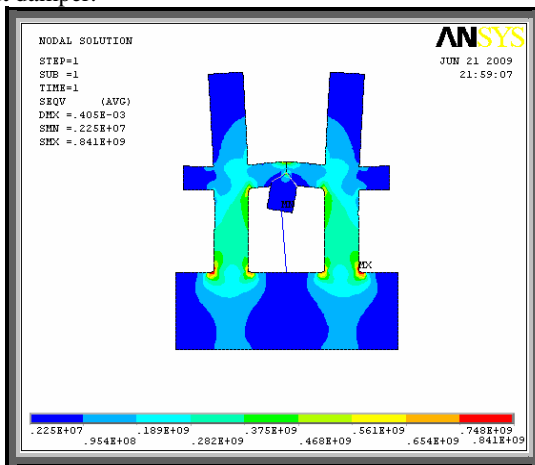
Fig. 10: von Mises stress distribution of the blade at 628 rad/s
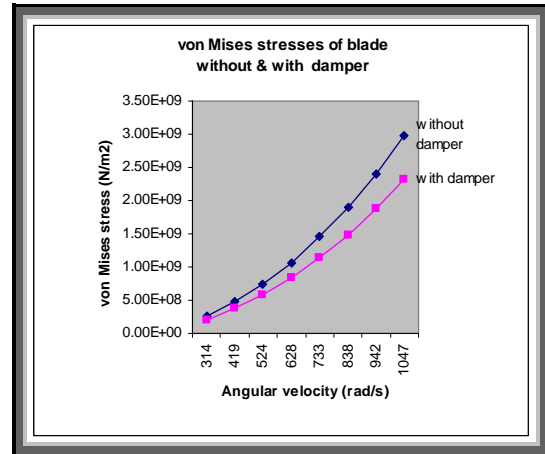
Fig. 11: von Mises stresses of blade without and with damper

6.2 Modal Analysis of Blade Without and With Damper

One of the most common failure modes of turbine blades in turbo machinery is due to high cycle fatigue. The resonant stresses developed due to blade vibration cause the blade failure through fatigue. The blade vibration characteristics are determined by the natural frequencies and the corresponding mode shapes. For the blade without and with damper, under free standing, natural frequencies are given in Table 4.

The comprehensive behaviour of the natural frequencies of blade without and with damper under free standing state is shown in the Fig. 12

Table-4: Natural frequencies of blade at no load

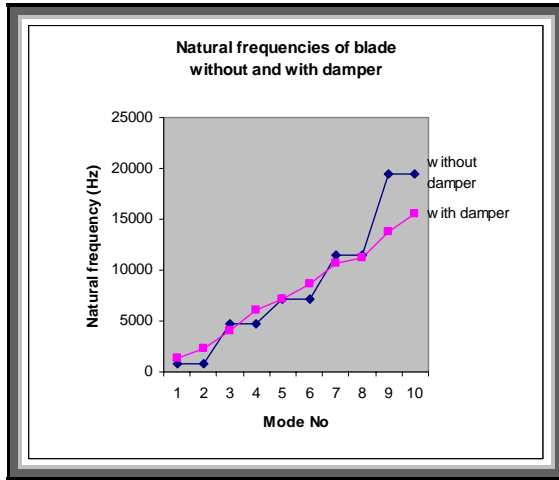| Mode No	Natural frequencies of blade (Hz)	
	without damper	with damper
1	828.83	1337.2
2	829.02	2264.5
3	4671.1	4087.5
4	4673.2	6131.6
5	7105.8	7125.9
6	7114	8658.3
7	11472	10671
8	11483	11196
9	19415	13844
10	19476	15540

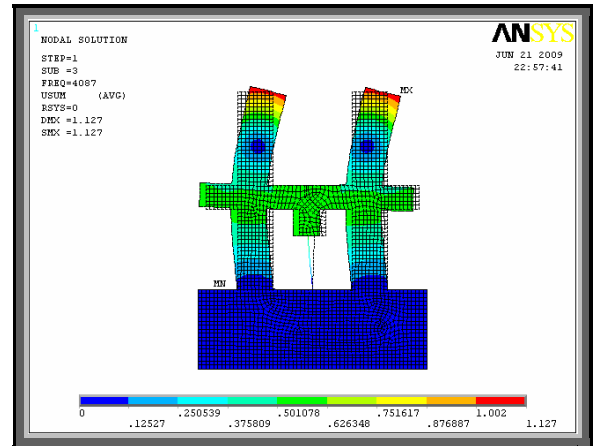Fig. 12: Natural frequencies of blade without and
with damper at no load

6.2.1   Mode Shapes of the Blade under No Load Conditions

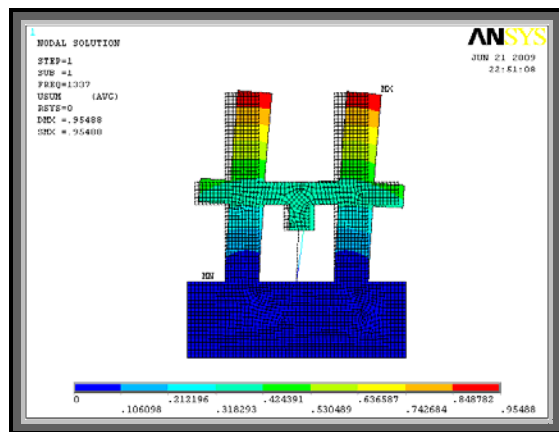Mode shapes of the blade with the damper at no load conditions is shown from Fig. 13 to 22
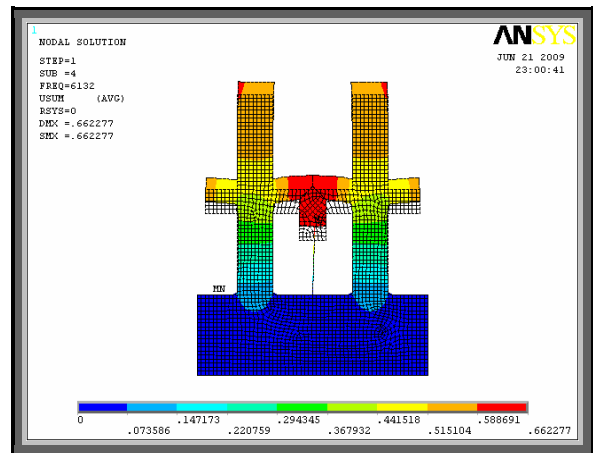

Fig. 13: Mode shape 1

Fig. 14: Mode shape 2
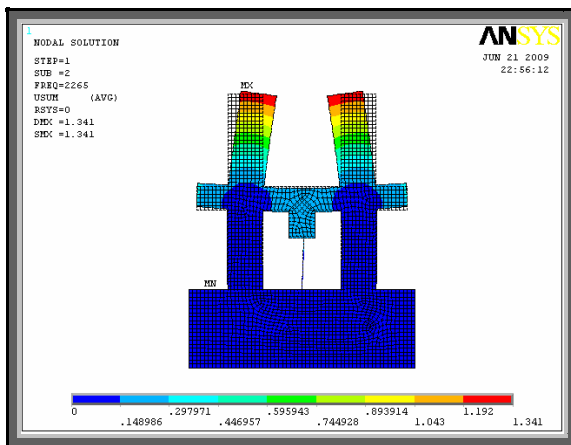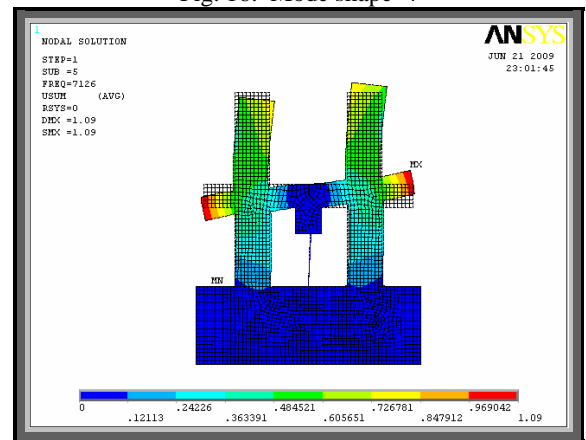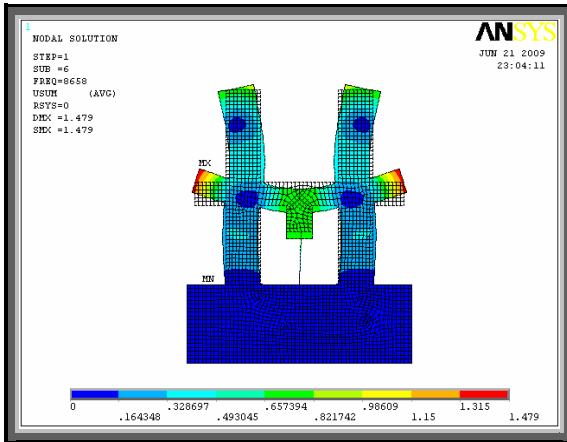

Fig. 15: Mode shape 3

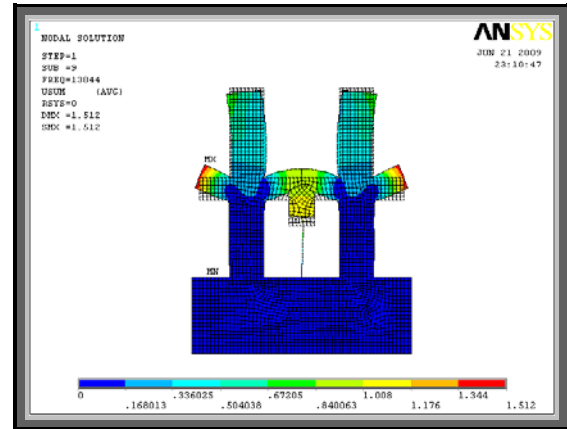Fig. 16: Mode shape 4

Fig. 17: Mode shape 5

Fig. 18 Mode shape 6
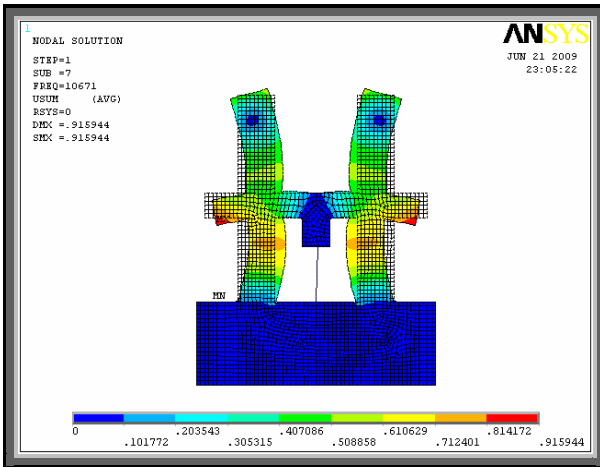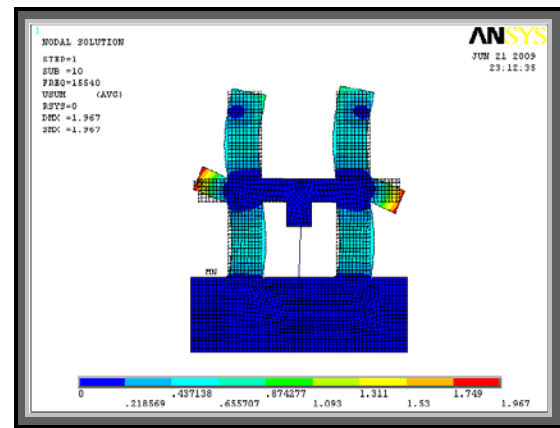

Fig. 19: Mode shape 7

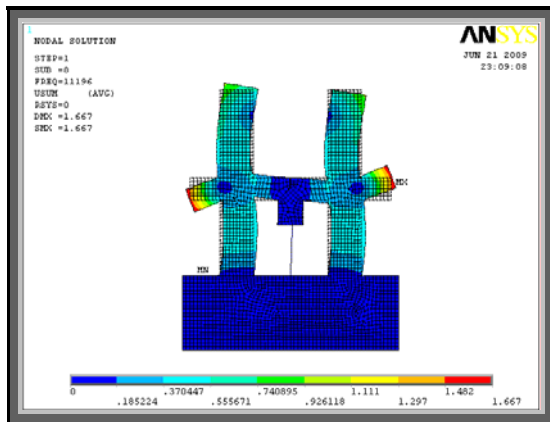Fig. 20: Mode shape 8

Fig. 21: Mode shape 9

Fig. 22: Mode shape 10

For the blade without and with damper, under loaded conditions for angular velocity 314 rad/s, 523, 733 and 1047 rad/sec the natural frequencies are given in Table 5 and Table 6

The comprehensive behaviour of the blade without and with damper at different loading conditions is shown in the Fig. 23 and Fig. 24

Table-5: Frequency analysis of blade without damper

Mode No	Frequency analysis of blade with out damper (Hz)				
	No Load	Omega 314 (rad/s)	Omega 523 (rad/s)	Omega 733 (rad/s)	Omega 1047 (rad/s)
1	828.83	837.98	854.02	877.49	925.35
2	829.02	838.18	854.21	877.69	925.54
3	4671.1	4679.7	4695.1	4718.1	4766.5
4	4673.2	4681.9	4697.3	4720.3	4768.7
5	7105.8	7106.7	7108.2	7110.4	7115.2
6	7114	7114.8	7116.3	7118.6	7123.4
7	11472	11482	11499	11525	11580
8	11483	11493	11510	11536	11592

| 9 | 19415 | 19426 | 19446 | 19476 | 19538 |
| 10 | 19476 | 19488 | 19508 | 19538 | 19603 |

Table-6: Frequency analysis of blade with damper

Mode No	Frequency analysis of blade with damper (Hz)				
	No Load	Omega 314 (rad/s)	Omega 523 (rad/s)	Omega 733 (rad/s)	Omega 1047 (rad/s)
1	1337.2	1343.1	1353.5	1368.9	1401.1
2	2264.5	2271.1	2282.6	2299.8	2335.9
3	4087.5	4095.2	4109.2	4130.2	4174.4
4	6131.6	6132.7	6134.6	6137.5	6143.7
5	7125.9	7128	7131.9	7137.6	7149.7
6	8658.3	8664	8675.7	8692.2	8727.1
7	10671	10675	10684	10697	10724
8	11196	11202	11212	11227	11259
9	13844	13846	13850	13855	13866
10	15540	15546	15556	15573	15606

Fig. 23: Frequency analysis of blade without damper

Fig. 24: Frequency analysis of blade with damper

7   Harmonic Analysis

Harmonic response analysis is a technique used to determine the steady-state response of a linear structure to loads that vary sinusoidally (harmonically) with time.

Harmonic response analysis gives the ability to predict the sustained dynamic behavior of structures, thus enabling to verify whether or not the designs will successfully overcome resonance, fatigue, and other harmful effects of forced vibrations.

Harmonic analysis of the blade with damper at 314 rad/sec (3000 rpm) is investigated. Frequency response and amplitude Ux and Uy at typical locations from bottom to the top of the blade is shown in the Fig. 25 and Fig. 26
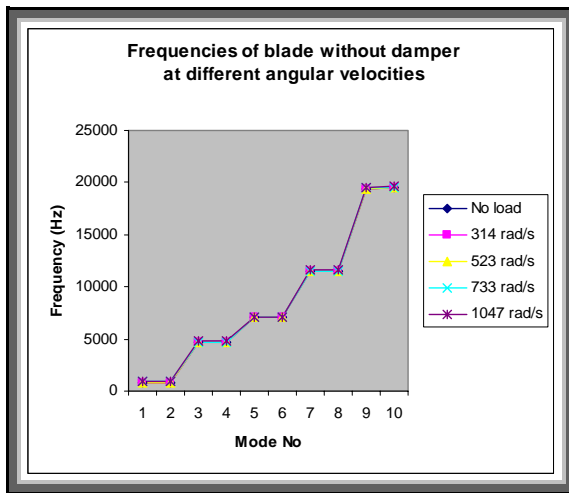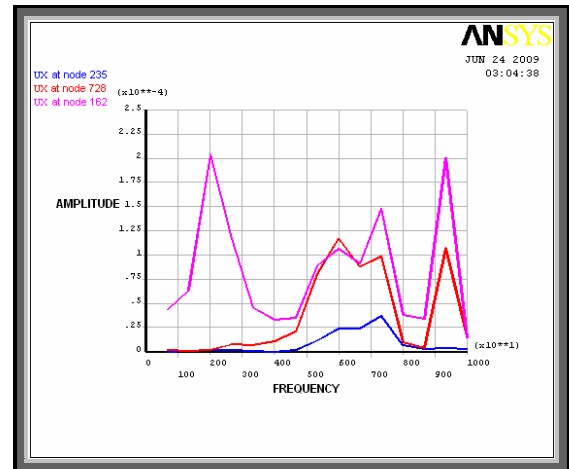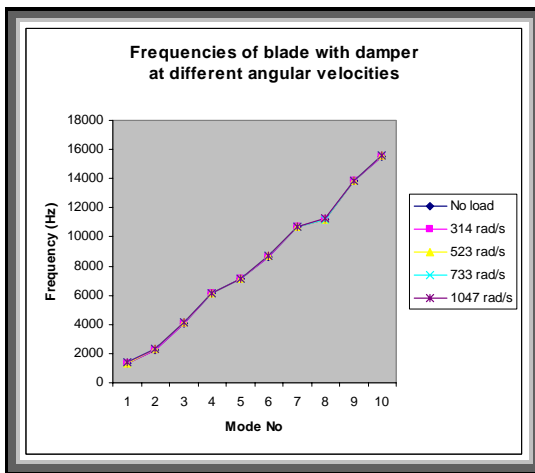

Fig. 25: Frequency response Vs amplitude Ux
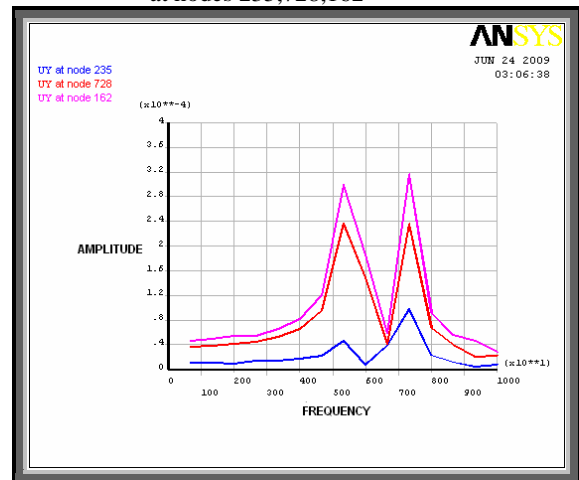at nodes 235,728,162

Fig. 26: Frequency response Vs amplitude Uy
at nodes 235,728,162

323

8   Conclusions

The performance of cottage roof damper affecting the vibrations of turbine blade is investigated in the present paper at different loading conditions, including no load. The results obtained are analyzed for assessing the efficacy of the elements in yielding the natural frequencies, mode shapes and the response of the vibrating blades. It is seen very clearly that the vibrational frequencies are increased. The vibrational displacements decrease substantially when there is locking of the damper with the blade i.e., when there is no gap between the damper and the blade plate. Since the finite element analysis yields results with in 2 to 6 % error and since the rotating blade instrumentation is very expensive, designers rely on FE analysis results.

Acknowledgment

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