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Analysis of Non-Linear Undamped Dynamic Vibration Absorber

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Abstract: **Most real-world phenomena show non-linear behavior. Given the linear behavior of a physical system, there are many situations in which satisfactory results can be obtained. On the other hand, there are situations and phenomena that require non-linear solutions. Non-linear structural behavior can be caused by geometric and material non-linearity, as well as changes in boundary conditions and structural integrity. Non-linear springs have nonlinear relationship between displacement and spring force. The plot of the spring force and displacement of non-linear spring is more complicated than straight line with varying gradients. The non-linearity of the springs is due to thelarge deflection which is the geometric non linearity considering all the above facts dealing with vibration analysis.**

Keywords* — *Absorber, DVA, Spring, Stiffness, Undamped.

I. INTRODUCTION

A spring is an elastic mechanical element / structural component because it deforms significantly under load. Their flexibility allows them to store recoverable mechanical energy. In vehicle suspension, when a wheel encounters an obstacle, the suspension causes the wheel to move over the obstacle and return it to its normal position. Compression springs are cylindrical, conical, tapered, concave or convex in shape and are usually spirally wound from a round wire. The maximum operating length of the spring should be significantly shorter than the free length so that no impact is generated when the contact between the spring and the structural member is lost and the contact is re-established. As the spring approaches strength, the pitch difference between the coils is small, so the contact between the coils progresses rather than suddenly contacting all the coils at once. Contact causes impact, surface deterioration, and increased rigidity.

Spring performance is characterized by the relationship between the force applied to the spring (P) and the resulting deflection (δ). Here, the deflection of the compression spring is taken into account from the unloaded free length. If the spring is tightly wound and the material is elastic, the P-δ properties are nearly linear. The slope of the characteristic curve known as spring stiffness k= P/δ. The design of the spring is very important the comparison is made by changing the wire diameter of the coil spring and mean coil diameter to see the optimum dimensionsof the spring.

II. LITERATURE

Sreenivasulu [1] Demonstrates a methodology for designing non-circular spiral prism springs and circular spiral non- prism springs using analytical and numerical techniques using CAD and FEM. Therefore, in this work, we used CATIA V5R19 to explore the modeling of coil springs with circular, oval, and square cross sections of various materials. Arvind et al. [2] The behavior of conical compression springs has a linear phase, but it can also have a non-linear phase. The

linear phase factor is easy to calculate, but there is no analytical model to accurately describe the nonlinear phase. This non-linear phase can only be determined by a discretization algorithm. They presented an analytical continuous length equation as a function of load for a conical compression spring with a constant pitch in a non-linear phase and vice versa. Verification of the new conical spring model compared to the experimental data is performed. It is now possible to analytically determine the law of behavior of conical compression springs. Valder Steffen et al. [3] studied a nonlinear vibration absorber using the saturation phenomenon, conducted theoretical research on improving its stability and effective frequency bandwidth, and realized a sophisticated nonlinear vibration absorber. Abdullah et al. [4] Illustrated the optimization criteria other than the traditional approach and obtained a slight improvement in steady-state response by using non-linear springs. The system is then analyzed analytically using the harmonic balancing method and compared to the tuned linear DVA. Hioyuki Kato et al [5] They explain that"nonlinear springs are a class of compliant mechanisms with a defined nonlinear load-displacement function measured at some point on the mechanism." "Curved areas of the shell

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connected by creases or hinge lines", or "discrete corrugated structures that can undergo large, reversible displacements" elsewhere in the publication. Vladmir et al. [6] They provide detailed derivations of equations that explain the elastic properties of spiral springs in large deformations. The Taylor expansion of the restoring force has been shown to include a second-order strain term in addition to the linear term. It also explains the physical meaning of this concept. A surprising example of this is that if the initial deflection value is large during spring compression, there may be strain interval intervals where the stiffness difference is negative.Ligia et al. [7] Using the Finite Element Method (FEM), show an analysis of the stresses and deformations that occurduring contact between two objects under compressive loads. The result is H. It is compared with the results obtained by calculation according to Hertz's theory. The analyzed situation is that two cylinders with parallel axes touch each other and both have the same radius R. Dumitru et al. [8] Illustrate the geometric and functional properties of variable wire diameter coil springs. Spring characteristics are determined by two factors. On the one hand, the minimum distance between two adjacent windings reduces with an unloaded spring as the diameter of the wire increases. On the other hand, the deflection of the coil decreases as the wire diameter increases. Therefore, the actual spacing between adjacent coils represents the difference between the two quantities. As a result, it is said that the adjacent coils come into continuous contact with each other according to the actual distance between the coils, and the rigidity of the spring increases. Muhammad et al. [9] Studied those dynamic measurements were performed to investigate the proposed dynamic behavior of the NDVA for each of the horizontal and vertical gap configurations performed in the quasi-static measurements. Feng et al. [10] Studied that determining all resonant frequencies, reapply the GA method to find new optimal positions for all DVAs. In this study, four major modes wereselected as control goals. You can see that in each mode, thethree DVAs can provide the desired bandwidth control performance. Note that the optimization only considers the location of the DVA.

III. MODELLING AND SIMULATION

A. MODELLING OF SPRING:

CATIA is the world's leading design and engineering software for superior 3D CAD product design. Used to design, simulate, analyze, and manufacture products in various industries such as aerospace, automobiles, consumer goods, and industrial machinery. CATIA stands for Computer Aided 3D Interactive Application. This goes far beyond the CAD (Computer Aided Design) software package. I used CATIA v5 R21 software to create geometry for analysis. Selected spring is plain ground end type.

Table 1: Dimensions of springs(mm)

parameter	cylindrical	conical	Taperedwire
Wire Dia(D)	15	15	10 to 25
Coil Dia(D)	90	40 to 130	90
No Of Turns(N)	10	10	10
Pitch	25	25	25
Free Length	225	225	225

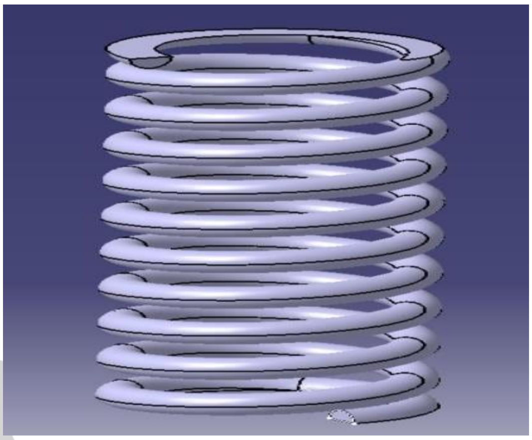

Fig. 1: Cylindrical spring

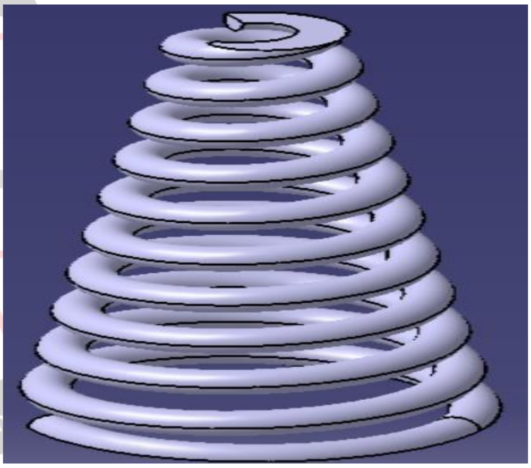

Fig. 2: conical spring

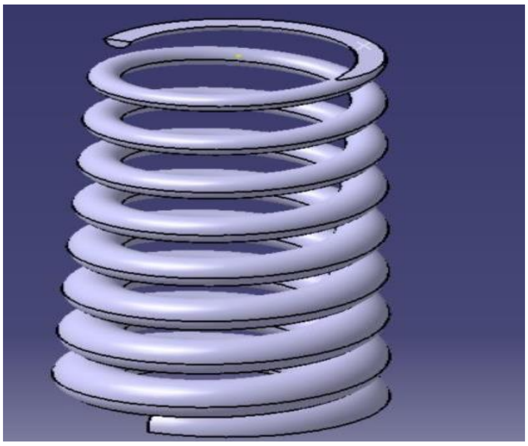

Fig. 3: Tapered wire spring

B. SIMULATION OF SPRING:

The Ansys Workbench platform allows you to integratedata

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from engineering simulations to create more accurate models of the more efficiently. The Analyze Ansys is used to determine how products with different specifications work without building test products or running crash tests. In this we perform the static structural analysis of spring to find out load vs displacement curve to find out the which springs havingmore slope i.e., stiffness of the spring.

Steps involved in the analysis:

I. Assigning the material and its properties.
II. Importing the geometry and update it.
III. Check connections and defining the contact.
IV. Generate the mesh.
V. Apply the boundary conditions
VI. Evaluate the results.

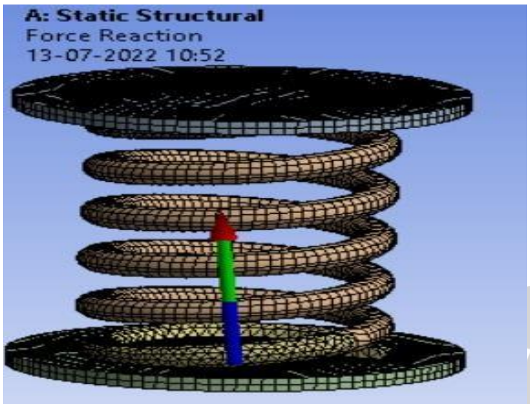

Fig 4.: Force rection of cylindrical spring

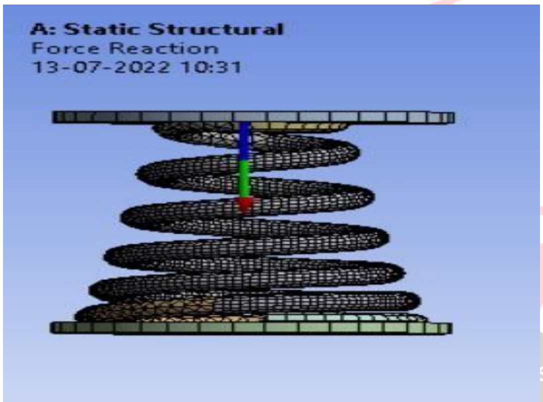

Fig. 5: Force reaction of conical spring

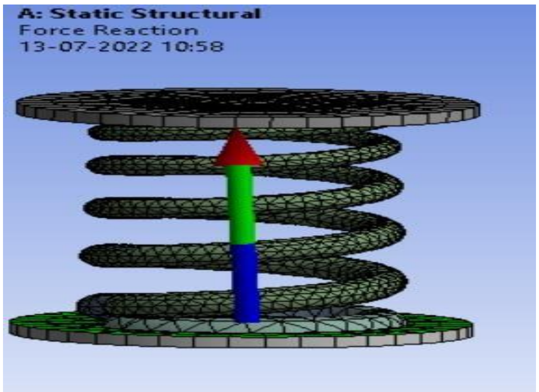

Fig. 6: Force reaction of tapper wired spring

IV. MANUFACTURING AND TESTING

Manufacturing of the spring is done by making a mandrel of the respective size and wrapping the wire on it with the required helix. After manufacturing, we need to test the spring stiffness behavior of the respective geometry of the spring with the help of a universal testing machine (UTM).

Fig. 7: Cylindrical Spring

Fig. 8 Conical Spring

Fig. 9: Tapered Wire Spring

V. DYNAMIC VIBRATION ABSORBER

The sinusoidal force F0sinwt acts on the undamped main mass spring system (if the absorber mass is not installed). If the forced frequency is equal to the natural frequency of the principal mass, the response is infinite. This is called resonance and can cause serious problems with the vibration system. Absorption mass When a spring system is attached to the main mass and the damper resonance is adjusted to the main mass resonance, the movement of the main mass is reduced to zero at that resonance frequency. Therefore, the energy of the main mass is clearly "absorbed" by the tuned dynamic absorber.

Note that the absorber movement is finite at this resonant frequency, even if neither oscillator has attenuation. This is because the system has been changed from a one-degree-of-freedom system to a two- degree-of-freedom system, with two resonant frequencies, neither of which corresponds to the original resonant frequency of the main mass (and absorber). In the absence of attenuation, the response of the 2-DOF system will be infinite at these new frequencies. This is not a problem if the machine is operating at its

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natural frequency, but infinite responses can cause boot and shutdown issues. The finite amount of attenuation of both masses prevents the motion of either mass from becoming infinite at any of the new resonant frequencies. However, if one of the mass spring elements has damping, the response of the principal mass will not be zero at the target frequency.

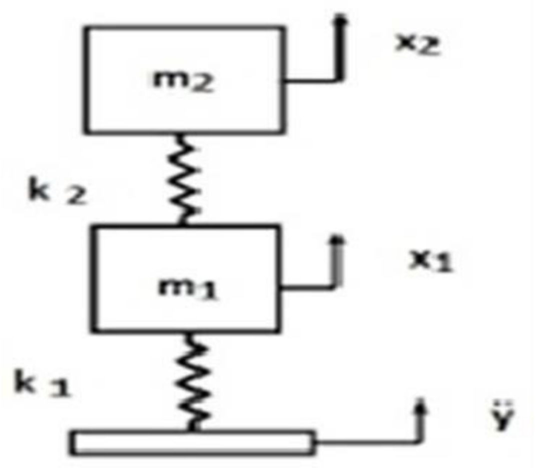

Fig. 10: Dynamic Vibration Absorber

A. Modal Analysis:

Modal analysis is the process of determining system-specific dynamic characteristics in terms of natural frequency, damping coefficient, and modal shape, and formulating a mathematical model of that dynamic behavior. The formalized mathematical model is called the modal model of the system, and the property information is called its modal data.

By selecting the value of mass and stiffness to find out the natural frequency. mass(m_1) =7.5kg Stiffness(K_1) =27910 N/m, m2=0.5kg, k2=1860 N/m

primary system:

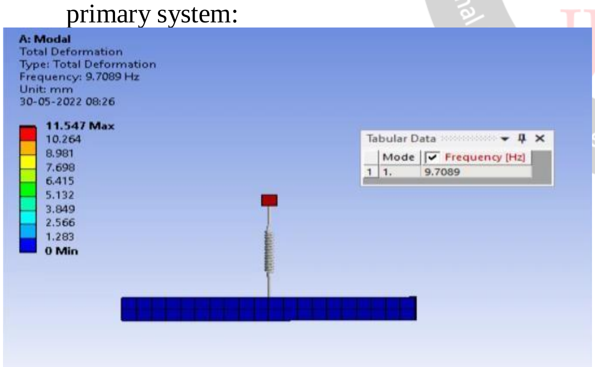

Fig. 11: amplitude of primary mass without DVA

a) With DVA:

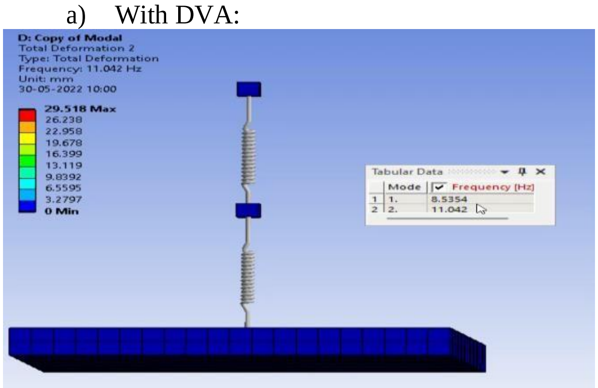

Fig. 12: mode shape of primary mass with DVA

B. Harmonic Analysis:

Harmonic analysis is an alternative method for simulating the response of a structure to forced vibration. In contrast to transient dynamic analysis, harmonic analysis solves structural dynamics in the frequency domain rather than in the time domain.so by providing ramped frequencywithin defined range and find out the resonance amplitude of the mass.

- Input ramped frequency: 0 to 15 Hz

a. primary system*:*

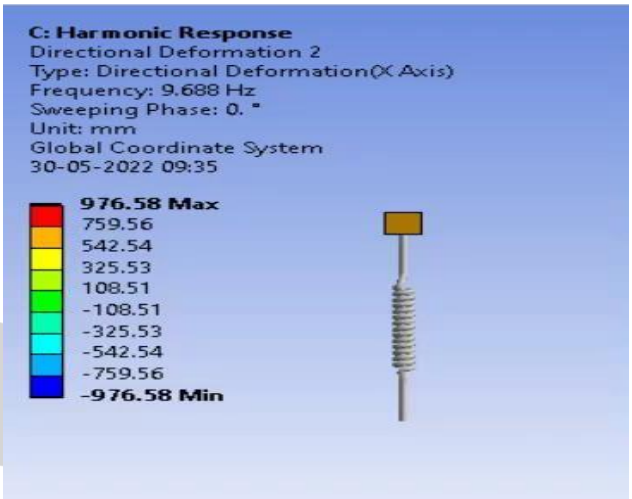

Fig. 13: Amplitude of primary mass at 9.688 Hz without DVA

Natural frequency = $\omega_n=\sqrt{\frac{K}{M}}$ = 61 rad/sec

F= $\omega_n/2\pi$ =9.708 Hz Amplitude(x)=11.547 mm

b. With DVA

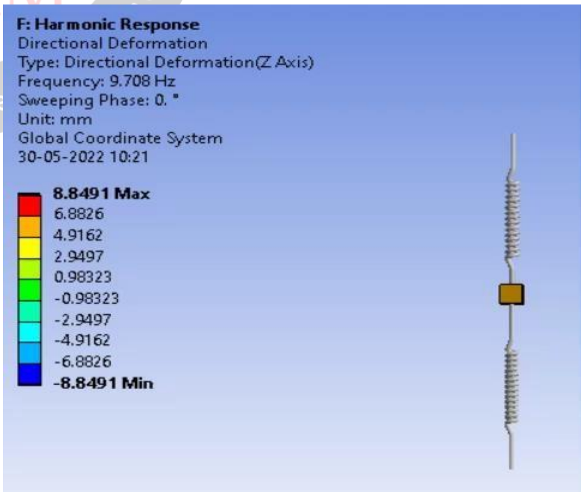

Fig. 14: Amplitude of primary mass at 9.708 Hz With DVA

C. Transient Vibration Analysis:

To find out the directional amplitude of mass element by providing sinusoidal displacement at resonance frequency. The behavior of the system when the time is very long is called a "steady state" response. As you can see, it does not depend on the initial position and velocity of the mass. The

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behavior of the system as it approaches steady state is called "transient response."

Input base excitation=20sin ωt

$\omega_n = \omega$ at this condition resonance occur. So,

y=20sin(61*t) input

a) Primary system

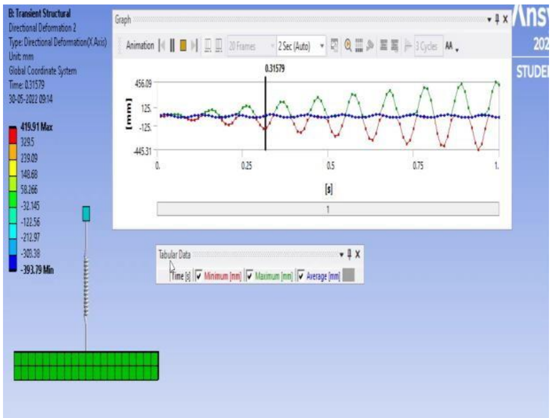

Fig. 15: Amplitude of primary mass without DVA

b) With DVA

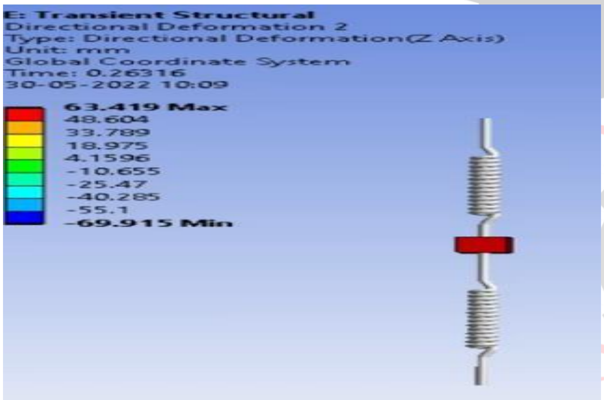

Fig. 16: Amplitude of primary mass with DVA

VI. RESULT AND DISCUSSION

A) As per the testing and simulation result of the various spring geometries Preliminary research has revealed that extensive research has been conducted on the design of springs with circular cross sections, especially to determine stiffness under different load conditions. This study focused on determining the stiffness of helical compression springs of various geometries like cylindrical conical and tapered wire diameter spring within the limiting or equivalent applications. using theoretical techniques and finite element analysis. FEMs using ANSYS-workbench based on geometric non- linearity and from that obtain the load vs displacement graph as shown below:

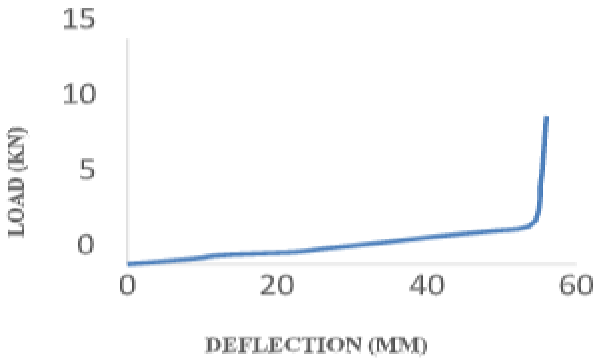

Fig. 17: Load-deflection graph (cylindrical spring)

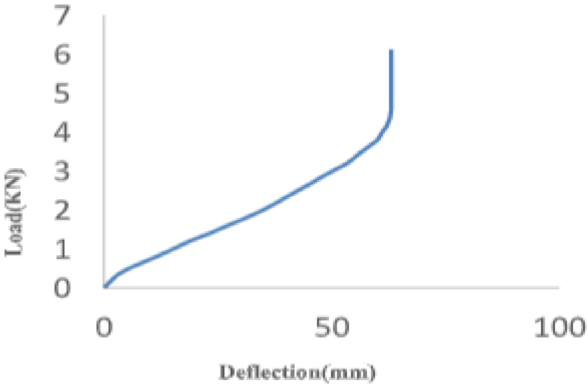

Fig. 18: Load- deflection graph (conical spring)

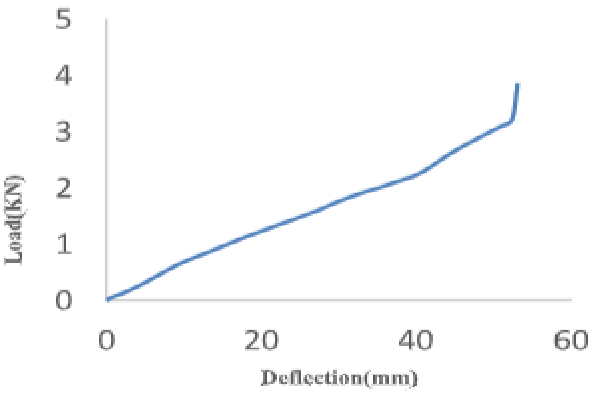

Fig. 19: Load- deflection graph (Taper wire spring)

B) By performing the analysis of primary system and finding out the amplitude of primary mass by using different module of vibrations and after that introducinga secondary system and then observed the amplitude ofprimary mass at resonance frequency.

a) In the Transient vibration analysis, the amplitude of primary mass is 456.09mm at resonance frequency and after introducing the DVA the amplitude becomes 63.41mm.

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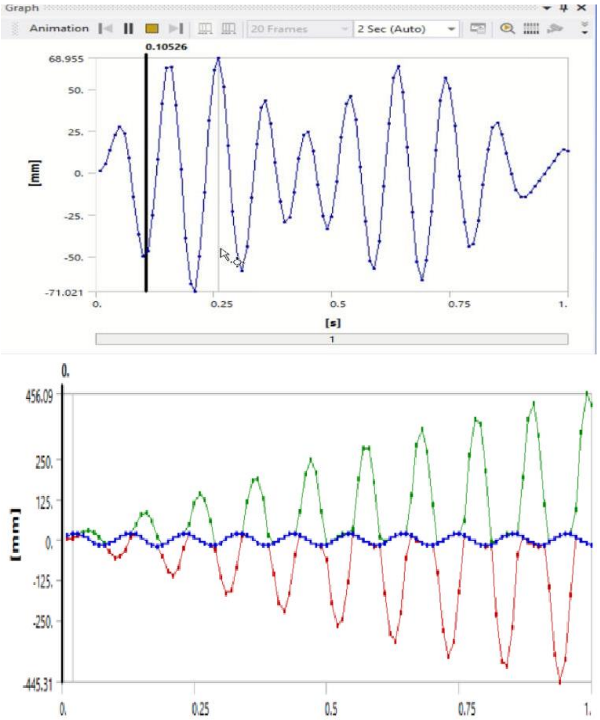

Fig. 20: amplitude of main system without and with DVA

c) In the harmonic analysis by giving the input as a ramped frequency and finding the amplitude of primary mass at resonance frequency and this result found out the1000mm and after introducing the DVA the amplitude becomes 8.849mm at the resonance frequency.

VII. CONCLUSION

a) In this study, the simulation and analysis of coil or helical spring which is the main part in the suspension system in modern vehicles were carried out by using CATIA V5R21 and ANSYS 22. In this analysis use different type of springs namely cylindrical, conical andtapered wire diameter spring to determine the load vs displacement relationship among them and finding the stiffness non linearity with the dimensions of the spring for equivalent dimensions. From that conical springhaving more slope than other two, so from that conical spring have stiffer than other two.

b) Future scope of this work is to change geometrical parameter and find the relationship of load and displacement.

c) In vibrational study various module is carried out to check the amplitude at various input and introducing the DVA to reduce the amplitude of the primary system.

REFERENCES

[1] Reddy Sreenivasulu, N. Yaswanth Krishna, M. Sukumar, O.N. Gouse Basha, N. Arun Kumar, K. Heamanth and M. Vamsi Krishna, "Modeling and Analysis of Helical Springs Using CATIA-V5R19 and ANSYS 16.0", *Akgec International Journal of Technology* (2019) pp41-50.

[2] Arvind Katyayn, Chitta Ranjan Tripathy "Design and Analysis of Helical Coil Spring to Improve Its Strength Through Geometrical Modification" *International Journal of Technical Research & Science* (2020) pp20-25.

[3] Romes Antonio Borges, Valder Steffen, Romes Antonio Borges, "Optimization of a Nonlinear Dynamic Vibration Absorber" *Dynamic vibration absorber* (2007) pp1-10

[4] Abdullah S. Alsuwaiyan, "Design and Analysis of a Simple Nonlinear Vibration Absorber" *IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)* (2014) pp84-90

[5] Hioyuki Kato, Hitoshi Suzuki, "Nonlinear deflection analysis of helical spring in elastic-plastic material: application to plastic extension of piano wire spring", *Mechanics of Material* vol 160(2021).

[6] Vladmir v. ivchenko, "Theoretical Study of Geometrical Non-Linearity of the Elastic Properties of Spring", *American Journal of Physics* 88(2020).

[7] Ligia cristina brezeanu, "contact stresses analysis by finiteelement method", *procedia technology*12(2014).

[8] Dumitru Pop, Simon Haragas, Dorian Popa. "Helical spring with variable wire diameter", *Applied Mechanics* Vol. 62(2017).

[9] Muhammad Harith Mustafer, Roszaidi Ramlan, Mohd Nazim Abdul Rahman, Azma Putra, "Experimental characterization and performance of dynamic vibration absorber with tunable piecewise-linear stiffness", *Journal of the Brazilian Society of Mechanical Sciences and Engineering* (2020).

[10] Feng Qian, Lei Zuo, "Tuned nonlinear spring inerter- damper vibration absorber for beam vibration reduction based on the exact nonlinear dynamics model", *Journal of Sound andVibration* (2021).

 DOI : 10.35291/2454-9150.2022.0379