ISSN No. 2395-0900
International Journal of Engineering Studies and Technical Approach

Design and Analysis of a Passive Damper to Reduce Vibration Effects in Machining Operation

Tojo Thomas*, Dr. S.B.Tiwari & P.Bhasuresan*****

***P.G Scholar, Department of Mechanical Engineering, Sree Buddha College of Engineering, Alappuzha, India**

****Scientist / Engineer SF, VSSC, Thiruvananthapuram, India**

*****Professor, Department of Mechanical Engineering, Sree Buddha College of Engineering, Alappuzha, India**

ABSTRACT :

Aeronautical and space components require better accuracy and surface finish for stringent and strategic applications. Tool chatter is an unwanted phenomenon during machining operation. If not controlled, it can lead to poor surface finish and hence inferior performance of the components. The objective of the present study is to design a passive damper for boring operation of a convergent-divergent nozzle and validation of design through numerical finite element method. Secondary mass is attached to the primary using an elastomer. Apart from this viscous fluid is also used to reduce the vibration of the boring bar. The MATLAB code is developed to solve the finite element mathematical model. The response of the tool with and without damper is compared. The design is validated through Finite Element Analysis (FEA). The frequencies of the primary mass is compared with the analytical solution and the solution obtained from the own developed MATLAB code. Response study and the finite element results are compared and the results obtained are matching well.

Keywords: *Finite Element Analysis (FEA),Parameter optimization, Passive damper, Tool chatter*

INTRODUCTION

Many machines and processes in engineering generate vibration. In a few cases this vibration is intentional as in vibrating sorting screens, ultrasonic cleaners and earth compaction machines. The vibration generated can cause a number of effects that are troublesome. The most serious are related to fatigue and injury to humans. Vibration may also contribute to excessive wear, fatigue failure and other premature failure of machine components. Many vibration problems are due to inadequate engineering design of a product, or the use of a machine in a manner that has not considered the possible effect of vibration.

Vibration in metal cutting is familiar to every machine tool operator. This phenomenon is recognized in operations such as internal turning, threading, grooving, milling, boring and drilling [1], to which there are several reasons why this problem occurs. Some are related to the machine tool itself, the clamping of the tool, the length and diameter of the tool holder and the cutting data to be used. Reducing the process parameters is one such consideration; however, this could have a negative effect on productivity. In order to improve stability against chatter vibrations, various types of vibration absorbers such as dynamic absorber, impact damper, etc. have been developed [2]. Among them, dynamic absorber type dampers which are installed in boring bars are practically used. The theoretical backgrounds and design procedures of boring bars with dynamic absorbers are based on the general dynamic vibration absorbers. However, even today, reliable means of solving the problems of instability in deep boring operations do not exist. Hence, it is generally considered that improvement of machining efficiency and machining accuracy in deep hole boring by boring bar tools is very difficult.

The stability behavior of a slender boring bar is studied and boring bar was modeled as a two degree of freedom mass-spring-damper system. The analysis of boring bar vibrations is usually based on the lower order bending modes of the clamped boring bar. This study is focused on the stability of a boring bar with a passive dynamic absorber [3]. The boring bar was modeled as a cantilever Euler–Bernoulli beam and only its first mode of vibration was taken into account [4]. Vibration absorbers are devices attached to flexible structures in order to minimize the vibration amplitudes at a specified set of points. Design of vibration absorbers has a long history. First vibration absorber proposed by Frahm in 1909 consists of a second mass-spring device attached to the main device, also modeled as a mass spring system [5], which prevents it from vibrating at the frequency of the sinusoidal forcing acting on the main device.

The objective of the present study is to propose a new modification for the boring tool and design a damper to reduce the response of boring tool in order to improve the surface finish of the job. Optimum design is arrived by analytical way by varying the design parameters. Design is validated through the structural analysis of the designed configurations using commercially available finite element software ANSYS.

MODELING OF BORING BAR

Finite element method is a numerical method for solving problems of Engineering and Mathematical Physics. In this method, a body or a structure in which the analysis to be carried out is subdivided into smaller elements of finite dimensions called finite elements. Then the body is considered as an assemblage of these elements connected at a finite number of joints called 'Nodes' or Nodal points. The properties of each type of finite element is obtained and assembled together and solved as whole to get solution.

In the present study the stiffness, damping and mass of the boring bar (primary) is simulated using a beam element. The dynamic absorber consists of heavy mass (secondary) supported at the end of an elastomer and it is surrounded by a viscous fluid. The stiffness and the damping of the DVA are simulated using as a spring and damper element. One end of the spring and damper element is connected to the primary beam element. The heavy mass of the DVA is modeled as a rigid beam element.

Define Shape Function Matrix

The displacements and slopes at the two nodes 1 and 2 of an element v_1, ϕ_1, v_2 and ϕ_2 are assumed to be known. At any location within the element they are obtained by using appropriate shape function or interpolation function. Since we have four known quantities, i.e., the element has four degrees of freedom, we can determine four constants in the shape function. Therefore, we assume the transverse displacement v (x) to be a cubic polynomial within the element to be

$$v(x) = a_1 x^3 + a_2 x^2 + a_3 x + a_4 \quad (2.1)$$

The above equation (2.1) satisfies the governing differential equation of a beam, viz. $EI\frac{d4y}{dx4}$ =0. In addition, we also note that the cubic displacement shape function satisfies the continuity condition of both the deflection and slope at the nodes. We now express the transverse displacement of the element as a function of the nodal degrees of freedom v_1, ϕ_1, v_2 and ϕ_2. With the help of the boundary conditions of the element at the two nodes, we have

$$v(0) = v_1 = a_4, \frac{dv\,(0)}{dx} = \phi_1 = a_3,$$

$$v(L) = v_2 = a_1L^3 + a_2L^2 + a_3L + a_4,$$

$$\frac{dv\,(L)}{dx} = \phi_2 = 3a_1L^2 + 2a_2L + a_3 \qquad (2.2)$$

Solving for the four constants a and substituting in (2.1), we get

$$v(x) = \left[\frac{2}{L3}(v1 - v2) + \frac{1}{L2}(\phi 1 + \phi 2)\right]x^3 - \left[\frac{3}{L2}(v1 - v2) + \frac{1}{L}(2\phi 1 + \phi 2)\right]x^2 + \phi_1 x + v_1 \qquad (2.3)$$

Collecting terms of the nodal degrees of freedom and writing in matrix form, we get

$$v = [N]\,\{d\} \qquad (2.4)$$

where [N] is shape function.

Equations of Motion using Lagrange's Approach

The equation of motion for the axial motion of a rod and the transverse motion of a beam can be derived based on Newton's second law of motion, but it is convenient to use an energy based method and Lagrange's equations of motion for solving beam problems. If T-represents the kinetic energy of a system and π-represents potential energy, then Lagrange's equations of motion in the independent generalized coordinates u are given by,

$$\frac{\partial\pi}{\partial u} + \frac{d}{dt}\frac{\partial T}{\partial \dot{u}} - \frac{\partial T}{\partial u} = F \qquad (2.5)$$

We know that, shape functions for a typical finite element, the displacement of an interior point can be written in terms of the nodal degree of freedom as,

$$\{u\} = N_1u_1 + N_2u_2 \qquad (2.6)$$

Differentiating with time the velocity at the point is given by,

$$\{\dot{u}\} = N_1\dot{u}1 + N_2\dot{u}2 \qquad (2.7)$$

The kinetic energy of an element mass, m = ρ v within the element is given by,

$$T = \frac{1}{2}mv^2$$

$$dT = \frac{1}{2}\rho\, dv\, (v)^2 \qquad [m = \rho v;\ dm = \rho\, dv]$$

$$dT = \frac{1}{2}\{\dot{u}\}^T\{\dot{u}\}\,\rho\, dv \qquad (2.8)$$

where $\{u\} = [N]\,\{u\}^e$ & $\{\dot{u}\} = [N]\,\{\dot{u}\}^e$ (2.9)

Substitute the equation (2.9) in equation (2.8),

$$dT = \frac{1}{2}\{\dot{u}\}^{e\,T}[N]^T[N]\{\dot{u}\}^e\rho\, dv \qquad (2.10)$$

Integrating the above equation, $T = \frac{1}{2}\{\dot{u}\}^{e\,T}\int_v (\rho\,[N]T[N]dv)\,\{\dot{u}\}^e$

$$T = \frac{1}{2}\{\dot{u}\}^{e\,T}[m]^e\{\dot{u}\}^e \qquad (2.11)$$

The total PE of the system, $\pi_p = \frac{1}{2}\{u\}^T[K]\{u\} - \{u\}^T\{F\}$ (2.12)

Equation (2.11) $\Rightarrow$ $T = \frac{1}{2}\{\dot{u}\}^T[m]\{\dot{u}\}$

$$\Rightarrow \quad \frac{\partial T}{\partial\{u\}} = 0, \qquad \frac{\partial T}{\partial\{\dot{u}\}} = [m]\{\dot{u}\}, \qquad \frac{d}{dt}\left(\frac{\partial T}{\partial\{\dot{u}\}}\right) = [m]\{\ddot{u}\}$$

We know that,

$$\frac{\partial \pi}{\partial\{u\}} = [K]\{u\} - [F]$$

Substituting the above values in Lagrange's equation of motion,(Equation no.(2.5))

$$[m]\{\ddot{u}\} + [K]\{u\} - \{F\} = 0 \qquad (2.13)$$

Where $[m]^e = \int_v \rho[N]T[N]\,dv$ is the consistent mass matrix for the element

$[K] = \int_v B^T DB\,dv$ is the stiffness matrix for the element.

Modal Damping Matrix

The damping of structure is assumed to be viscous and frequency dependent for the simplicity in analysis. Damping ratios can be calculated using the Rayleigh damping method and it is known as proportional damping or classical damping and it expresses damping as a linear combination of the mass and stiffness matrices. That is,

$$C = \alpha M + \beta K \qquad (2.14)$$

Where M - mass matrix

K - stiffness matrix

α and β are damping coefficients,

$$\alpha = \frac{2\,\omega i\;\omega j}{\omega i + \omega j} \quad \text{and} \quad \beta = \frac{2}{\omega i + \omega j} \qquad (2.15)$$

ωi and ωj are frequencies of initial and higher modes.

System Matrices

We can form system matrix by combining primary system and secondary system. We must note two points when we combine primary system with secondary system. For global system matrix secondary mass doesn't contribute cross coupled elements whereas in stiffness and damping matrices will have direct and cross coupled components showing the nodal connectivity between primary and secondary.

The off-diagonal terms of the inertia matrix are zero, while the off-diagonal terms of the stiffness and damping matrices are non-zero. In addition, all of these matrices are symmetric matrices. The equations governing the system are coupled due to these non-zero off-diagonal terms in the stiffness and damping matrices.

The symmetric matrices represent the corresponding governing equations for each element and we got a set of governing equations equal to that of degree of freedom of the system. Solve the matrix for displacement x and we get the corresponding amplitude of vibration.

DESIGN PROCEDURES

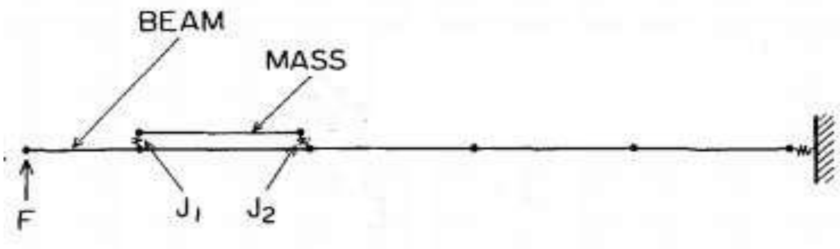

Figure.1: Finite Element Model of the Boring Bar

Figure shows models for structural analysis of the boring bar. The joint part clamping the boring bar is constrained in all six degrees of freedom. Solid circles in figure indicate nodal points for structural analysis. Design variables for this problem are the outside diameter D_o and the inside diameter D_i of the boring bar, the length L of the additional mass elements, and spring stiffness K and viscous damping coefficients C which simulate rubber elements at both ends of additional mass.

The design variables are determined using mathematical models shown in figure and a mathematical programming method for searching optimum design variables, according to the following steps.

- **Length and diameter of the tool:** The overhanging length L and the outside diameter D_o of a boring bar are determined depending on practical requirements. Therefore a boring tool having diameter less than the minimum bore diameter and length greater than boring depth is required. Then only the required geometry can be made using the boring bar.
- **Material selection:** The commercially available conventional boring bars are made up of tool steel. This paper deals with the effect of a boring bar with and without damper. That is the effect of vibration between a conventional boring bar and a damped boring bar. For that same tool materials are selected for the study.
- **Position of the secondary system:** The length of primary is determined by considering the boring depth. The single point cutting insert is attached at the free end. The dynamic force component is acting on the insert. This force leads to the vibration of boring tool. As we study the theory of dynamic vibration absorbers, it is clear that the absorber mass is attached to the point at which the vibration amplitude is maximum.
- **Determination of secondary system parameters:** After the selection of position of the secondary mass, the next step is to find out the absorber mass. In theory secondary mass should be as large as possible. But constraints due to the availability of space limit the secondary mass. In practice, typically it is selected as 25% of the modal mass. The mass of the absorber depends upon the dimensions and the density of the secondary material. For that purpose an optimization procedure is carried out by using MATLAB program.
- **Setting of clearance:** Clearance is nothing but it is the annular gap between primary and secondary and the gap is filled by the viscous fluid. The clearance should be large

enough to accommodate secondary response in order to reduce the response of the primary.

- **Selection of fluid:** The selections of damper elements are based on optimization process. The viscous fluid only gives the damping effect and the elastomer provides both stiffness and damping property. We have the clearance value and working eccentricity, thus using the squeeze film damper relationships for a given damping we can find out the viscosity of fluid.

Damping, $C = \frac{\mu R L^3 (2\Sigma^2 + 1)}{c^3 (1 - \Sigma^2)^{5/2}}$

(3.1)

Where R is the damper radius, L is the damper axial length, *c* is the radial clearance, μ is the oil viscosity, ∑ is the eccentricity ratio

DEVELOPMENT OF MATLAB PROGRAM

To design the suitable vibration parameters the mathematical modeling of the boring bar is made using the MATLAB program. The code is developed for both boring bar with and without damper. The bar is considered as a beam element and six degrees of freedom are taken into account for every node in the bar. The boring bar is considered as a cantilever beam and one end of the bar is fixed. The flow charts for the preparation of MATLAB program are as follows.

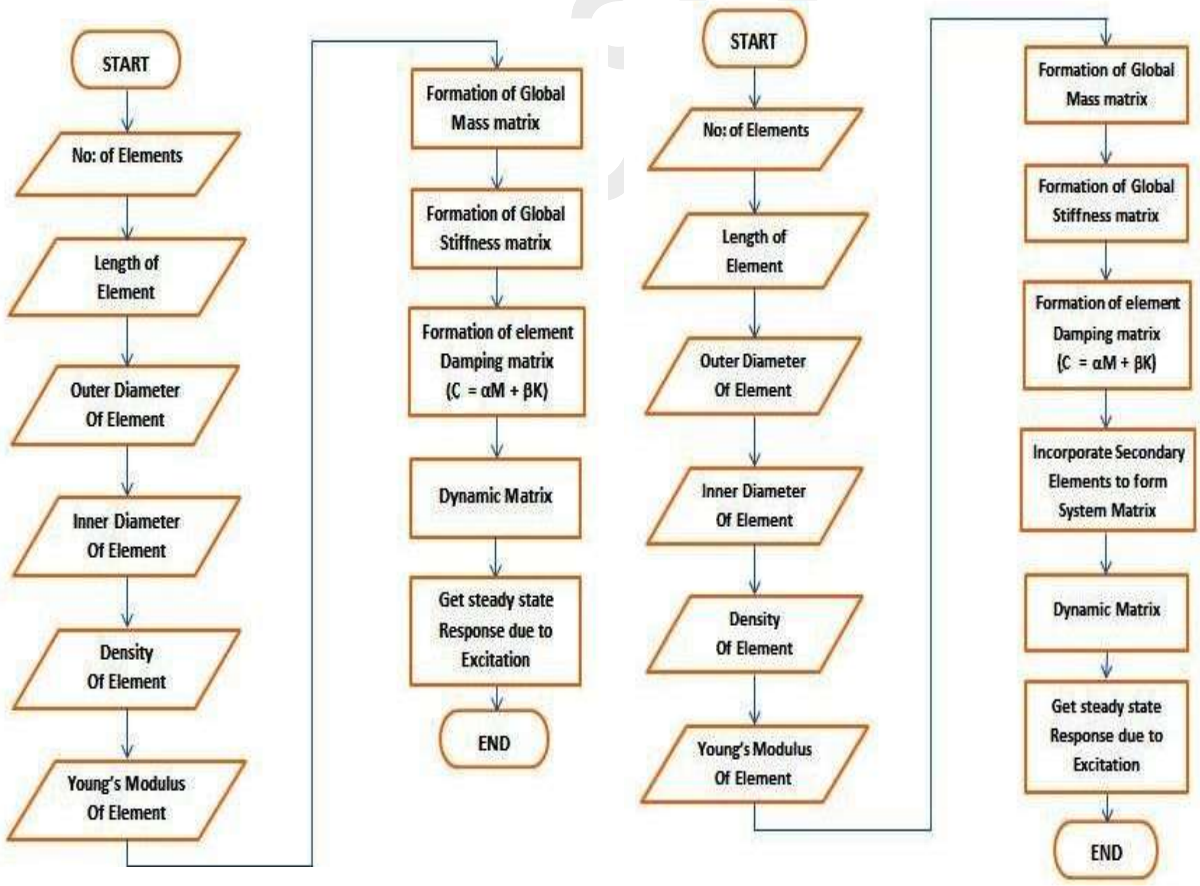

Figure.2: Flow Chart for MATLAB Program without Damper and with Damper

Damper Design through Optimization of Vibration Parameters

The effectiveness of damper device primarily depends on three parameters, modal mass, stiffness and damping ratio. An iterative study was carried out to finding out the optimum parameters. The cantilever beam with and without secondary system is appropriately modeled using finite element methodology and analyzed for the harmonic response. By varying the input parameters like secondary mass, stiffness and damping values the optimum design parameters can be identified. There is a lot of combinations are iterated for obtaining the suitable combination of parameters. The close values nearer to the optimum design are shown below.

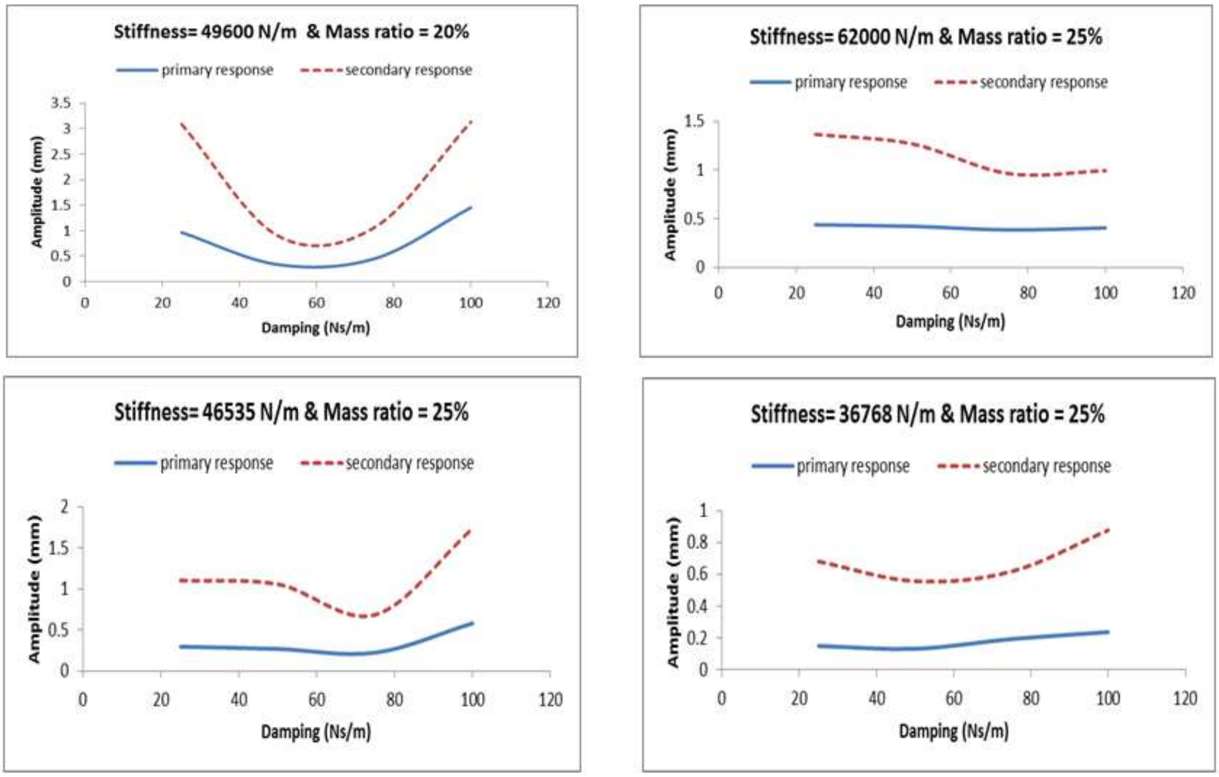

Figure.3: Responses for Different Combinations

VALIDATION OF DESIGN THROUGH ANALYSIS

T

he geometrical details of the primary mass are given in Table.1

Table.1: Primary Geometry Descriptions

TITLE	DESCRIPTION
Length	0.308 m
Volume	1.17e-4 m^3
Mass	0.919 kg

Primary mass is made of steel. The material property of the primary mass considered for the analysis is given in Table.2

ISSN No. 2395-0900
IJESTA International Journal of Engineering Studies and Technical Approach

Table.2: Material Properties of Primary Mass

TITLE	DESCRIPTION
Young's Modulus	2.1e11 Pa
Poisson's ratio	0.3
Density	7850 Kg/m^3

The fundamental frequency of the primary mass are obtained using Finite element methods, Analytical method and own developed MATLAB program and the comparison of the frequencies are given in Table.3

Table.3: Comparison of Frequencies of Primary Mass

Mode	Frequency (Hz) Analytical	Frequency (Hz) MATLAB Program	Frequency (Hz) ANSYS
1	167.8	167.8	167.18
2	1051.62	1051.63	1028.8
3	2944.87	2944.76	2802.1

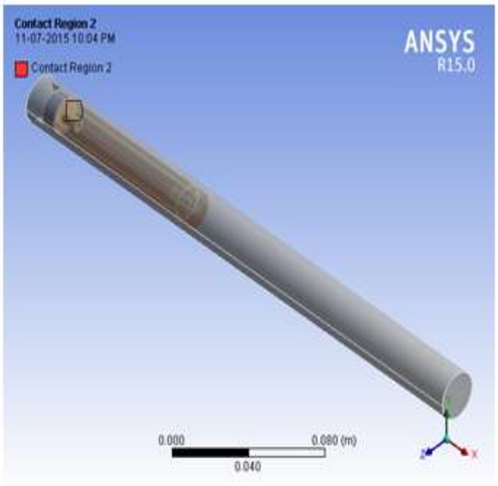

Figure.4: Assembled Geometry

Figure.5: Damped boring bar

RESULTS AND DISCUSSION

A modal analysis of the boring tool with the given dimensions has been conducted by using MATLAB program and ANSYS software. The results have been compared with analytical values. The first mode which is cantilever free end bending mode is coming at 167.8Hz. MATLAB gives the same frequency whereas ANSYS gives 167.18Hz. The second mode which is a one noded cantilever bending is coming at 1051.62Hz as per theory. MATLAB gives a frequency of 1051.63Hz and ANSYS gives 1028.8Hz. The MATLAB results give a close agreement with analytical values.

A steady state response analysis has been conducted for the boring tool with and without damping. With the introduction of Rayleigh damping or material damping the peak response is reduced by 9.92%. An optimization procedure is carried out by changing the mass ratio, frequency ratio and damping ratio. The primary and secondary responses have been plotted

ISSN No. 2395-0900
International Journal of Engineering Studies and Technical Approach

for each variation and an optimum value for mass, stiffness and damping have been arrived. These values have been incorporated in ANSYS model and the response so obtained has been checked with MATLAB results. The response analysis also gave a satisfactory agreement between ANSYS and MATLAB.

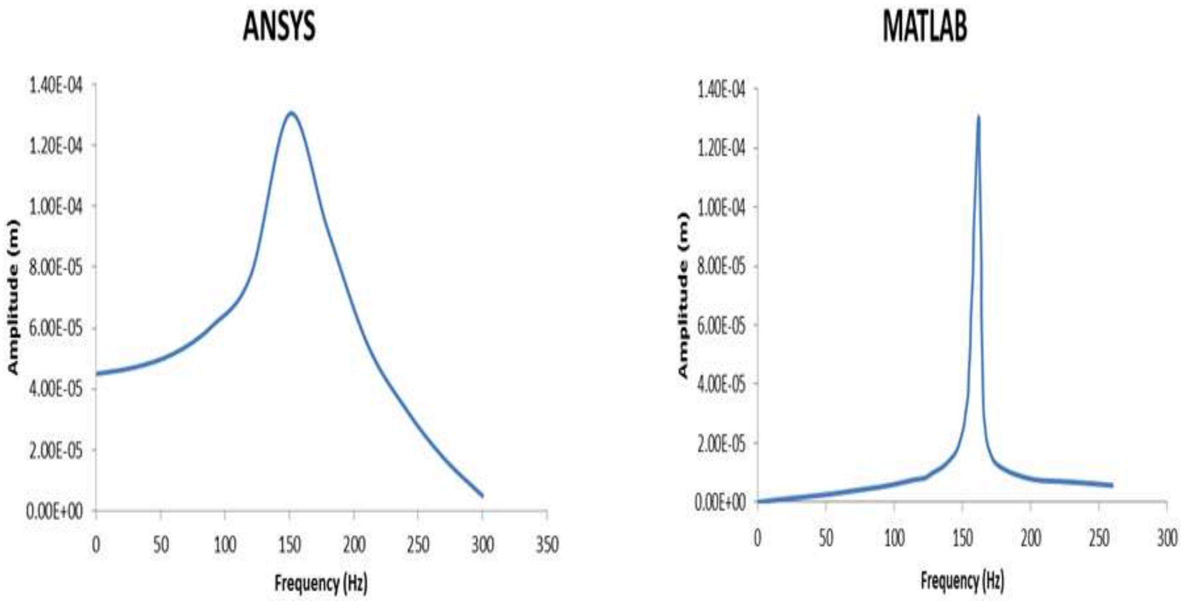

Figure.6: Response with damper obtained by using ANSYS and MATLAB

CONCLUSIONS

Design of passive absorber is carried out for reducing vibration effects in machining operation. Optimum design parameters (mass, stiffness and damping) is arrived using numerical iteration by own developed MATLAB program. Design is validated through commercial FEA software ANSYS. Results obtained using Analytically, MATLAB program and ANSYS is matched well.

The primary response seems to be reducing with the introduction of optimum values of mass, stiffness and damping for secondary system. The reduction in amplitude leads to a better surface finish for the machined product. The surface finish can further be improved by the introduction of additional dampers.

ACKNOWLEDGEMENTS

I humbly take this opportunity to thank God for His grace bestowed upon me to complete the thesis work. I sincerely thank Dr. S.B.Tiwari (Scientist/Engineer SF, VSSC), Mr. K. Narasaiah (Scientist/Engineer SG, VSSC), Mr. SajeevPadmanabhan (Former Scientist, BARC) and Prof. P.Bhasuresan, Professor, Department of Mechanical Engineering who guided me throughout the project and for rendering valuable guidance and instructions that paved the foundations for a successful project. I express my deep gratitude to all my friends and family for their inspiration and sincere cooperation. Last, but not the least, I thank all my well-wishers who have been a constant source of encouragement and help to me.

ISSN No. 2395-0900

REFERENCES

i Neil D. 'Sims, "Vibration absorbers for chatter suppression: A new analytical tuning methodology" Journal of Sound and Vibration, December 2006, doi:10.1016/j.jsv.2006.10.020

ii Lonnie Houck, Tony L. Schmitz, K. Scott Smith. "A tuned holder for increased boring bar dynamic stiffness", Journal of Manufacturing Processes, October 2010, doi:10.1016/j.jmapro.2010.09.002

iii L. Rubio, J.A. Loya, M.H. Miguelez, J. Fernandez-Saez, "Optimization of passive vibration absorbers to reduce chatter in boring", Journal of Mechanical Systems and Signal Processing, August 2013, doi.org/10.1016/j.ymssp.2013.07.019

iv Masataka Yoshimura, "Vibration-proof design of boring bar with multi degree-of-freedom dampers", Design Engineering Technical Conference, December 1986, Vol.108

v M.D.Thomas, "The Impact Damper as a Method of Improving Cantilever Boring Bars", Journal of Engineering for Industry, August 1975, Paper No.74-WA/DE-9

vi Fredric F. Ehrich, *HandBook of Rotor Dynamics* (Krieger publishing company, Third edition)

vii W.T.Thomson & Chandramouli Padmanabhan, *Theory of Vibrations with Applications* (Pearson Education, 2012)