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Stability Analysis of an Optimized Tuned Mass Damper System for Chatter Suppression in Turning Operation

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***Abstract* - This paper deals with the mitigation of unnecessary tool chatter vibrations in turning operations using a passive control device commonly known as a dynamic vibration absorber attached to the tool post of the lathe. In turning operation on a lathe, the development of chatter is mainly due to the effect of regenerative coupling during machining. The lathe machine with the absorber was modeled as a 2DOF system and corresponding differential equations were formulated. Parameter optimization of the absorber is done using the well-known metaheuristic technique called the Genetic Algorithm (GA) by numerically solving corresponding differential equation of motion. Time response and frequency response of system with and without absorber were studied and the chip thickness was also analyzed for different depth of cut under variable spindle speeds was investigated. Stability lode diagram (SLD) was used for analyzing the stable and unstable regions of operations of the system with and without absorber. It was found that for smooth surface finish and constant chip width formation is achievable through the vibration absorber.**

I. INTRODUCTION

In machining (turning) process, vibration is a dynamic volatility of the metal removing process, which can study through the anatomical dynamics and the interface of the dynamics of the metal removing process of the machine tool [1-3]. In turning operation on a lathe machine, vibration/chatter will leads to affect the workpiece surface finish accuracy, reduce the life time of the cutting tool and create annoying intolerable sound. By accounting all of that, there are numerous approaches have been implemented to mitigate the vibration in machining (turning) operation [4-8]. To reduce the vibration of the cutting tool, deliberate the vibration response characteristics of the cutting tool and reduce that chatter by developing an optimized vibration absorber. Fundamentally passive control method is used for reduce the vibration of a system for dynamic tuned vibration absorber (TVA). Even if active control methods have turn out to be more and more well-liked, but passive control methods provide significant and effectual vibration manage tool. Among passive control and active control, passive control shows the advantages of effortless implementation, reduced price, and requirement of peripheral power is not need. Passive control method will never go on to the unstable region. But in active control method there is a chance of instability performance will occur onto the system [9]. Frequency response plot and displacement/response plot for the machine tool on a lathe turning operation is obtained. Observed the behavioral performance of the chatter development, and suppressed the vibration development through software analysis and experimental analysis. Primarily conduct a software analysis onto the system, after that conducted an experimental analysis, by accompanying all parametric condition and available resources, developed a tuned mass damper system for the machine tool to suppress chatter in turning operation on a lathe.

Usually the manufacturing process is completed only through by done machining (metal removing) process. While doing machining operations reducing the surface roughness and increment of rate of removal of material is the main priority**.** Turning operation guarantees good surface finish and high tolerance capacity. Among two types of turning operation, peripheral turning operation induces number of constraints into the working operation. In turning operations, to mitigate the chatter vibration an active control method (piezoelectric inertia actuator) is used, which is attached on the machine tool and it will play a role for tuned dynamic vibration reducer. Stability of the cutting in turning process be able to increases by adjusting the frequency response function of the machining tool, that frequency response function modification is done by the tuned dynamic vibration reducer. If the stability of cutting is improvised then vibration will efficiently reduces [16].

II. EXPERIMENTAL SETUP

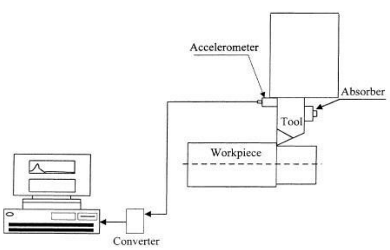

Fig. 2.1 Schematic diagram for experimental setup

Fig. 2.1 is the schematic diagram for experimental setup. It consists of accelerometer, absorber, tool, workpiece, converter and monitor. Tool is attached to the toolpost of the lathe. Only a tolerable depth of cut is provided. During

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machining, the material will remove from workpiece due to the interaction of tool and workpiece. At that time there is a chance of chatter development will occur. To attenuate that chatter, a vibration absorber unit is provided. For measuring that chatter response, a uniaxial direction accelerometer is connected. Using this sensor device, the frequency response will generate in the monitor. Frequency response of the system is analyzed with and without absorber connected. The workpiece rotating at three different speeds ($240rpm$, $310rpm$ and $740rpm$) with the help of a lathe machine.

TABLE I
PARAMETER VALUES

Parameters	Values
Mass of primary system (m)	$3.220\ kg$
Mass of secondary system (m_a)	$0.240\ kg$
Stiffness of spring (k)	$1.45\times10^7 N/m$
Stiffness of spring (k_a)	$4.66\times10^6 N/m$
Damping coefficient (c)	0.01
Damping coefficient (c_a)	0.03
Cutting force ($f\ or\ f_0$)	$10^5 N/m^2$

III. RESULT AND DISCUSSION

Most of the software analysis of this work is done by using MATLAB. Stability analysis of the system is carried out through two different cases, Frequency response system and Stability chart analysis respectively. Frequency response analysis is carried out through four different cases. A single degree of freedom system with linear/non-linear conditions and the system attached to the linear/non-linear tuned mass damper system respectively. SLD is obtained from the governing equation of delayed differential equation from the system. With/without absorber, three modes of experimental analysis is done. Frequency response plot is carried out by using uni-axial accelerometer, SLD by using chip width measurement and feed rate reading respectively. Three speed ($200\ rpm$, $310\ rpm$ and $740\ rpm$) machining lathe is used for conducting this experiment. Some results are not shown in this paper for sake of the simplicity.

3.1. Time Response

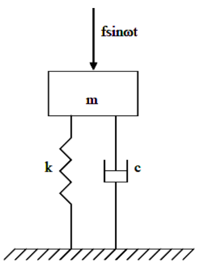

Fig. 3.1 Primary SDOF System

Equation of motion is,

$$m\ddot{x} + c\dot{x} + kx = fs(\omega t) \quad (3.1)$$

Where m, c and k are the mass, damping and stiffness of the system respectively. f is the force acting on the system. x is the displacement of spring along in vertical direction.

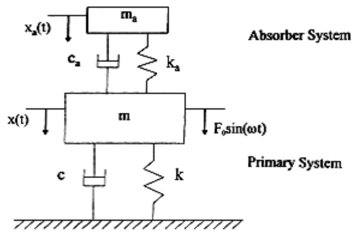

Fig. 3.2 Primary system connected with linear absorber

Governing equation of motion is,

$$m\ddot{x} + k\dot{x} + k_a(x - x_a) + c\dot{x} + c_a(\dot{x} - \dot{x}_a) = f_0 sin(\omega t) \quad (3.2)$$
$$m_a\ddot{x}_a + k_a(x_a - x) + c_a(\dot{x}_a - \dot{x}) = 0 \quad (3.3)$$

Where m, c and k are the mass, damping and stiffness of the primary system respectively. f_0 is the force acting on the system. m_a,c_a and k_a are the mass, damping and linear/nonlinear stiffness of the absorber. x and x_a are the displacement of spring of primary and secondary systems respectively.

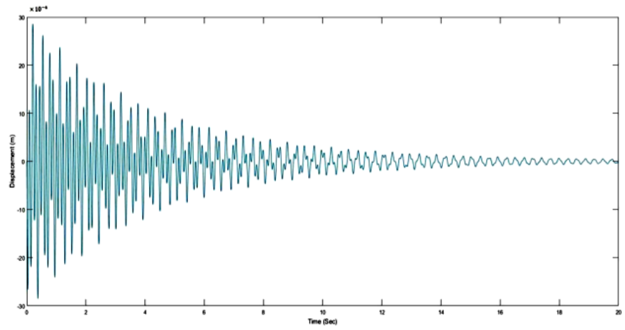

Fig 3.3 Time response plot of system connected with non- linear absorber

System operating under, an absorber with nonlinear stiffness is attached to the primary system, the time response plot of the system is shown in Fig. 3.3.

2. Chatter Stability

Workpiece is rotating in $N\ rpm$ in counter clock wise direction. For metal removal process a tool is fed into the workpiece along in x direction. Where K and k_1 are the damping and stiffness of the workpiece respectively. m, and k are the mass, damping and stiffness of the tool. F_c is the cutting force acting on workpiece. Governing equation of motion is,

$$-(m\ddot{x} + c\dot{x} + kx) = k_1\,(x\,(t) - \mu x\,(t - \tfrac{2\pi}{N})) + K\frac{2\pi}{N}\frac{dx}{dt} \quad (3.4)$$

For non-dimensionalization,

$2\zeta = \frac{C}{\omega_n}$, $Q = \frac{1}{2\zeta}$, $\omega_n^2 = \frac{k}{m}$, $\tau = \omega_n t$

Where ζ is the damping factor and Q is the reciprocal of half of the damping factor.

Therefore,

$$\frac{d^2x}{dt^2} + \left(\frac{1}{Q} + K\frac{K}{k\Omega}\right)\frac{dx}{dt} + \left(1 + \frac{k_1}{k}\right) - \frac{k_1}{k}\mu x\left(\tau - \frac{1}{\Omega}\right) = 0 \quad (3.5)$$

Formation of characteristic equation

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put, $x = Ae^{\lambda\tau}$
Hence,

$$\lambda^2 + \left(\frac{1}{Q} + \frac{K}{k\Omega}\right)\lambda + 1 + \frac{K_1}{K}\left(1 - \mu e^{-\lambda/\Omega}\right) = 0 \qquad (3.6)$$

Put, $\lambda = j\omega$ and then separate it into real and imaginary parts and then equate to 0.
Real,

$$\omega^2 = 1 + \frac{k1}{k}\left(1 - \mu\cos\left(\frac{\omega}{\Omega}\right)\right) \qquad (3.7)$$

Imaginary,

$$\frac{1}{Q} + \frac{K}{k\Omega} + \frac{\mu k_1}{k}\frac{\sin\left(\frac{\omega}{\Omega}\right)}{\omega} = 0 \qquad (3.8)$$

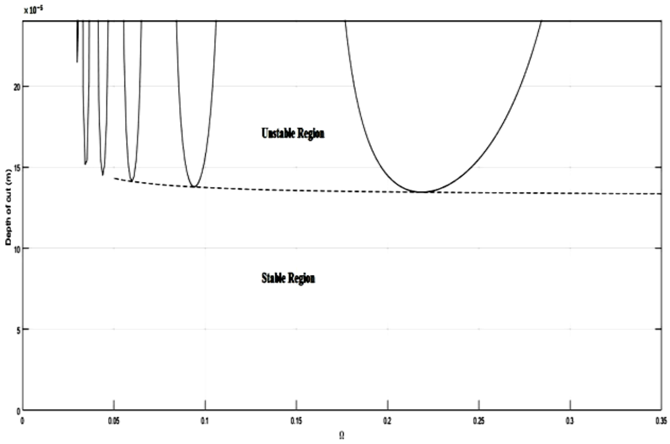

Fig. 3.4 Stability Lobe diagram

3 SLD Experiment with Tuned Mass Damper

A vibration absorber is attached to the tool holding device and then turning operation is done on to the workpiece. In turning operation, regenerative chatter is the main phenomenon for producing surface roughness on the workpiece. At three different speeds, chip width or the axial depth of cut were noted down. At each speed, five sets of different chips were randomly selected and measured the chip thickness. Then plotted against with the spindle speed, shown in Fig 3.5. Chip thickness studies and measurements are done using profile projector (magnification 20:1) and screw gauge. Depth of cut up to 1.49mm, the system is in stable condition, and above that region the system is under unstable condition.

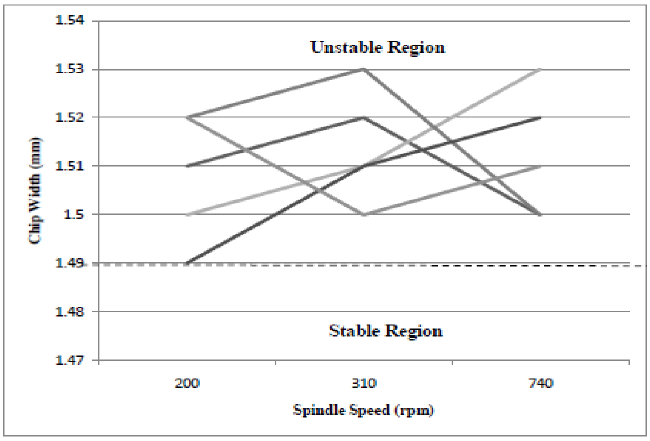

Fig 3.5 Stability lobe diagram with absorber

4. System with Optimized Tuned Mass Damper

Optimization of the vibration absorber is done by using Genetic Algorithm (GA) method for the theoretical modal analysis. Optimization parameter for the vibration absorber is mass. At Spindle speed 740 rpm, the amplitude of vibration is 12μm at the range of frequency of 8100Hz. Fig 3.6 shows that the frequency response of a single DOF system having an optimized absorber is attached to it. The response of the system is 18μm at the range of (8000-9000 Hz). Due to the attachment of optimized vibration absorber, the highest response peak of the system reduces from 35μm to 18μm. The shallow portion between the two peaks is very low. The system will not lead to the resonant condition. Which indicates the system is much more stable.

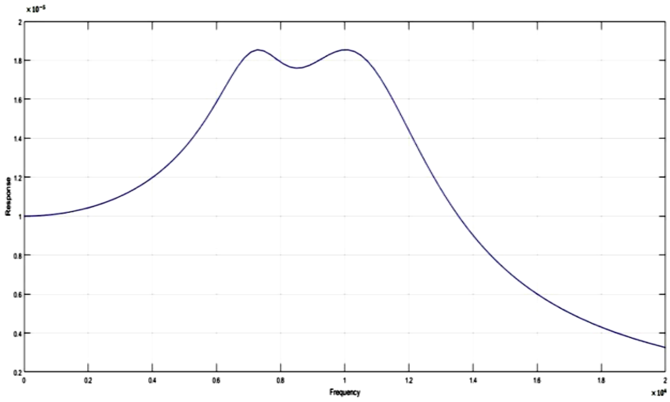

Fig 3.6 System with optimum absorber

Systems acceleration response is obtained using an accelerometer device readings. Noise formation is one of the major disadvantages. Extraction of desired output from experimental analysis is done through noise reduction techniques. The extracted frequency response of the system is compared through with/without absorber attachment.

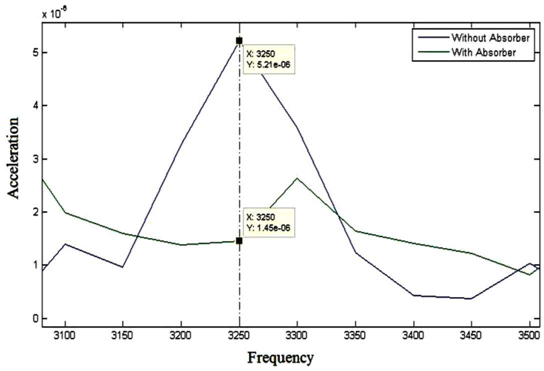

Fig 3.7 System with 200 rpm

Above Fig 3.7 shows the frequency response of the system having 200 rpm with and without absorber is connected. The highest peak is observed when the system operates under without absorber. That is at 3250 Hz the acceleration is 5.29μg. But when the system operates with optimized vibration absorber, at 3250 Hz the acceleration is reduced from its peak value to 1.38μg. At three different speeds frequency-response of the system is shown below.

TABLE II
FREQUENCY-RESPONSE

Depth of cut (mm)	Readings	Without Absorber		
		200 rpm	310 rpm	740 rpm
1	Frequency (Hz)	3250	12450	8100
	Response(μg)	5.29	1.5	20.27
With Absorber				
1.5	Frequency (Hz)	3250	12450	8100
	Response(μg)	1.38	0.79	12

CONCLUSION

In this work experimental study and optimization of dynamic vibration absorber is done. When the forcing frequency becomes equal to the natural frequency of the system, then the system will fail. The developed optimized vibration absorber for suppress the unwanted vibration or chatter is very effective for the system. Through experimental analysis and software analysis the vibration of the system attenuated due the attachment of optimized vibration absorber to the system. Mostly in manufacturing production industries are need of proper vibration absorbers.

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