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Analysis of chatter suppression in high-speed machining using tuned mass dampers

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Abstract:

A tuned mass damper also known as a harmonic absorber is a device mounted in structures to reduce the amplitude of mechanical vibration. Their application can prevent discomfort, damage or structural failure. Tuned mass dampers stabilize against violent motion caused by harmonic vibration. Machining vibrations called chatter correspond to the relative movement between the work piece and the cutting tool. These vibrations result in waves on the machined surface. The purpose of this project was to show the amplitude vibration of thin ribs during high-speed milling with two mass dampers is low as compared to thin ribs with one and no mass dampers

Keywords — **Tuned mass damper, natural frequency, thin ribs.**

I. INTRODUCTION

A tuned mass damper (TMD) is a device consisting of a mass, a spring, and a damper that is attached to a structure in order to reduce the dynamic response of the structure. The frequency of the damper is tuned to a particular structural frequency so that when that frequency is excited, the damper will resonate out of phase with the structural motion. Energy is dissipated by the damper inertia force acting on the structure. The TMD concept was first applied by Frahm in 1909 to reduce the rolling motion of ships as well as ship hull vibrations. It is a vibrating mass that moves out of phase with the motion of the structure it is suspended to. With its out of phase motion, the inertial force of the TMD mass abates the resonant vibration of the structure by dissipating its energy. The ideal extent of phase difference between the motion of the TMD mass and that of the structure, i.e., 90 degrees, is attained by tuning the TMD to the natural frequency of the structural mode targeted for damping. Tuned mass dampers have been used for adding tuned damping to various structures, successfully. A TMD is normally (but not always) tuned to the first natural frequency of the structure. The energy dissipation effectiveness of a TMD depends on a) the accuracy of its tuning, b) the size of its mass compared to the modal mass of its target mode, i.e., its mass ratio, and c) the extent of internal damping built into the tuned mass damper.

TYPES OF TUNED MASS DAMPER.

1. Active mass damper
2. Passive mass damper

An active control system is one in which an external power source the control actuators are used that apply forces to the structure in a prescribed manner. These forces can be used to dissipate energy from the structure.

A passive control system does not require an external power source. Passive control devices impart forces that are developed in response to the motion of the structure. Total energy cannot increase, hence inherently stable

ADVANTAGES OF TUNED MASS DAMPER

The main advantages of TMD is attractive as it dissipates a substantial amount of vibration energy of main structure without requiring any connection to ground

1. They do not depend on an external power source for their operation

2. They can respond to small level of excitation
3. Their properties can be adjusted in the field
4. They can also be introduced in upgrading structure

II. MATERIALS AND METHODS

The analyzation of the project is done by using aluminium alloy. An aluminium alloy is a chemical composition where other elements are added to pure aluminium in order to enhance its properties, primarily to increase its strength. These other elements include iron, silicon, copper, magnesium, manganese and zinc at levels that combined may make up as much as 15 percent of the alloy by weight. Alloying requires the thorough mixing of aluminium with these other elements while the aluminium is in molten – liquid – form. 6061 is a precipitation-hardened aluminium alloy, containing magnesium and silicon as its major alloying elements. Originally called "Alloy 61S", it was developed in 1935. It has good mechanical properties, exhibits good weld ability, and is very commonly extruded

TABLE I
PROPERTIES OF ALUMINIUM ALLOY

Density	2700kg/$\mathbf{m^3}$
Young's modulus	68.9 GPa
Poison's ratio	0.33
Tensile strength	160 MPa
Yield strength	280 MPa
Thermal conductivity	154W/mK

TABLE II
CHEMICAL COMPOSITION OF ALLUMINIUM ALLOY

Aluminium	95.85-98.56
Magnesium	0.8-1.2
Silicon	0.40-0.8
Iron	0.0-0.7
Copper	0.15-0.40
Chromium	0.04-0.35
Zinc	0.0-0.25
Titanium	0.0-0.25
Manganese	0.0-0.15

In this analysis we find the optimum location of dampers by positioning it at various locations and compare the amplitude response of thin ribs without, with single and double tuned mass damper.

III. DESIGN AND MODELLING

Modal analysis is a technique used to study the dynamic characteristics of a structure under vibration excitation. Natural frequencies and mode shapes of a structure can be determined using modal analysis.

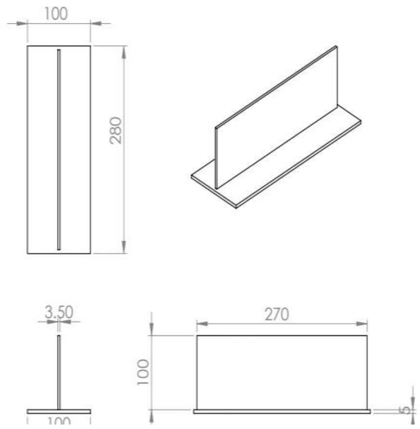

Fig.1. Dimensions of thin rib

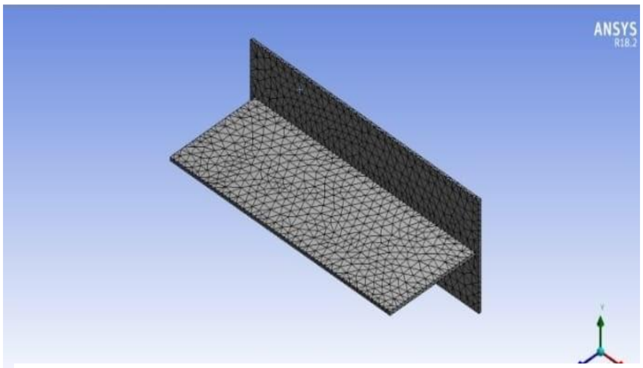

Fig.2.Thin rib meshing

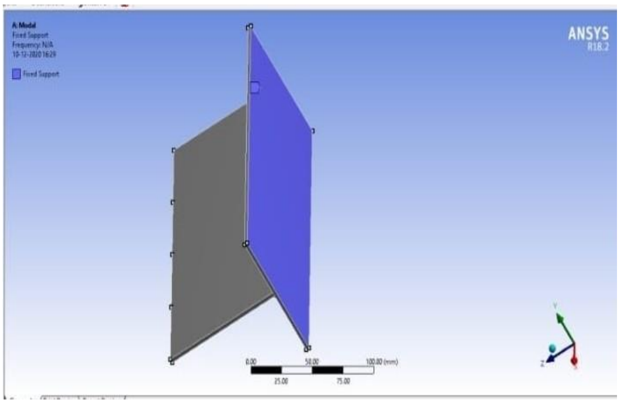

Fig. 3. Thin rib with bottom face fixed

IV. ANALYSIS

ANSYS Mechanical and ANSYS Multi physics software are non-exportable analysis tools incorporating pre-processing (geometry creation, meshing) solver and post-processing modules in a graphical user interface. These are general purpose finite element modelling packages for numerically solving mechanical problems, including static/dynamic structural analysis (both linear and non-linear) heat transfer and fluid problems, as well as acoustic and electric magnetic problems.

The analysis of thin ribs made up of aluminium alloy using ANSYS software using following boundary condition as the bottom face is fixed to analyse it as a cantilever beam. During harmonic analysis force is given at five locations.

V. RESULTS AND DISCUSSION

The results obtained from modal analysis and harmonic analysis are tabulated and discussed below.

TABLE III NATURAL FREQUENCY OF THIN RIB

TYPE	NATURAL FREQUENCY					
MODE	1	2	3	4	5	6
FREQUECY	268.0	355.4	588.6	994.8	1590	1713

Deformation is the transformation of a body from the initial condition to final condition i.e., at each mode number minimum, maximum deformation was given.

TABLE IV TOTAL DEFORMATION OF THIN RIB

MODE	1	*2*	*3*	*4*	*5*	6
Minimum(mm)	*0*	*0*	*0*	*0*	*0*	0
Maximum(mm)	*136.2*	*202.8*	*223.3*	*214.9*	*208.0*	225.8

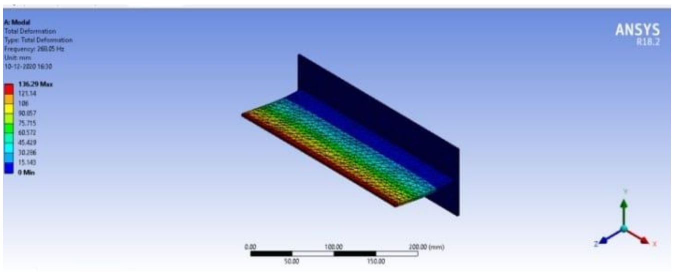

Fig.4.Total deformation at mode 1

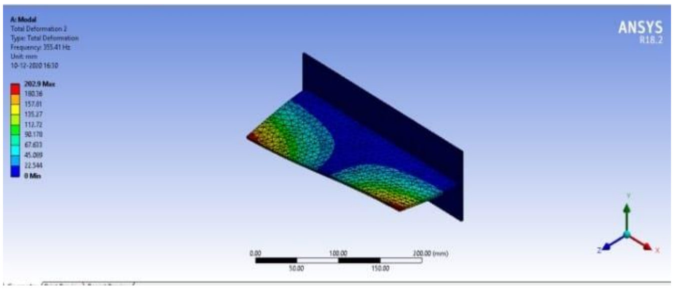

Fig.5.Total deformation at mode 2

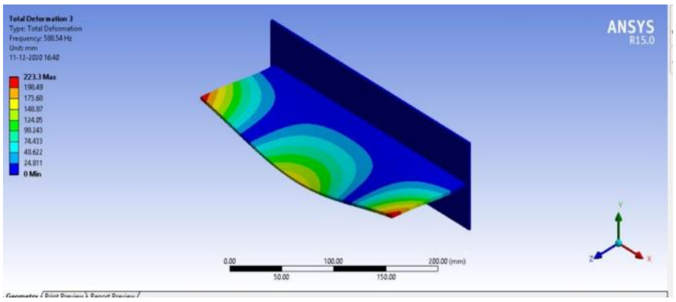

Fig.6.Total deformation at mode 3

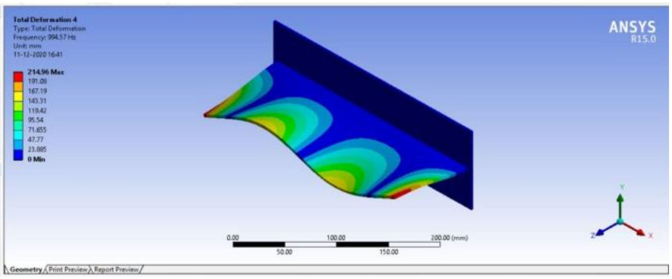

Fig.7.Total deformation at mode 4

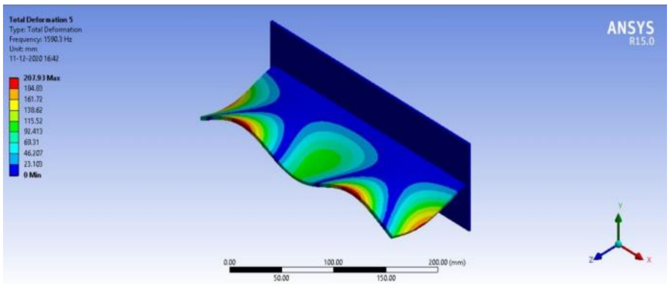

Fig.8 Total deformation at mode 5

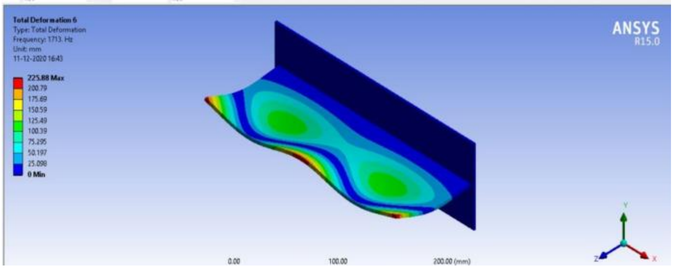

Fig.9. Total deformation at mode 6

Harmonic response analysis in ANSYS can be used to obtain steady-state response of a model subjected to a load that changes harmonically in time. Such load has specific amplitude and requency at which it acts

TABLE V RESPONSE AT POINT 1

Frequency (Hz)	Amplitude(mm)
220	0.104
240	0.119
260	0.842
280	0.128
300	0.175
320	0.323
340	0.646
360	2.864
380	0.595
400	0.13

TABLE VI RESPONSE AT POINT 2

Frequency (Hz)	Amplitude(mm)
220	0.119
240	0.204
260	0.842
280	0.128
300	0.175
320	0.132
340	0.236
360	1.364
380	0.395
400	0.13

TABLE VIII RESPONSE AT POINT 3

Frequency (Hz)	Amplitude(mm)
220	0.119
240	0.204
260	0.842
280	0.128
300	0.017
320	0.013
340	0.023
360	0.034
380	0.039
400	0.013

Response obtained by placing single mass damper at various location at 350Hz are given

TABLE IX OPTIMUM LOCATION OF DAMPER

		Location along length (mm)		
		51	**52**	**53**
Location Along height (mm)	**100**	0.097	0.096	0.098
	99	0.364	0.362	0.361
	98	0.356	0.357	0.355
	97	0.096	**0.0946**	0.096
	96	0.145	0.145	0.143

The point at which amplitude of vibration is minimum is chosen as the optimum location of damper.

Response obtained by placing single mass damper at optimum location, and force at all five points are tabulated below.

TABLE X RESPONSE WITH 1 TMD

Frequency(Hz)	Amplitude (mm)				
	With respect to the point of application of load				
	At point 1	At point 2	At point 3	At point 4	At point 5
220	0.14	0.12	0.14	0.12	0.11
240	0.26	0.22	0.19	0.17	0.13
260	1.15	1.09	1.03	0.94	0.69
280	0.15	0.17	0.15	0.22	0.24
300	0.16	0.05	0.14	0.11	0.30
320	0.20	0.11	0.16	0.13	0.31
340	0.56	0.3	0.11	0.48	1.10
355	0.10	0.17	0.12	0.12	0.22
380	0.63	0.27	0.11	0.11	0.48
400	0.14	0.12	0.11	0.12	0.11

Response obtained by placing Two mass dampers at optimum location, and force at all five points are tabulated below.

TABLE XI RESPONSE WITH 2 TMD

Frequency Hz	Amplitude (mm)				
	With respect to the point of application of load				
	At point 1	At point 2	At point 3	At point 4	At point 5
200	0.11	0.14	0.11	0.14	0.11
250	0.26	0.52	0.53	0.51	0.48
300	0.13	0.11	0.12	0.12	0.18
355	0.11	0.11	0.11	0.31	0.14
400	0.27	0.26	0.13	0.22	0.12

VI. COMPARISON OF RESULTS

Comparison of frequency responses of thin ribs without damper, with1 TMD and with 2 TMD for force at point1 are shown in the figure

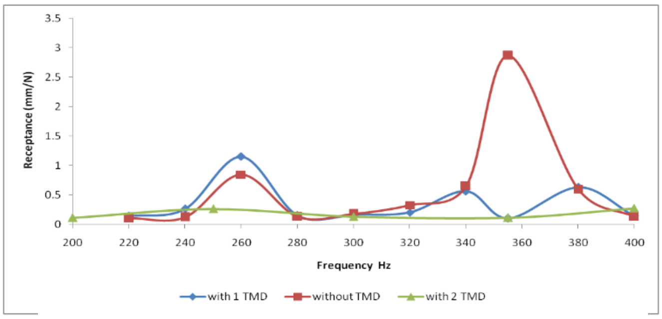

Fig.10.comparision of response at point 1

From the above graph, by using thin ribs with 2 TMD the receptance at 355 Hz get reduced by 0.01 mm and 2.7 mm comparing thin ribs with 1 TMD and thin ribs with no mass damper

Comparison of frequency responses of thin ribs without damper, with1 TMD and with 2 TMD for force at point 2 are shown in the figure

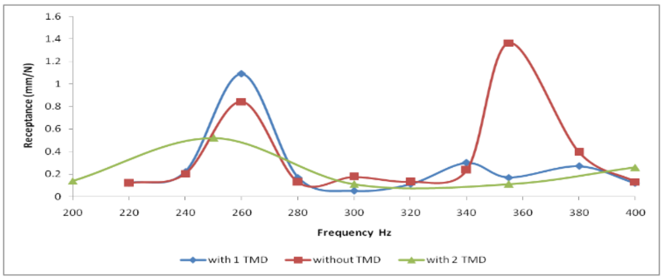

Fig.11.comparision of response at point 2

From the above graph by using thin ribs with 2 TMD the receptance at 355 Hz get reduced by 0.06 mm and 1.25 mm comparing thin ribs with 1 TMD and thin ribs with no mass damper.

Comparison of frequency responses of thin ribs without damper, with 1 TMD and with 2 TMD for force at point 3 are shown in the figure

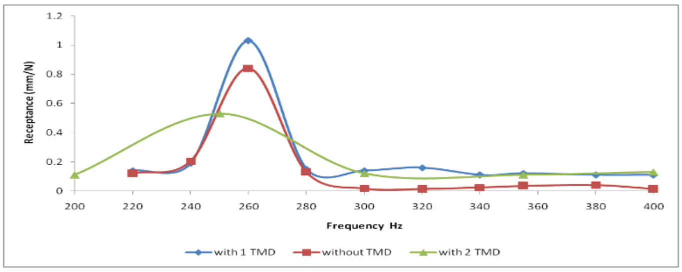

Fig.12.comparision of response at point 3

From the above graph by using thin ribs with 2 TMD the receptance at 355 Hz get reduced by 0.01mm comparing thin ribs with 1 TMD and increased by 0.076 mm by comparing thin ribs with no mass damper.

Comparison of frequency responses of thin ribs without damper, with 1 TMD and with 2 TMD for force at point 4 are shown in the figure

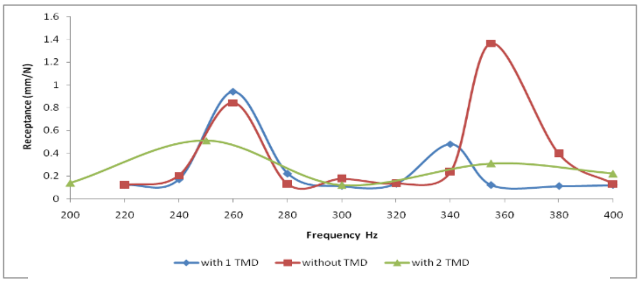

Fig.13.comparision of response at point 4

From the above graph by using thin ribs with 2 TMD the receptance at 355 Hz get increased by 0.19 mm comparing thin ribs with 1 TMD and reduced by 1.05 mm comparing thin ribs with no mass damper.

Comparison of frequency responses of thin ribs without damper, with 1 TMD and with 2 TMD for force at point 5are shown in the figure

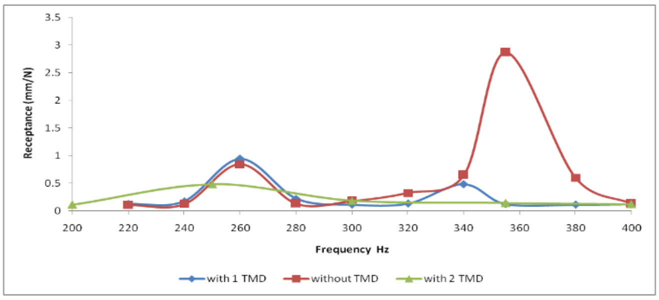

Fig.14.comparision of response at point 5

From the above graph by using thin ribs with 2 TMD the receptance at 355 Hz get reduced by 0.08 mm and 2.724 mm comparing thin ribs with 1 TMD and thin ribs with no mass damper

VII. CONCLUSION

From the above results we can conclude that, by using tuned mass dampers the mean amplitude of vibration reduced. The effectiveness of using mass damper system in a vibration system is greatly influenced by the location of mass damper. Thus, reducing the amplitude of vibration in milling process is essential as it greatly influences the surface finish of the work piece. It is also clearly shown that two tuned mass dampers is more efficient compared to single tuned mass damper. In this project by using Two Tuned mass dampers at optimum location the amplitude of vibration at 355Hz during force at point 1, point 2, point 4 and point 5 get reduced and amplitude of vibration during force at point 3 slightly increases. But in overall the mean amplitude of vibration of thin rib with 2TMD is less compared with thin rib having 1TMD and no TMD.

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