Effect of Air Damper to Improve the Dynamic Performance for High Speed Vibration Cutting Tool; Simulation and Experiment.

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1. Introduction

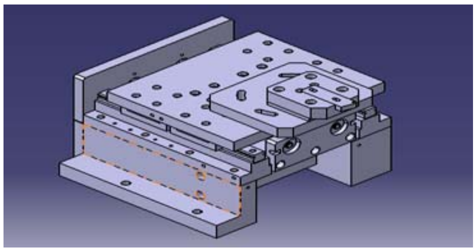
This paper describes an ongoing project to model a cutting tool post with help of computer using multibody simulation software and evaluating the system developed by the laboratory experiment. The objective is to make the virtual tool post frequency behavior close to the real frequency of the tool post itself. This process has the capabilities to suppress the resonant frequency.

W.O Wong and Y.L Cheung[1] designed a damped dynamic absorber for vibration control to reduce the transmission of motion from the support to the mass of the structure and were able to suppress the resonance frequency. Here we tried to find out the natural frequency of a virtual tool post designed and compared the simulation results with the experimental results. All these analysis is done to apply vibration cutting to cut the PMMA (Polymethylmethacrylate), in which one dimensional vibration cutting will be applied instead of the elliptic vibration cutting[2-4] due to amorphous material characteristic, low space at the cutting tool and light weight of the tool post. It is already proved that in vibration cutting the required amount of force is low to cut the material and as a result of it the surface quality is better than the material cut by conventional cutting. In our current application the challenge that we faced with the tool post to suppress its natural frequency as it is vary light and when we applied two pneumatic dampers at the both side of the tool post the natural frequency has reduced at an increasing rate as the number of mode increased. When we have found that in some ongoing project for the vehicle system to increase the durability by virtual simulation[5] we became interested to make a virtual part of our system and optimize it by suppressing its natural frequency which will eventually help us to apply the ultrasonic vibration for cutting PMMA. In most of the vibration cutting the machine tool that used for cutting has very high mass [4-used CNC lathe] as a result the natural frequency was very low for which the vibration of the machine tool has very low almost no effect on the vibrating cutting tool and there was no chance to resonate between the machine tool and the cutting tool itself. But our tool post is very small and that's why we analyzed the whole tool post to find out its natural frequency and applied the air damper to suppress the resonance frequency so that it can not resonate with the applied frequency and the motion from the upper part transmits to the lower fixed part.

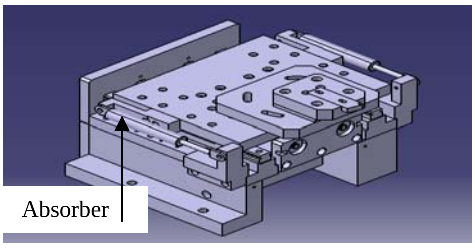
2. Modeling and Simulations

Here we divided the whole system into two main parts, one part that remain stationary and another part that moves. The upper moving part is connected with the base by air damper. It was possible to connect these two parts by some fixed support but as the upper part is used to give particular feed rate by servo motor we used air damper. Two lead screw and two linear motion guides has been used as guide. For modeling purposes most of the standard has been maintained and some data has been used from the manufacturer standard. In our present analysis we have used Catia V5 tool as modeling and simulation software, some parts were not designed like bearings, linear motion guide and servo motor, belt, timing gears and pulley. Among those only the linear motion guide design parameters has been considered during the simulation and for others we assumed some parameters like we assumed there is no effect of the bearings in our vibration analysis but for real case it has effect.

We did our simulation considering both with absorber and without absorber for without absorber we consider the hole part as an assembly with no external load and did the simulation in frequency analysis case but when we considered the presence of the absorber and due to the vibration as the absorber will react to the vibration of the tool plate and will transfer its motion to the fixed part. It is wiser to do the analysis for the harmonic dynamic response case but we assumed that the absorber has no effect on shock absorbing without increasing the mass and the stiffness of the system and we followed the free vibration analysis here.



a. without absorber



b. with absorber

Fig.1 Three dimensional model of tool post.

3. Simulation Results (Free Vibration):

Table.1 Comparison of simulation results.

Mode	Without Absorber (Hz)	With Absorber(Without Considering Damping Force) Hz
1	872	512
2	1321	786
3	1471	868
4	2134	1121
5	2216	1288
6	3324	1509
7	3818	1827
8	4096	1831
9	4422	1859
10	4496	1968

This simulation results contains only the comparison between without absorber and with absorber and here when we consider with absorber that time we consider only the mechanical parts of the absorber and assuming the damping force of the absorber is zero and here we have observed two things, one is for first mode the natural frequency reduced almost 41% and as the number of mode increasing the difference between the frequencies of tool post , for the case of both with and without is increasing but the increasing rate for individual one is higher for the without absorber condition than with absorber condition. In our analysis case we ignored the effect of the damping force for the very low damping effect but in real case this is impossible for which we were expecting some deviations from the simulation results, moreover in our real system we have a timing belt mechanism but for simulation simplification we ignored.

4. Experimental results and the comparison with the simulation results:

Experiment is done on the basis of load excitation for which we have used Triaxial Charge Accelerometer (B&K) and this was limited within the range of 1.6 kHz. So we compared the results only for first few modes and we have found some missing modes from the simulation results which we have considered the effect of not considering the damping effect and the timing belt stiffness.

Table.2 Experimental data table and error estimations.

Modes	With absorber	Without absorber	Error(%) For with absorber	Error(%) For without absorber
1	M*	741	NA	15
2	747	136 1	4. 96	3. 03
3	824	147 5	5. 06	0.27
4	119 7	M**	6. 77	NA
5	141 3	M**	9.70	NA

M: Not found in experiment, M**:Impact hammer Limitation..*

During resonate vibration, the response displacement may increase until the structure experiences buckling, yielding and fatigue or some other failure mechanism. So we were also interested to find out the stress concentrated points and found as the motor plate, bearings, linear motion guide for the tool post without absorber, and for with absorber we have observed that the stress in concentrated on the absorber part too.

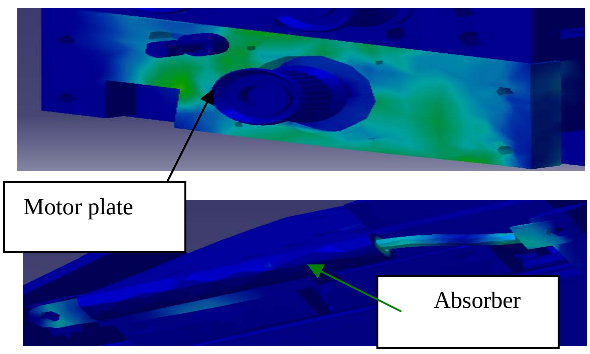


Fig.2 Stress Concentration for motor plate and for the absorber.

5. Conclusion:

From present analysis we have found that using air damper is quite useful to suppress the natural frequency when the cutting tool post is very small as it reduces the natural frequency 56.23% up to mode ten and at the first mode 41% theoretically and experimentally 45.11 %(2nd mode as example). After using the absorber we have seen that the natural frequency increasing rate became almost linear and as the damping ratio of the system should increase after using the absorber the stabilization of the system will be much faster than the system without absorber. But another consideration is that we are cutting very soft material PMMA and for which the cutting force will be very small without at the initial impact. As without applying the damping force the natural frequency has been reduced that may be the result for the additional mass and reaction force of the two absorbers (30N) that increased the stiffness of the system.

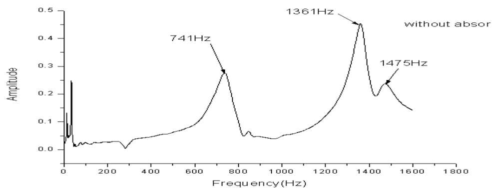


Fig.3 Experimental Fourier Analyzed Data Plot

6. Further predictions and objectives:

As our present analysis is free vibration analysis we are more interested to find out the real vibration manner during the cutting experience and for that we are needed to measure the real cutting force which we will use for our load excitation and as the real cutting process is very fast nearly 0.5m/s feed rate so we are assuming the impact force will be high and during writing this paper we were developing the system to measure the cutting force by using dynamometer. We have already measured the Global damping ratio which is nearly 1.25% and after measuring the cutting force we will use it for our load excitation in harmonic dynamic response case analysis purpose and then we will compare Frequency Response Function and the experimental results to find out the vibration behavior of the real system. It will help us to apply Ultrasonic Vibration Cutting in choosing the applied frequency for cutting purposes and especially for one dimensional vibration cutting to select the mode.

7. References:

1. W.O.Wong, Y.L.Cheung, "Optimal design of a damped dynamic vibration absorber for vibration control of structure excited by ground motion," Engineering Structures," 30,282-286(2008).
2. N. Suzuki, M. Haritani, J.Yang, R.Hino, E.Shamoto, "Elliptic vibration cutting of tungsten alloy molds for optical glass parts," Annals of the CIRP Vol.56/1/2007
3. Jung-Hwan Ahn, Han Seok Lim and Seong-Min Son,"Characteristic of micro-machining using two dimensional tool vibration," International Journal of the Korean Society of Precision Engineering vol.2, No, 3, September 2001.
4. Chandra Nath and Mustafizur Rahman ,"Evaluation of ultrasonic vibration cutting while machining Inconel 718,"Internationl Journal Precision Engineering and Manufacturing. vol.9, No.2.
5. W.B.Ferry, P.R.Frise, G.T.Andrewsa, Malik ,"Combining virtual simulation and physical vehicle test data to optimize durability testing,"2002,DaimlerChrysler Corp, Fatigue Fract Engng Matter Struct 25.1127-1134