OPTIMUM DESIGN OF A DAMPED ARBOR FOR HEAVY DUTY MILLING

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ABSTRACT

The design method for a tuned mass damper imbedded tool arbor was investigated which suppress chatter vibration and improve cutting performance during machining of large mechanical parts. The design of the damped arbor is depends upon the dynamic stiffness and also the spring constant. Rayleigh's method coupled with displacement ratio is taken into account to calculate the dynamic stiffness of the damped arbor body. The design of the damped arbor is changed by making the cylindrical hallow to tapered. The optimal design is to be found by considering the different dimensions of the tapered hollow damped arbor by keeping the volume constant. firstly the optimal stiffness constant is to be found from the minimum compliance value obtained from the Rayleigh's method and using this as the threshold constant, optimum design of the tapered hollow damper is achieved. The results of the tapered hollow arbor is taken into graph from which optimum dimensions are taken using the threshold spring constant. experimentally which proves that damped arbor prototyped by the proposed design method has better cutting performance than a conventional one.

The natural frequency of the damped and undamped arbors calculations done theoretically computational model also made by using catia and model analysis is carried out for the analytical result of natural frequencies. The material body used for the damped arbor is Chromium Molybdenum and fabrication of the arbor is will be according to the dimensions calculated and experimental tests conducted for the dynamic stiffness.

I. INTRODUCTION

In recent years, Metal cutting process is improved in its efficiency in the surface finish of the material. The efficiency depends on the properties of work piece, feed rate, spindle rotation and time taken in machining. Long slender cutting tools are used to make large cavities and holes for the large materials where the chattering noise and vibrations occur. These noises could damage the cutting edges and deteriorate the surface finish. Properties of work material and dynamic stiffness of the tool affect the chatter noise. Every property depth of cut, revolution of speed, dynamic stiffness of cutting tool, material are effective to minimize the chattering noise. Dynamic stiffness is increased by increasing the static stiffness or by improving the damping behaviour. As the size of the tool has a limitation, static stiffness cannot be increased. So damper is attached to the structure which increases the damping ratio of the structure.

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Imbedding the viscous damper or mass damper into the bar or machine tool was proposed. According to the researches done in by putting practical use, stated its possible that damping increases the cutting efficiency and chatter suppression. Reyleigh's method is used for deriving compliance by introducing displacement ratio where the minimization of the maximum negative real part is taken up. The natural frequency of the damped arbor is obtained by model analysis using ANSYS software.

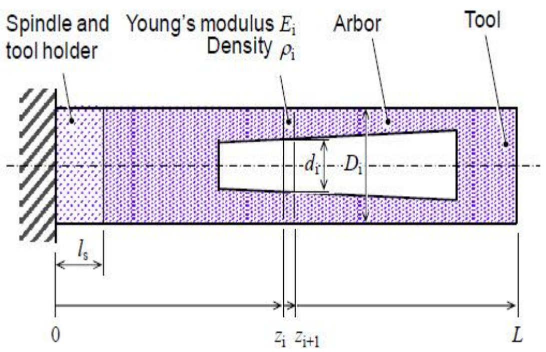

Fig 1: Damped arbor

II. OPTIMIZATION OF MASS DAMPER FOR A DAMPED ARBOR WITH GENERALIZED PROFILE

Arbor is composed with the damper mass inside the hallow space provided in the arbor. Here, we consider two degree of freedom with arbor body stiffness k_1, mass m_1 (chromium molybdenum) and damping coefficient c_1. The second degree of freedom is taken considering damper mass m_2(cemented tungsten) attached to the spring stiffness k_2 with damping coefficient c_2. Where x_1 and x_2 are the displacement with cutting force f_1 acting on arbor tool.

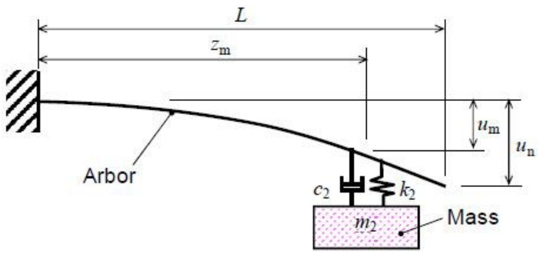

Fig 2: Consideration of the position of mass

Length of the damped arbor L including the spindle holder is taken and assuming n elements with thickness Δz . The deflection at the nth element is given by u_n. The distance Z_m from the centre of gravity is taken to define the displacement at the m_{th} element u_m. And we introduce displacement ratio α to the Rayleigh's method to determine the dynamic stiffness.

$u_n = \sum_{j=1}^{n} \Delta\,\theta_j\,\Delta z \qquad u_m = \sum_{j=1}^{m} \Delta\,\theta_j\,\Delta z$

$\alpha = \frac{u_m}{u_n}$ $\quad \Delta\theta$ = angle increment

The model mass m_1 is given by the following equation

$m_1 = \sum_{j=1}^{n} w_j \left(\frac{u_j}{u_n}\right)$ ....Eq 1

The equations of motion for the above equation is taken and compliance of the vibration system is given by

$[\emptyset] = ([K] + j\omega[C] - \omega^2[M])^{-1}$........Eq 2

$c_1 = 2\sqrt{m_1 k_1 \vartheta_1}$; $\quad c_1 = 2\sqrt{m_2 k_2 \vartheta_2}$

The compliance value of the above equation 2 is calculated for both the cylindrical and tapered hallow damped arbor considering hallow space l_1 = 150 (fig 3) . The spring constant k_2 which minimizes the maximum negative real part of the compliance is taken(fig 4) as the optimum value that decreases the chatter

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noise and improves the dynamic stiffness. The static stiffness value may decrease slightly so that threshold stiffness(fig 5) is taken to define the optimum dia of the hallow space in both the damped arbors.

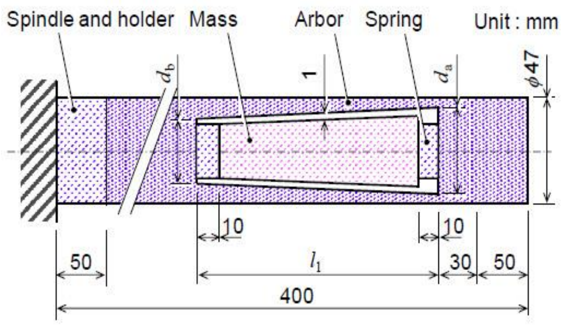

Fig 3 : Geometry of damped arbor

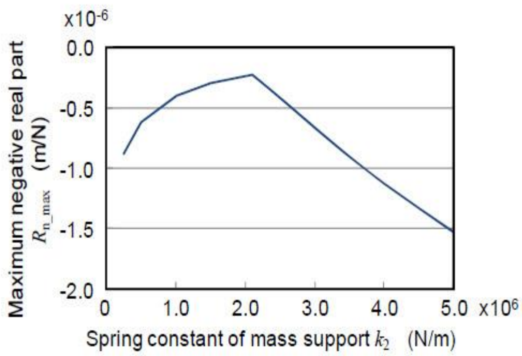

Fig 4: Relation between k_2 and R_{n_max}

By calculating the maximum negative real part with respect to all the spring stiffness from 0 to 5, where 2.2×10^6 found to be optimum value which minimize the compliance value or increase the dynamic stiffness.

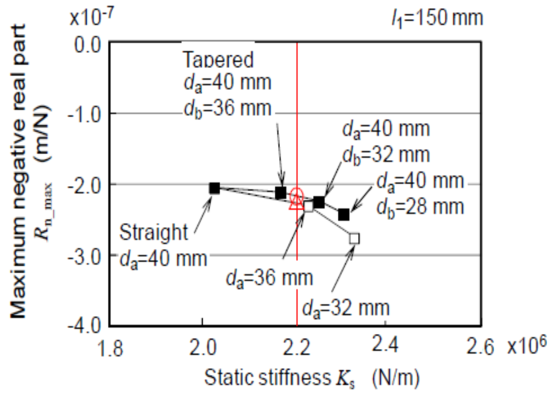

Fig: 5 comparison between static and dynamic stiffness

From the above figure negative real part is calculated for both the cylindrical and tapered damped arbor by considering threshold stiffness 2.2×10^6. Both the lines coincide at threshold point where dia of cylindrical taken as 36mm and taper outer dia 40 and inner dia 32 is consider where as both the volumes considered to be same. With which the kinetic energy at the end of the tool is reduces more by tapered damped arbor than cylindrical damped arbor.

Theoretical Calculation for the Natural Frequency of Damped Arbor

The frequency at which a system oscillates when not subjected to a continuous or repeated external force**.** Newton's second law is the first basis for examining the motion of the system. Natural frequency of both the damped arbor is calculated from the following formulae.

The natural period of the oscillation is established from $\omega_n \tau = 2\pi$, or

$$T= 2\pi\sqrt{m/k}$$

$k_2 = 2.2\text{x}10^6$ N/m, $m_2 = 1.77$Kg,
density = $8\text{X}10^3$ Kg/m^3

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Natural frequency of the 1st order mode fc_1cylindrical damped arbor and f_{t1} tapered damped arbor is expressed as the following equation.

$$f_1 = \frac{1}{2\pi}\sqrt{\frac{k_2}{m_2}},\ f_{c1} = 174 \text{ Hz} \ \& \ f_{t1} = 184 \text{ Hz}$$

Frequency of natural vibration of twisting mode fc_2 and f_{t2} is expressed as follows.

$$f_1 = \frac{1}{2\pi}\sqrt{\frac{k_2}{J_2}},\ f_{c2} = 298 \text{ Hz} \ \& \ f_{t2} = 322 \text{ Hz}$$

III. MODELLING AND ANALYSIS

Modelling of the arbor is done by using CATIA V5 software. The dimensions of the damped arbor is taken from the optimum values obtained from the results. Diameter of the spring is taken as 30 and length 10mm on the both sides of the damper mass which actuates the damping when cutting process is going on. Arbor is modelled including its tool holder length 50mm and maximum dia of the tapered hallow is taken near the tool and damping mass is inserted inside the hallow space with 1mm clearance.

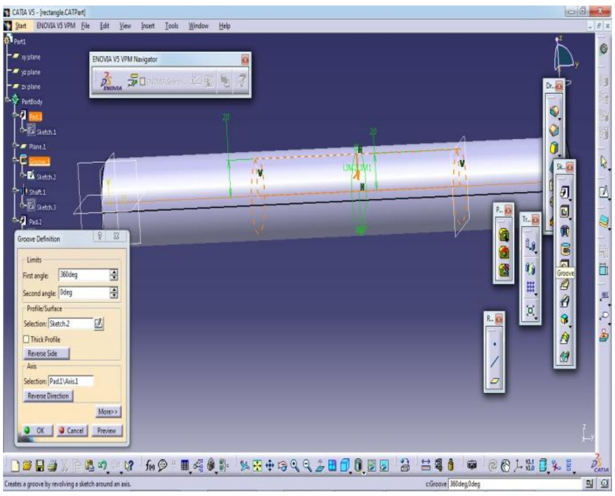

Fig 6 Cylindrical damped arbor

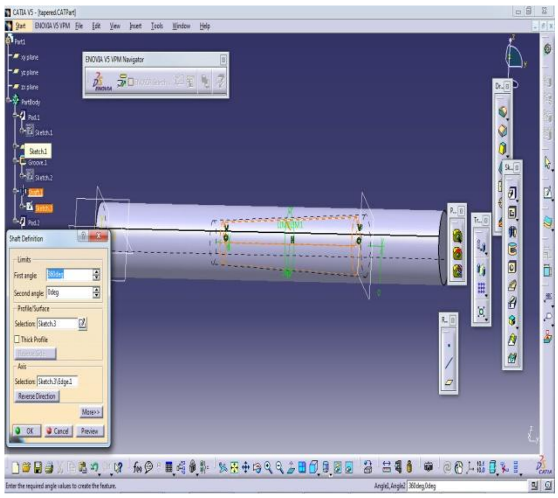

Fig 7 Tapered damped arbor

Modal analysis is done by using ANSYS software where natural frequency for both the circular (fig 8) and tapered hallow damped arbor (fig 10) is obtained and compared with the calculated results. In which we observe 5% error rate between them. Model analysis is done in two modes, the damped mass is observed to be twisted in second mode where we find the maximum damping that subjected to the arbor body. According to the results obtained, tapered hallow damped arbor has slightly high natural frequency than cylindrical hallow.

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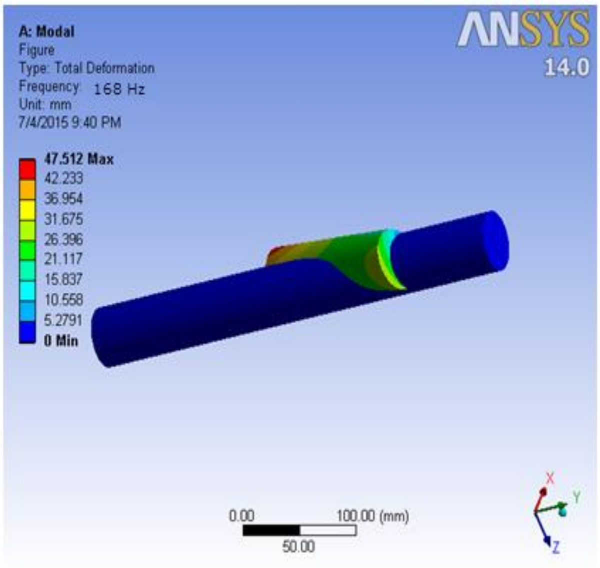


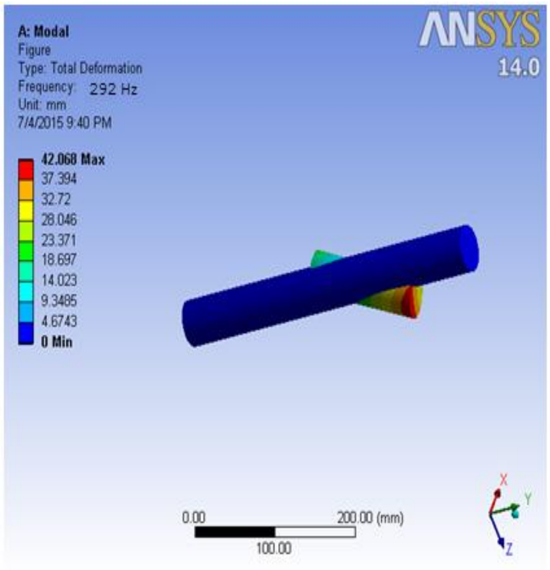
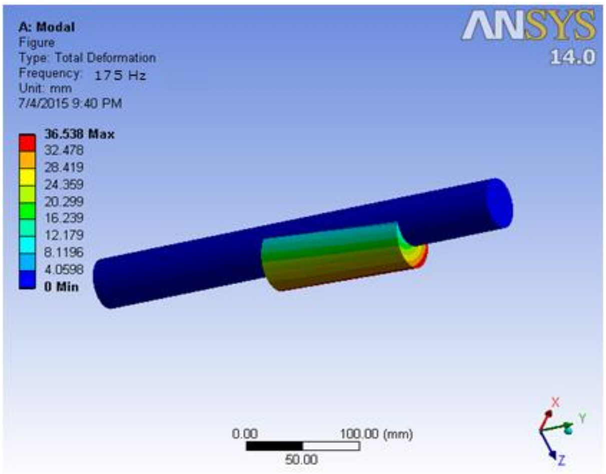


Fig 8: Mode 1 frequency

Fig 9: Mode 2 frequency



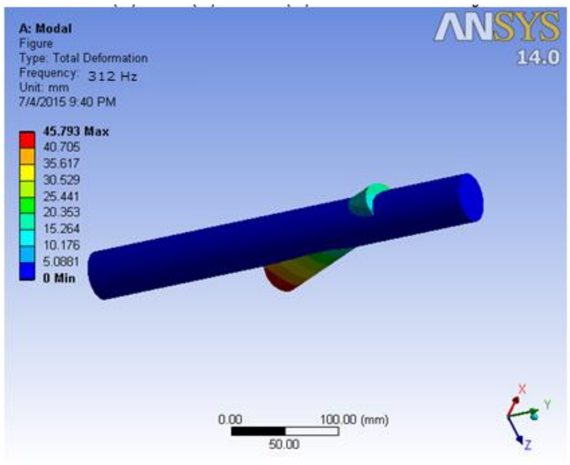


Fig 10: Mode 1 frequency

Fig 11: Mode 2 frequency

IV. FABRICATION OF DAMPED ARBOR

Fabrication of the damped arbor is done by using lathe machine. The material chromium molybdenum steel (AISI 4140) has a 47mm diameter, density is $8.0x10^3$ kg/m^2 and Young's modulus is $2.1x10^2$GPa is taken as a arbor body and tungsten carbide is used for the mass material it's density is $16x10^3$ kg/m^2 and young's modulus is $6.8x10^2$GPa . A hole of 32mm is made through the body of the arbor and 40 mm dia hole is done up to 80 from the end of the arbor. The tapering is done from 40 to 32mm by inclining the tool post to the required angle. The damping mass is made such that it fits inside the tapered hallow space with 1mm clearance and providing springs on the both ends of the mass. The tool is attached to the end

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of the arbor. Then the damped arbor is ready for machining the material with given feed rate and spindle rotation.

Fig 12: Fabrication process

Fig 13: Tapered hallow arbor

DISCUSSION

- The rayleigh's method is coupled in the proposed analysis for the dynamic stiffness assuming that mass give a counter force at the centre of gravity of it. Amplitude ratio is introduced for the calculation of the compliance where the compliance is calculated without the amplitude ratio and compared. It states that mass is connected at the end of the tool when amplitude is not included. So assuming amplitude ratio gives best results.
- The bending moment will be minimum near the tool and maximum whilst at the tool holder. The kinetic energy is more at the tool end. The tapered hallow damped arbor reduces the kinetic energy more than the cylindrical hallow arbor considering same volume for both the arbors.
- The stiffness constant affect compliance with the length of the arbor drastically. when the stiffness is increased compliance will be changed for the larger arbor. So optimum stiffness value of larger arbor is considered for the small arbor with negligible deterioration of dynamic stiffness.

V. CONCLUSION

Main objective is to suppress chatter vibration and improve cutting performance during the machining. The dynamic stiffness is calculated by using Reyleigh's method implementing displacement ratio. The maximum negative compliance value is calculated for using different spring constants and optimal spring constant is taken which by value of maximum negative compliance. By making the optimal stiffness value as a threshold value the different diameters are considered for both cylindrical hollow arbor and tapered hollow arbor and compliance values are determined. dimensions of tapered hollow arbor is measured at the threshold stiffness value and also the volume of both the damped arbors are kept same. The natural frequencies of the damped arbor is calculated theoretically in two different modes. The theoretical and analysis are done. The dimension of the optimal design is noted and fabricated using lathe

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machine. Chromium Molybdenum is used as arbor body material, Tungsten carbide is used as a mass material. The springs are also selected according to the spring constant calculated.

The experimental test is carried on the fabricated tapered hollow arbor and it is concluded that the depth of cut is more when compared to solid and cylindrical hollow arbors. The dynamic stiffness of the arbor also obtained. The tapered hollow space can reduce kinematic energy more than the straight one.

REFERENCES

(1) Seto, K. and Yamada, K., An investigation on boring bars equipped with a dynamic absorber, Proceeding of the 4th International Conference on Production Engineering, Tokyo, (1980)pp.422-427.

(2) Seto, K., Dynamic vibration absorber and its applications, (2010), Corona Publishing.

(3) Madoliat, R., Hayati, S. and Ghalebahman, A.G., Modeling and analysis of frictional damper effect on chatter suppression in a slender endmill tool, Journal of Advanced Mechanical Design, Systems and Manufacturing, Vol.5, No.2, (2011) pp.115-128.

(4) Rashid, A. and Nicolescu, C.M., Design and implementation of tuned viscoelastic dampers for vibration control in milling, International Journal of Machine Tools and Manufacture, Vol.45, No.9 (2008) pp.1036-1053.

(5) Sims, N.D. Vibration absorbers for chatter suppression: A new analytical tuning methodology, Journal of Sound and Vibration, Vol.301, (2007) pp.592-607.

(6) Saffury, J. and Altus, E., Optimized chatter resistance of viscoelastic turning bars , Journal of Sound and Vibration, Vol.324, (2009) pp.26-39.

(7) Rivin, E.I. and Kang, H., Enhancement of dynamic stability of cantilever tooling structures, International Journal of Machine Tools and Manufacture, Vol.32, No.4, (1992) pp.539-561.

(8) Pratt, J.R. and Nayfeh, A.H., Chatter control and stability analysis of a cantilever boring bar under regenerative cutting condition, Philosophical Transactions of the Royal Society of London, Part A, Vol.359, (2001) pp.759-792.

(9) Semercigil, S.E. and Chen, L.A., Preliminary computations for chatter control in end milling, Journal of Sound and vibration, Vol.249, No.3, (2002) pp.622-633.

(10) Merritt, H.E., Theory of Self-Excited Machine-Tool Chatter, Transactions of ASME, Journal of Engineering for Industry, Vol.87, (1965) pp.447-454.

(11) Den Hartog, J.P., Mechanical Vibrations, 4th Edition, (1956), McGraw-Hill.

(12) Tlusty, J. and Polacek,M., The stability of machine tool against self-excited vibrations in machining, ASME Proceeding of Engineering Research Conference, Pittsburgh, (1963), pp.454-465.

(13) Tobias, S.A., Machine Tool Vibration, (1965), Blackie.

(14) Ota, H., Kito,M. and Handa.T., A study on primary chatter caused by mode coupling of machine tool structure, Transactions of JSME, Series C, Vol.73, No.726, (2007), pp.208-214.

(15) Kondo, E., Tanaka, H. and Kawagoshi, N., Detection of self-excited chatter caused by mode coupling(1st report, Study on criterion of detection), Transactions of JSME, Series C, Vol.64, No.625, (1998), pp. 413-418.

(16) Zhang, X. J., Xiong, C.H., Ding, Y., Feng, M.J. and Xiong, Y.L., Milling stability analysis with simultaneously considering the structural mode coupling effect and regenerative effect, International Journal of Machine Tools & Manufacture, Vol.53, (2012), pp.127-140.

ISSN: 2278 – 7798

(17) Suzuki, N., Ikada, T. Hino, R. and Shamoto, E., Comprehensive study on milling conditions to avoid forced / self-excited chatter vibrations, Journal of the Japan Society for Precision Engineering, Vol.75, No.7,(2009), pp.908-914.

(18) Madoliat,R. , Hayati,S. and Ghalebahman,A.G. , Modeling and analysis of frictional damper effect on chatter suppression in a slender endmill tool , Jounal of Advanced Mechanical Design , Systems , and Manufacturing , Vol.5 , No.2 (2011), pp.115-128.

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