A Flexible Mechanism Based Vibration Isolator for Machine Tool Application

Nitin Vijay Satpute and Marek Iwaniec

1Vishwakarma University, Pune, India, email: nitinsatpute123@gmail.com
2AGH University of Science and Technology, Al. Adama Mickiewicza 30, 30-059 Kraków, Poland, email: iwaniec@agh.edu.pl

Abstract. The paper presents novel design of vibration absorber with innovative features including use of flexible link based mechanism at the interface of tool holder and cutting tool. The mechanism ensures modification of the dynamic force interaction at the damping element and results in lower force transmissibility. It ensures amplification of the relative velocity at the damping element, which results in significant reduction of the damping element mass used for energy dissipation. The presented absorber has advantages of passive and economical operation in comparison to the active and semi-active solutions. Further, the proposed solution results in up to 53% reduction in the force transmissibility. A real size design has been presented for frequency range of 0-1100 Hz and maximum force amplitude of 700 N. Numerical simulations have been performed with consideration of flexible joint and structural element dynamics. Simulation results with FEA and PRBM approach have been compared with detailed analysis of the important design parameters.

1 Introduction

Boring is an important operation to produce internal cavities and have the tools in the form of a cantilever. Boring tools have higher slenderness ratio and lower dynamic stiffness. Deep internal boring operation is a typical example of high chatter due to these reasons. Increased chatter results in inferior surface finish, limits cutting speed and productivity. Further, recent trends of high speed machining have further aggregated the problem of machining chatter during boring. Recently attempts are being made to incorporate controlled damping and stiffness at the tool holder to attenuate the chatter vibrations and ensure better productivity.

Resonance frequency of the machines used during boring operations is 750-800 Hz, whereas the dominant vibration amplitude between the frequency range of 590Hz -630 Hz [1]. Passive damping solution involves incorporation of a dissipative element at the interface of tool holder and machine. Takeyama et al. recommended statically high stiffness structure along with incorporation of high damping material in the tool holder for better machining performance [2]. Schmitz et al. investigated contact stiffness and damping at the tool holder interface in order to attenuate vibrations near 800 Hz and 1200 Hz [3]. Freyermuth et al.integrated a damping element in a boring tool holder in the form of an elastically deformable mass with controlled stiffness [4]. Denkena et al. integrated a friction damper into a long tool holder shaft for increasing depth of cut by 75% for a long...
projecting tool holder [5]. Lee et al. determined optimal location of the passive vibration absorber in the tool holder to minimize vibrations due to an external source [6]. Authors have demonstrated application of the method for multi-variable isolator placement in ultra-precision machining. Sasaki et al. patented a tool holder with a ring shaped anti vibration element, which has a provision for adjustable damping [7]. Mohan and Natarajan reduced relative displacement between tool and work piece by using MagnetoRheological damper to improve chatter vibration performance of a boring operation [8]. Eichelberger fitted a passive vibration absorber in the central cavity of a boring bar. Further an adjustable wedge was provided to facilitate stiffness tuning of the absorber according to the forcing frequency [9]. McCormick and Filho used an axial sleeve in the form of damping element for machining vibration attenuation [10]. Ziegert et al. investigated use of internal dampers in the cutting tool for milling chatter vibration control [11]. The tool holder geometry was modified to include the damping material and 53% increase in depth of cut was reported. Langbein and Lembke used shape memory alloy based damping element in tool holder chucking for reduction of machining vibrations [12]. Singh et al. modulated damping factor and natural frequency of a viscoelastic structure with microscopic and macroscopic channels [13]. Improvement in the damping factor has been attributed to frictional energy dissipation at the interface of the micro-channels. Chang et al. investigated vibration isolation in a machining center using fixture fitted with a passive isolator [14]. Experimental observations were used in an orthogonal table to estimate effect of various process parameters on surface finish achieved during the machining. Butt et al. recommended use of passive damper for an efficient machining of thin walled components subjected to multi axes milling operation [15]. Ram and Saravanamurugan implemented a passive solution with a viscoelastic layer sandwiched between two metal layers for vibration attenuation of turning process [16]. Finite element simulation indicates that the stability margin can be increased by 1.7 % to 25% with use of natural rubber as the viscoelastic material. Aggogeri et al. proposed damped structure with high stiffness material including aluminum foam, sandwich panels and carbon fiber reinforcement for better damping performance during turning process [17]. Lawrence et al. modulated stiffness and damping of an elastomer in vibration mitigation of a boring bar. Taguchi method was used to investigate the effect of process parameters like current density and damper location on the surface finish achieved [18]. Kajiwara et al. derived an optimum arrangement and shape of dielectric elastomer for high frequency vibration control of complex shapes [19]. Mouleeswaran and Sathishkumar modified construction of a boring bar to include particle damping with metal pieces to achieve broadband vibration attenuation [20]. Filho designed a boring tool in the form of a monolithic bar having an internal cavity wherein a dynamic absorber is placed with an elastomeric buffer arrangement [21, 22]. Howell patented a tool holder with a ring shaped vibration absorber having a provision for preload to adjust the damping [23]. Similarly a passive vibration damping ring has been proposed by Haimer and Cook for incorporation in a machine tool chuck to minimize vibrations in the circumferential direction [24, 25]. Paros et al configured a machine tool holder with two layers of elastomer damping material for driving a rotating cutting tool in order to reduce chatter vibrations [26]. Rubio-Mateos applied elastomer layer as the passive damping material at the chuck and work piece interface [27]. Use of nitrile butadiene as the damping element resulted in increase of cutting velocity and feed in comparison to that of the conventional arrangement.

Active vibration attenuation systems generally uses a piezoelectric actuator for application of force and an accelerometer located close to the cutting tool, that measures the self-generated vibrations. Aggogeri et al. designed piezo-based anti-vibration mount for micro-milling operation to reduce chatter vibrations [28]. Authors have presented a theoretical model with a control algorithm and have predicted up to 58% reduction in the vibration
amplitude. Matsubara et al. installed a piezoelectric actuator and an inductor-resistance circuit to achieve damping for machine tool chatter reduction [29]. Elastomeric springs incorporated in a tapping tool holder improved cutting speed, thread surface quality and tool life [30]. Zaeh et al. automated identification of machine dynamics for auto-tuning of an active vibration controller for chatter attenuation [31]. Tootoonchi and Gholami achieved significant reduction in the machine tool vibration amplitude by tuning the conventional passive shock absorber into a marginally stable resonator [32]. Parus et al. integrated high stiffness parts, low stiffness elements and an actively controlled piezoelectric block for milling machine vibration attenuation [33]. Authors have used closed loop control with a displacement sensor and ensured 100% increase in the depth of cut for the given cutting speed. Munoa et al. developed a biaxial active actuator and demonstrated few possible control strategies to attenuate chatter vibrations of a heavy duty milling machine [34]. Kishore et al. implemented a controllable damper in machine tool to achieve improvement in surface roughness by 22.2% and reduce tool wear by 20% [35]. Okwudire and Lee investigated two dimensional dynamics of a machine tool to determine optimal location of the vibration isolators and ensure 500% improvement in vibration amplitude [36]. Kalinski and Glewski proposed instant modulation in spindle speed with a non-linear control to reduce vibration level in an end milling operation [37]. Zou et al. incorporated four dual chamber air springs at the corners of ultra-precision machine tool support [38]. In case of vibrations, compressed air was transmitted within the four chambers to achieve auto leveling and vibration attenuation effect. Mani and Senthilkumar utilized temperature dependent elastic property of shape memory alloys in order to tune the dynamic absorber stiffness according to the forcing frequency [39]. Siddhpura and Paurobally investigated theoretical relation between chatter vibration and tool wear for estimation of the tool life [40]. Yang et al. investigated negative real part of the frequency response function at the tool and work piece interface [41]. Authors have designed multiple Tuned Mass Dampers (TMD) that need more accurate tuning of stiffness and natural frequency but exhibit better damping performance than that of single TMD. Long et al. implemented robust mixed sensitivity method in feedback controller synthesis for active vibration control of peripheral milling operation [42]. Yang et al. (2015) investigated multi TMD possessing rotational and translational degree of freedom for machine tool vibration control [43]. Implementation of the system on work piece fixture with a single dominant mode exhibited 100% increase in the critical depth of cut.

Compliant mechanisms are capable to transmit larger force and sustain harsh conditions with better durability. These can be designed with the required characteristic of joint stiffness to achieve either force or displacement amplification. Chen and Wang performed topology optimization and geometric evolution to characterize stiffness of a compliant mechanism [44]. Wang and Chen used linear elastic theory representing stiffness matrix to determine intrinsic stiffness of a compliant mechanism [45]. Finite element analysis along with experimental results demonstrated that combination of displacement amplification and higher natural frequency can be achieved in the mechanism. Pederson and Seshia derived configuration of compliant mechanism to deliver the required force transmission characteristics [46]. Bhaskar et al. incorporated displacement amplifying compliant mechanism in an accelerometer based on Hall Effect sensor [47]. Dynamic force interaction can be favorably modified to reduce mass of the dissipative element, with use of velocity amplification mechanism [48].

Macko et al. investigated evaluated efficiency of grinding process by evaluating various parameters including sample alignment and blade geometry changes by following experimental approach [49, 50]. Numerical simulation tools including computational fluid dynamics and Solid Works Flow Simulation have been used for productivity improvements.
Computer assisted simulations can be used for design of mechanical systems to suit compact and desired kinematic performance [52, 53]. Requirements of higher sensitivity to dynamic requirements of the cutting tools make the active vibration control costly and complex. Passive vibration isolators reported in the literature for boring/turning have been designed with use elastomer layer at the interface of tool and machine structure. However, using the elastomer layer affects both stiffness and damping of the isolator. Using a thick layer reduces the isolator stiffness, resulting in higher vibration amplitude leading to inferior surface finish. On the other hand, thin layer of the elastomer does not make a significant increase in the damping of the isolator. Therefore, the ideal solution for passive vibration isolator will require higher vertical stiffness and increased damping. In the proposed work, Flexible Mechanism based Vibration Absorber (FMVA) with velocity amplification has been proposed for passive solution of vibration isolator that can be used for boring/turning operations. The conceived vibration absorber has higher vertical stiffness along with greater effective damping. The conflicting characteristics of higher stiffness and damping have been achieved by operating the dissipative element (i.e. damper) with higher relative velocity.

## 2 Design of the FMVA

Proposed design of FMVA is illustrated in Figure 1 and the arrangement in the machine tool is shown in Figure 2. FMVA comprises of rigid links, flexible joints and an elastomer as the energy dissipating (damping) element, where vertical stiffness is contributed by rotational stiffness of the flexible joints.

![Fig. 1. Design of FMVA.](image)

FMVA has been designed for vibration attenuation in a machine tool suitable for boring operation. The cutting tool has been fastened to the link 1 whereas link 4 is fastened to the tool holder of the machine. Intermediate links 2 and 3 are connected to the damping element. It can be observed that the linkage mechanism has been designed to attenuate the vertical vibrations with the damping element operated in horizontal direction. The arrangement ensures that the damping element operates with amplified relative velocity than that of the vertical relative velocity of Link 1, resulting in lower force transmissibility.
FMVA has been designed for vibration control of boring operation where the resonance frequency of the machine is about 800-850 Hz and dominant forcing frequencies are within 590-700 Hz. Further detailed parameters of FMVA are given in Table 1.

![Fig. 2. Arrangement of FMEADS in the machine tool.](image)

<table>
<thead>
<tr>
<th>Table 1. Design parameters of FMVA.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Parameter</strong></td>
</tr>
<tr>
<td>Overall dimension</td>
</tr>
<tr>
<td>Flexible link length</td>
</tr>
<tr>
<td>Flexible link thickness</td>
</tr>
<tr>
<td>Material</td>
</tr>
</tbody>
</table>

### 3 Numerical simulation

#### 3.1 Frequency analysis of the prototype

Pseudo Rigid Body Method (PRBM) and Finite Element Method (FEM) have been used for evaluation of frequency response and force transmissibility of FEVA. PRBM approach is used during preliminary analysis and involved rigid elements connected together with rotational stiffness and damping. On the other hand, FEM involves flexible links which are connecting together lumped masses.

Stiffness of the as per PRBM approach, stiffness at the rotational joint is calculated as;

\[
K_i = 2\gamma K_\theta \frac{EI}{l_c}
\]

(1)

where, \(\gamma\) : constant depending on characteristic pivot

\(K_\theta\) : stiffness coefficient
\( E \): modulus of elasticity of the material
\( I \): moment of inertia of the complaint joint cross section
\( l_c \): length of the complaint joint

Figure 3 shows PRBM and FEM simulation results for resonance frequency of FMVA, where the two methods exhibit close agreement with a difference of 8.7%.

3.2 Force transmissibility analysis

Followed by the frequency analysis for natural frequency, simulations have been performed to estimate force transmissibility within the frequency range. PRBM and FEM approach have been used for the simulations with the results shown in Figure 4, which indicates that FMVA will exhibit reduced force transmissibility.
4 Results and discussion

Simulations have been performed to estimate effect of flexible link dimensions on natural frequency and the results are plotted in Figure 5. It can be noted that the force transmissibility is maximum near the resonance frequency, which increases with the joint thickness.

![Figure 5. Effect of flexible joint thickness on resonance frequency.](image)

Machining quality is affected by amplitude of vibration of the cutting tool which has to be lower in order to ensure chatter free operation. The numerical model of FMVA has been used to determine the displacement of the cutting tool and the same has been compared with the values reported in the literature. Li et al. have experimentally measured deflection of the cutting tool during boring operation which varies from 0.0216 mm to 0.0396 mm [54]. The simulation results indicate that the maximum deflection of the cutting tool with use of FMVA, will be within 0.001 mm to 0.0025 mm when the natural frequency has been ensured at 820 Hz. On the other hand if the natural frequency is ensured at 500 Hz, the force transmissibility varies from 0.6 to 1.4 within the frequency range of 150 Hz - 350 Hz with maximum deflection of the cutting tool within 0.02 mm - 0.03 mm.

Passive vibration isolators used for vibration control use application of a damping layer at the interface of cutting tool and the tool holder, as has been illustrated in Figure 6. Simulations have been performed to compute force transmissibility with use of the damping layer and the results have been compared with that of the FMVA in Figure 7. It can be noted that FMVA will exhibit better vibration isolation within the frequency range of 0-700 Hz with about 53% reduction in the transmitted force. However, the force transmissibility will be higher for FMVA during the frequency range of 700-1000 Hz. Simulation results in Figure 7 also indicate that FMVA will require lower volume and mass of the dissipative element in comparison to conventional arrangement.

In order to overcome the limitation of FMVA, integration of the flexible mechanism and damping layer as illustrated in Figure 8 has been proposed. It comprises of a damping element mounted in the flexible mechanism, which operates in horizontal direction. This damping element located in the mechanism will ensure better vibration isolation within the frequency range of 0-700 Hz. On the other hand the damping layer (attached to Link 1 and Link 4) which will act in vertical direction and ensure better vibration isolation above 700 Hz. The gap between the two damping elements as illustrated in Figure 8 will be selected according to maximum displacement of the cutting tool relative to the tool holder.
Fig. 6. Application of damping layer in the conventional vibration isolator.

Fig. 7. Force transmissibility comparison of FMVA and damping layer.

Fig. 8. Proposed vibration isolator with FMVA and damping layer.
5 Conclusion

Novel design of vibration isolator comprising of flexible mechanism and passive damping element has been presented in the paper. Damping element is operated with amplified relative velocity than that of the cutting tool to ensure reduction in force transmissibility up to 53% than that of the conventional methods of vibration isolation with a damping layer. Matlab simulations have been performed with PRBM and FEA approach to calculate the force transmissibility and cutting tool deflection. Results indicate that the proposed solution will give lower force transmissibility and tool deflection with the frequency range up to 700 Hz. Finally a modified design of FMVA has been proposed with two damping elements acting in vertical as well as horizontal direction to ensure better vibration isolation within the frequency range up to 1100 Hz.

References

34. Munoa et al. (2013).


