CIRP Annals - Manufacturing Technology 65 (2016) 389–392

Contents lists available at ScienceDirect

CIRP Annals - Manufacturing Technology

journal homepage: http://ees.elsevier.com/cirp/default.asp

Design of self-tuneable mass damper for modular fixturing systems

Jokin Munoa (2)[a,c,*], Alex Iglesias[a], Aitor Olarra[b], Zoltan Dombovari[c], Mikel Zatarain (1)[a], Gabor Stepan (2)[c]

[a] IK4-Ideko, Dynamics and Control, Elgoibar, Basque Country, Spain
[b] IK4-Tekniker, Eibar, Basque Country, Spain
[c] Department of Applied Mechanics, Budapest University of Technology and Economics, Budapest, Hungary

ARTICLE INFO

Keywords:
Milling
Chatter
Damping

ABSTRACT

Enhancing dynamic characteristics of fixtures for large workpieces is essential to assure chatter free machining of heavy-duty milling operations. Variable stiffness tuned mass dampers (VSTMD) can effectively improve the dynamic stiffness of modular fixtures by changing their dynamic characteristics. The theory of a new VSTMD concept is presented. Their realisable optimal tuning is determined and the results are compared to the standard constant stiffness TMDs. By means of the developed automatic tuning procedure, stiffness is varied via a rotary spring, while damping is provided by eddy currents. The prototype and the effectiveness of the concept are experimentally validated by heavy-duty milling tests on a modular fixture.

1. Introduction

In heavy-duty machining operations, the selection of proper workholding equipment is one of the most important technical aspects to consider for an optimum process performance. Large machined parts usually require a complex custom fixturing system designed especially for very small production batches. This requirement is usually covered by modular workholding systems based on standard elements that can be combined to hold parts of different shapes and sizes to machine.

Workholding equipment is usually designed just to fulfil requirements in terms of geometry and/or clamping forces, whereas the importance of its dynamic flexibility is usually underrated. In roughing operations of large workpieces, however, process instability often occurs due to the dynamic flexibility of the workholding system (see Fig. 1a). For this reason, the adequate dynamic performance of the workholding is crucial for an optimum machining performance. Many solutions have been engineered to improve the dynamic stiffness of the machine tool, the spindle and/or the implemented cutting tools.

Clamping fixture dynamic design has recently come into the spotlight. Several approaches can be followed for this purpose, from active control solutions [1–3] to conventional passive damping techniques, all aiming to improve the dynamic performance of clamping fixtures. As an intermediate option, tuned mass damper (TMD) technology is regarded as a proper solution for workholding vibration attenuation [4–6].

Active techniques have the advantage of versatility, since they adapt to changing conditions but they are usually rather complex and costly for conventional processes. On the other hand, TMDs are simple and affordable in most cases but they are not effective enough when the frequency of the critical mode changes.

In this context, Slavicek and Bollinger proposed the application of variable stiffness tuned mass dampers (VSTMD) for which the natural frequency and damping of the system can be varied [7]. They changed the tuning of the damper squeezing an elastomeric O-ring, creating a relation between the preload of the ring and the tuning frequency. However, the damping and

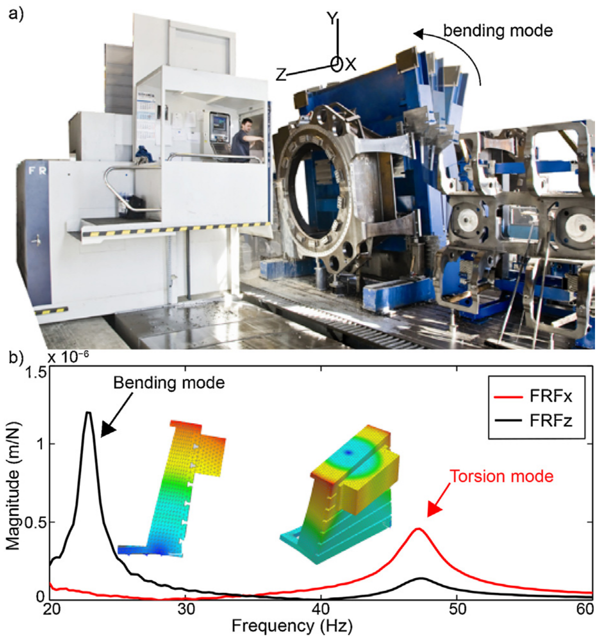


Fig. 1. Critical vibration modes of vertical workholding equipment.

* Corresponding author.
E-mail address: jmunoa@ideko.es (J. Munoa).

http://dx.doi.org/10.1016/j.cirp.2016.04.112
0007-8506/© 2016

stiffness were totally coupled [7,8]. Seto [9] solved this problem changing the connection point between the moving mass in a cantilever beam. The viscous damping was provided in parallel by means of fluids or magnetically using eddy current effect.

In this study, the classic VSTMD approach is upgraded proposing an automatic optimal tuning of the damper at any cutting condition. This way, the developed VSTMD presents the same performance as an active solution, while it keeps the simplicity and moderate price of the classical TMDs.

First, the possible chatter problems on large-scale modular fixtures are briefly summarised. Then a new frequency dependent tuning criterion is derived for VSTMD. The issues of mechanical design and the iterative algorithm for the optimal tuning of the VSTMD are followed by the experimental validation of the prototype. The results open the way for a new generation of self-tuneable VSTMDs implemented on large clamping fixtures based on several modular components where the VSTMD is another module.

2. Chatter problem in modular clamping fixtures

The critical modes related to workholding equipment for large parts present mainly low frequency, clear and simply shaped vibration modes, like bending or twisting of the whole fixture and workpiece assembly (see Fig. 1b). These modes are usually dominant and show a clear frequency separation with respect to other adjacent secondary modes. In these cases, basic modal analysis techniques can describe the essential dynamic behaviour of workholding systems with 2–3 essential modes only.

One of the most effective techniques to eliminate chatter problems in workholding systems is to enhance the damping of the modes which are related to chatter vibrations. This can be achieved by the addition of TMDs to the workholding system. In these cases, the workholding is assumed to have a single dominant mode in the frequency range without coupling between further modes, and consequently, the well-known analytical formula for the axial depth of cut at the stability limit can be considered [10,11]:

$$a(\omega) = \frac{-2\pi}{K_t Z \beta_0 Re(H(\omega))}, \quad (1)$$

where K_t is the tangential cutting coefficient, Z is the number of teeth of the cutting tool, β_0 is the average value of the directional factor and $H(\omega)$ is the frequency response function (FRF).

Eq. (1) shows that the limit depth of cut can be increased in the same proportion as the real part of the FRF is maximised or minimised (depending on the sign of β_0).

The modular nature and varying mass of the part to be machined result in different possible natural frequencies and related chatter frequencies. Therefore, a wide tuning frequency range is required to solve the possible chatter problem.

3. Variable stiffness tuned mass damper (VSTMD)

3.1. Tuning criteria for a TMD

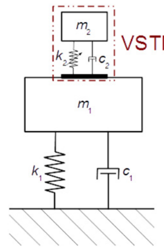
A TMD is an inertial mass added to the system in question. The mass is connected via a linear spring of stiffness k_2 and damping c_2. The values of these parameters are tuned to damp the critical mode of the original system that may produce chatter.

The most popular tuning strategies of TMDs are based on the classical analytical expressions developed by Den Hartog [12] and Sims [6]. The tuning requirements for chatter suppression in machining processes may differ from Den Hartog's optimum. Sims [6] used the real part $Re(H(\omega))$ instead of the magnitude $|H(\omega)|$ of the FRF for the tuning optimisation. This is done in order to maximise the achievable depth of cut a within the classical stability limit given in Eq. (1).

For chatter suppression, Sims' values provide optimal solution. The dimensionless parameters presented in Table 1 are defined in

Table 1
Dimensionless dynamic parameters.

Tuning (f): Mass damper natural frequency to original system natural frequency ratio	$f = \frac{\omega_2}{\omega_1}$
Mass ratio (μ): Mass ratio between the damper and the original system	$\mu = \frac{m_2}{m_1}$
Dimensionless freq. (g): Excitation frequency over the natural frequency of the system	$g = \frac{\omega}{\omega_1}$
Damping ratio (χ): Damping of VSTMD to Den Hartog critical damping [12]	$\chi = \frac{c_2}{2m_2\omega_1}$



order to simplify the tuning equations. The real part of the system formed by the critical mode to be damped and the additional TMD can be expressed by these dimensionless parameters, assuming a dominant mode and a negligible damping of the original system ($c_1 = 0$) [6]:

$$Re(H(\omega)) = \frac{1}{k_1} \times \frac{(f^2-g^2)((1-g^2)(f^2-g^2)-\mu g^2 f^2)+4\chi^2 g^2(1-g^2-\mu g^2)}{((1-g^2)(f^2-g^2)-\mu g^2 f^2)^2+4\chi^2 g^2(1-g^2-\mu g^2)^2}. \quad (2)$$

Based on this expression, and considering all the possible tuning parameters for a TMD, Sims found three invariant frequency points, so-called locked frequencies. The optimum tuning frequency is found when two of these locked frequencies present the same amplitude in the real part and are local maximum or minimum depending on the sign of β_0:

$$f_{S,\pm} = \sqrt{\frac{\mu+2\pm\sqrt{2\mu+\mu^2}}{2(1+\mu)^2}}, \quad (3)$$

where $f_{S,+}$ and $f_{S,-}$ stand for $\beta_0 > 0$ and $\beta_0 < 0$, respectively. These fixed parameters assure maximum stability when the system is limited by one dominant mode with negligible damping.

3.2. Optimal tuning of VSTMD

In this work, a new TMD concept is presented. Stiffness variation capability is added, hence, a new optimal tuning definition is possible. While Den Hartog and Sims set a constant optimum tuning valid for every condition, VSTMD allows a variable tuning which depends on the frequency of interest, which could be the chatter frequency or the spindle speed of machining.

Thus, the search of the local optimum for each frequency improves the overall optimisation. Eq. (2) is derived with respect to the frequency tuning f to find the local maximum or minimum, and

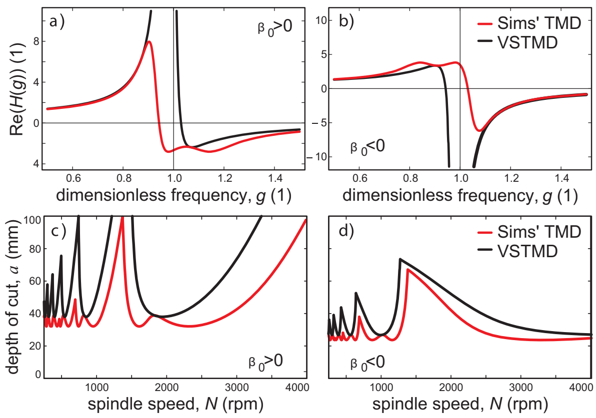


Fig. 2. Comparison of real parts (a) and (b) and stability lobes (c) and (d) of the optimally tuned frequency response function versus Sims' tuning for $\beta_0 > 0$ and $\beta_0 < 0$.