International Journal of Machine Tools & Manufacture 107 (2016) 41–49

Contents lists available at ScienceDirect

International Journal of Machine Tools & Manufacture

journal homepage: www.elsevier.com/locate/ijmactool

Suppression of the time-varying vibration of ball screws induced from the continuous movement of the nut using multiple tuned mass dampers

Min Wang [a,b,*], Tao Zan [a,b], Xiangsheng Gao [a,b], Songwei Li [a,b]

[a] Advanced Manufacturing Technology Lab, School of Mechanical and Applied Electronics Engineering, Beijing University of Technology, Beijing 100124, China
[b] Beijing Key Lab of Electrical Discharge Machining Technology, Beijing 100124, China

ARTICLE INFO

Article history:
Received 12 January 2016
Received in revised form
9 May 2016
Accepted 12 May 2016
Available online 13 May 2016

Keywords:
Ball screw
Multiple tuned mass dampers
Time varying
Vibration control

ABSTRACT

Considering the time-varying vibration characteristics of the ball screw caused by the continuous movement of the nut, this paper takes the form of a hollow screw shaft structure containing multiple tuned mass dampers (MTMD) to achieve the lateral multi-mode vibration control of the screw shaft. The screw shaft is modeled as an Euler–Bernoulli beam with elastic supports at both ends and the position of the nut. Each tuned mass damper (TMD) is connected to the screw shaft via an elastic spring and a viscous damping element. After establishment of the lateral dynamic model of the screw shaft which taking into account the changing positon of the nut and the multiple resonant responses of the shaft, the optimum design parameters of each TMD can be determined using a numerical optimization algorithm based on the mode summation method. The multiple resonant responses of the screw shaft installed with the optimally designed MTMD are analyzed to demonstrate the robust design of the MTMD for the lateral time-varying vibration control of the screw shaft. Theoretical studies and experimental results show that the proposed design method of the MTMD can remarkably improve the lateral dynamic stiffness of the screw shaft.

© 2016 Elsevier Ltd. All rights reserved.

1. Introduction

A feed ball screw mechanism is often used in precision machine tools to accurately move worktable by a predetermined amount. However, the operation of a ball screw with a long stroke at high rotational speed has been considered difficult. As these kinds of ball screws have long and narrow shafts supported at both ends, the rigidity in radial direction is small and the screw shaft is very liable to resonate dramatically during operation at high speed. It has been found that the vibration of the screw shaft leads to noise of the entire mechanical system of machine tools. At the same time, the severe vibration of the screw shaft also adversely affects both the positioning accuracy and service life of the ball screw [1,2]. Therefore, it is very important to suppress the lateral vibration of the screw shaft.

The ball screw mechanism is a closed complex drive mechanism consisting of three main components: screw shaft, nut and balls. During the operation of the ball screw, vibratory forces are nearly impossible to be eliminated which are produced by rolling or circulation of balls between the screw shaft and the nut [3]. These vibratory forces become greater along with the increasing of the operation speed. These forces always act on the screw shaft and induce the resonant vibration of the screw shaft. In addition, the resonance frequency exists over a considerable range because of the continuous movement of the nut along the screw shaft [4]. The critical rotation speed and resonance frequency of the screw shaft are physically determined by its dimensions and supporting stiffness. However, it is difficult to improve the supporting stiffness and suppress the resonant vibration of the screw shaft because the nut moves along the screw shaft and the interference cannot be avoided when supporting the intermediate portion of the screw shaft.

Generally, structural vibration can be effectively controlled by using passive dampers or by using active vibration control techniques. These two methods have been successfully used to suppress the screw shaft's axial and torsional vibration [5–8]. However, the lateral vibration of the screw shaft is difficult to be dampened using the controller or a passive damper because of the above mentioned time varying characteristics of the resonance frequencies. On the other hand, the ball screws used in machine

* Corresponding author at: Advanced Manufacturing Technology Lab, School of Mechanical and Applied Electronics Engineering, Beijing University of Technology, Beijing 100124, China.
E-mail address: wangmin0821@hotmail.com (M. Wang).

http://dx.doi.org/10.1016/j.ijmachtools.2016.05.003
0890-6955/© 2016 Elsevier Ltd. All rights reserved.

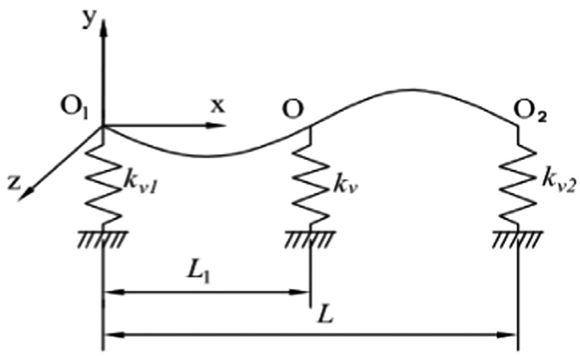

Fig. 1. Model of the screw shaft.

tools have been and will continue to be required to achieve higher operational speed. This makes it possible that there exist more than one main resonant peak, namely a multi-mode vibration problem. The conventional passive dampers are not suitable to suppress a multi-mode vibration.

The most widely used passive damper in practice is tuned mass damper (TMD), which consists of an additional mass-spring-damper sub-system, and needs accurately tuning of its natural frequency and damping ratio to match the main structure for vibration control [9–10]. However, errors in identifying the structural frequency or in manufacturing the TMD may lead to detuning of the TMD from the frequency of the main system. For improving the robust of the TMD, the multiple TMDs (MTMD) system was proposed, of which each TMD is tuned to a specific frequency of the main system [11]. For suppressing single-mode vibration, the MTMD was proven to have better robust performance and higher effectiveness compared to a single TMD [12]. On the other hand, it has also been demonstrated that multiple dampers can be used to damp a multi-mode vibration in milling of thin walled parts [13,14].

The purpose of this paper is to investigate the feasibility and analyze the performance of the MTMD for reducing the lateral time-varying vibration of the ball screw. In the present study, multiple TMDs are incorporated in a hollow screw shaft to suppress its lateral vibration. For calculating the optimal parameters of the MTMD, a dynamic model of the screw shaft containing multiple TMDs is established which takes into account the changing positon of the screw nut and the multiple resonance problem. Based on this dynamic model, the analytical solution of the frequency response function (FRF) of the screw shaft can be derived. Considering the special nature of the lateral vibration of the screw shaft, the optimal design parameters of the MTMD are those that minimize the magnitude of the amplitude of the FRFs of the screw shaft at the position of the nut. Assuming a given total mass and damper number of the MTMD, the optimal design parameters of each TMD can be calculated by employing a numerical optimization algorism based on the mode summation method. Based on the analysis of the results from numerical simulations, the effects of the total mass and damper number on the screw shaft vibration control were investigated. The simulation results show that more than tenfold improvement in the lateral dynamic rigidity of the screw shaft at the positon of the nut can be obtained by using the proposed method. At the same time, the multiple resonant responses of the screw shaft installed with the optimally designed MTMDs were analyzed to demonstrate the robust design of the MTMD for the lateral vibration control of the screw shaft. Finally, an MTMD was designed and installed in a hollow screw shaft. The experimental results show that the proposed MTMD design method is practical and feasible for the lateral time-varying vibration control of ball screws.

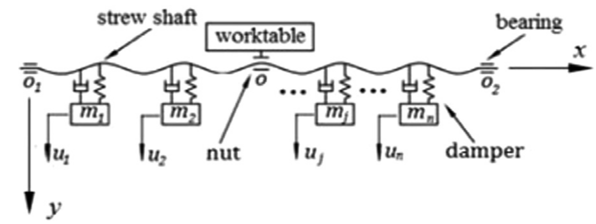

Fig. 2. Vibration system of the ball screw with MTMD.

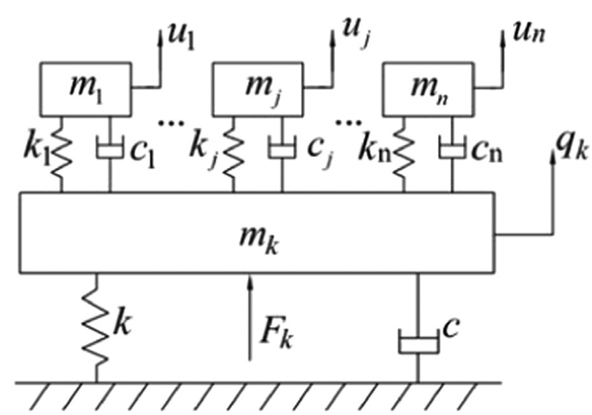

Fig. 3. Dynamic model of the *k*th mode.

2. Modeling of the screw shaft with the MTMD

2.1. Lateral vibration of the screw shaft

Considering the bearing stiffness and the contact stiffness between the screw shaft and the nut, the screw shaft is modeled as an Euler–Bernoulli beam with elastic supports at both ends and the position of the nut, as shown in the Fig. 1. In Fig. 1, k_{v1} and k_{v2} represent the radial support stiffness of the left and right bearings, k_v represent the radial contact stiffness between the nut and screw shaft, L is the length of the screw shaft, and L_1 is the position of the nut. The vibration shape functions at any mode for the left and right of the nut can be given as

$$\phi_{k,l}(x) = C_{k,1}ch\lambda_k x + C_{k,2}sh\lambda_k x + C_{k,3}\cos\lambda_k x + C_{k,4}\sin\lambda_k x \tag{1}$$

$$\phi_{k,r}(x) = C_{k,5}ch\lambda_k x + C_{k,6}sh\lambda_k x + C_{k,7}\cos\lambda_k x + C_{k,8}\sin\lambda_k x \tag{2}$$

where $C_{k,1}$–$C_{k,8}$ are the vibration shape coefficients associated with the *k*th mode, and the *k*th mode's natural frequency are defined as

$$\omega_k = \lambda_k^2\sqrt{\frac{EI}{A\rho}} \tag{3}$$

where EI, ρ and A are the flexural rigidity, the density and beam cross-sectional area, respectively.

Regarding this fact that both ends and the position of the nut are with elastic support, the boundary condition and continuity condition of the screw shaft can be given as

$$\frac{\partial^2\phi_{k,l}(0)}{\partial x^2} = 0;\ EI\frac{\partial^3\phi_{k,l}(0)}{\partial x^3} = -k_{v1}\phi_{k,l}(0) \tag{4}$$

$$\frac{\partial^2\phi_{k,l}(L)}{\partial x^2} = 0;\ EI\frac{\partial^3\phi_{k,r}(L)}{\partial x^3} = k_{v2}\phi_{k,r}(L) \tag{5}$$

$$\phi_{k,l}(L_1) = \phi_{k,r}(L_1);\ \frac{\partial^2\phi_{k,l}(L_1)}{\partial x^2} = \frac{\partial^2\phi_{k,r}(L_1)}{\partial x^2} \tag{6}$$

Download English Version:

https://daneshyari.com/en/article/781353

Download Persian Version:

https://daneshyari.com/article/781353

Daneshyari.com