International Journal of Machine Tools & Manufacture 90 (2015) 50–59

Contents lists available at ScienceDirect

International Journal of Machine Tools & Manufacture

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

A method for stiffness tuning of machine tool supports considering contact stiffness

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ARTICLE INFO

Article history:
Received 15 October 2014
Received in revised form
29 December 2014
Accepted 5 January 2015
Available online 6 January 2015

Keywords:
Machine tool support
Contact stiffness
Rocking vibration
Stiffness tuning

ABSTRACT

A methodology for tuning the stiffness of machine tool supports is described based on a stiffness model using the contact stiffness approach. Using this model, the mathematical relationship between the load of the support and its stiffness is established. The relationship is separated into three regions. When the total stiffness of all supports is maximized, the load must be tuned so that the stiffness–support load relationship is in the critical region, whereby the contact stiffness is slightly larger than the bulk stiffness. Correspondingly, a placement method of supports is proposed that increases their stiffness without anchor bolts. The effectiveness of the proposed method is verified in two experiments. In the first experiment, the natural frequency of a small machine tool prototype is compared for several placements of three supports. The lowest natural frequency of the machine tool under the proposed placement scheme is maximized. In the second experiment, the proposed method is applied to increase the lowest natural frequency of a horizontal milling machine. The lowest natural frequency with a distinct arrangement of three supports is increased by 15–55%, compared to other popular placements of these three supports. The experimental results show that the proposed placement method is effective for enhancing the stiffness of machine tool supports.

1. Introduction

Vibration of machine tools must be suppressed to reduce dynamic motion errors in highly productive machining. Ground vibrations transmitted from the floor and drive disturbance vibrations caused by the driving force in feed drives represent the major vibration sources. Invariably, these vibrations excite vibration modes at low frequencies. In particular, rocking vibrations typically represent the lowest two vibration modes of machine tools [1,2]. Correspondingly, the dynamic property of the rocking vibration is greatly influenced by the stiffness of supports [2]. Therefore, it is critically important that the stiffness of the supports is properly designed and tuned.

Many studies have previously been reported on the design and the computational evaluation of machine tools. The static and dynamic behaviors of the machine have been evaluated by multi rigid body analysis and finite element analysis (FEA) [3–6]. The optimization of the machine tool design has been studied using these analysis schemes [7–13]. In addition, the fixture design of the workpiece has been also optimized [14,15]. However, the design of supports has not been a popular aspect of study in spite of its significant effect on the dynamic performance of machine tools. Rivin discussed the location of supports for reducing the static deformation of the machine [16]. To reduce vibration, Okwudire et al. proposed an optimal location of vibration isolators for an ultraprecision machine tool [17]. Vibration analyses using simplified models were utilized to determine the stiffness of supports for minimizing the vibration [1,18].

However, not many machine tool manufacturers design or tune the stiffness of supports on the basis of dynamic performance evaluation of the machine. One reason is that the stiffness of supports cannot be calculated simply from design parameters, even if FEA is used in a detailed computational model. Proper boundary conditions are required to obtain the stiffness of supports, particularly because the stiffness at contact interfaces has an influence on the stiffness of supports. Although anchor bolts are sometimes used to practically increase the stiffness of supports, their effects have not been clearly explained.

To address this problem, a model based on the contact stiffness of machine tool supports has been developed in one of our previous studies [19]. In this paper, a method for tuning the stiffness of supports is proposed using the developed model. In this approach, a basic idea is first described for tuning the preload and the stiffness of supports. Then, a placement method of supports is proposed, on the basis of increasing their stiffness without the use of anchor bolts. An experimental modal analysis of a small

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http://dx.doi.org/10.1016/j.ijmachtools.2015.01.001

machine tool prototype is then conducted to verify the proposed method. Finally, the proposed method is applied to increase the lowest natural frequency of a horizontal milling machine.

2. Stiffness tuning of machine tool supports

2.1. Stiffness model of machine tool supports based on contact stiffness

In this study, the stiffness model of machine tool supports, proposed in our previously published study, is used [19]. Fig. 1 (a) shows the schematic drawing of a screw jack as an example type of a machine tool support. The stiffness of one support is modeled in three-dimensions (3D) as shown in Fig. 1(b). The stiffness along each direction, K, is obtained from the bulk stiffness and the contact stiffness connected in series, and determined using

$$\frac{1}{K} = \frac{1}{k_{ball}} + \frac{1}{k_{call}} \tag{1}$$

where k_{ball} and k_{call} are the total bulk stiffness and the total contact stiffness for one support, respectively. In an actual case, the support consists of several components and contacted interfaces. Therefore, the above total stiffness is also obtained from their stiffnesses connected in series.

The bulk stiffness can be calculated from the modulus of elasticity and the geometry of the support. The contact stiffness is obtained using the model shown in Fig. 2. Fig. 2(a) shows a schematic of an interface at the machine tool support. The preload W acts on the interface; k_{cn} and k_{ct} are the contact stiffnesses in directions normal and tangential to the interface, respectively. The stiffness values k_{ci} (i=n, t) are obtained from a series of coupled springs, spread over the interface using

$$k_{ci} = \frac{\delta k_{ci1} \delta k_{ci2} W}{(\delta k_{ci1} + \delta k_{ci2}) p_m} \tag{2}$$

where the subscript i represents the normal or the tangential direction, δk_{ci1} and δk_{ci2} are the contact stiffnesses per unit of real contact area (unit contact stiffness) of materials 1 and 2, respectively, and p_m is the lowest value of the two yield pressures associated with materials 1 and 2.

In the Hertz contact model, the relationship between the preload and the contact stiffness at one roughness asperity on the interface is non-linear and not proportional to the preload. However, in real contact of flat surfaces, the number of contacted asperity increases with the preload. Greenwood et al. have reported that the number of contacted asperity is proportional to the preload and the average size of the contacted spot is constant [20]. Therefore, the real contact area and the contact stiffness increase more sharply than those in the Hertz contact of one asperity. They describes that their discussion can be applied to rough curved surfaces as well as nominally flat surfaces. When the contacted spot is considered as the stiffness spread over the interface, the study by Greenwood et al. supports our contact stiffness model.

2.2. Stiffness tuning technique

Generally, if the materials of the support and the floor are not changed, the stiffness of the support can be tuned by either varying the bulk stiffness, the preload, the number of supports or other constitutive parts of the system. A basic idea for selecting any of these approaches is described using the stiffness model presented above.

According to Eqs. (1) and (2), the stiffness of the support depends on the preload. Fig. 3 qualitatively shows the relationship between the preload and the stiffness of one support. The relationship is calculated with two different bulk stiffness values using the model described in Subsection 2.1. In both cases, the total stiffness is positively correlated with respect to the preload.

The graph representing the total stiffness–preload relationship can be distinguished into the following three regions: the increasing region in which the stiffness increases almost linearly with the preload (because the contact stiffness value is smaller than the bulk value and dominates total stiffness); the saturation region in which the total stiffness is saturated (because the bulk stiffness value dominates the value of total stiffness); and the critical region, which represents the intermediate region between the increasing and the saturation regions. When the bulk stiffness is increased, the critical region shifts towards larger preload regions and the saturated stiffness increases.

The three regions are used to qualitatively show the efficiency for exchanging the preload to the stiffness. Therefore, the criteria for the regions can be defined using the stiffness differentiated by the preload. Fig. 4 shows the relationship between the preload and the stiffness differentiated by the preload. The efficiency is at maximum at 0 N preload. In this study, the threshold for the boundary between the increasing and the critical regions was determined as 20% of the maximum value. The threshold for the boundary between the critical and the saturation was determined as 10%. However, these thresholds can be changed because the required efficiency can be different on a case by case.

The stiffness of the support should be tuned according to the total stiffness–preload relationship shown in Fig. 3. When the total stiffness of all supports must be increased within the operating range of the increasing region, the preload should be increased by anchor bolts. On the other hand, when operating within in the saturation region, the bulk stiffness should be increased. The number of supports can also be increased within the saturation

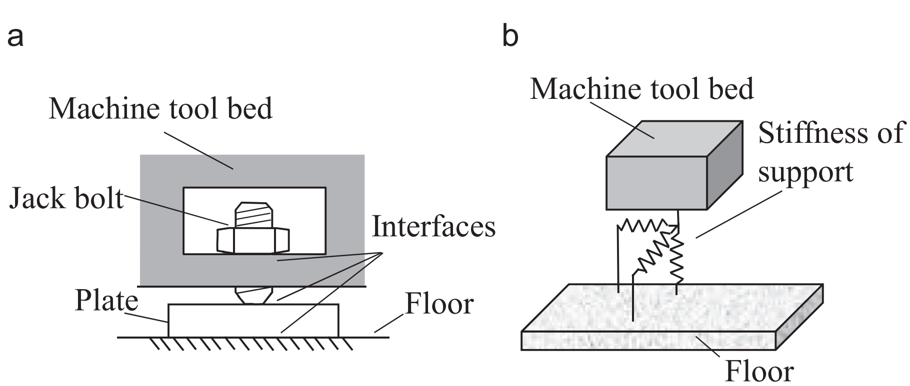

Fig. 1. Machine tool support and its model. (a) Schematic drawing of screw jack, and (b) 3D stiffness model of support.