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Damping models for machine tool components of linear axes

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ABSTRACT

To simulate the dynamic behaviour of machine tools, the stiffness, damping and inertia parameters of the structure are needed. While masses and stiffness parameters of structural parts can be obtained with a static measurement, the determination of damping parameters requires a thorough methodology. In this paper the common methodology for the identification of local damping parameters of machine components was extended by an additional step to isolate the damping of the test object more precisely. Furthermore test benches as well as the identified damping models for components of an exemplary linear axis are presented.

1. Introduction

Today's complex machining tasks demand in-depth consideration of the dynamic behaviour of machine tools. During the development phase, the calculation of the dynamic behaviour saves time and prototyping costs. To date, software and high performance computing have provided comprehensive ways to calculate frequency response functions (FRF), which are characteristic for the dynamic behaviour of the machine tool structure. However, the correct calculation of mode amplitudes is particularly difficult. This is due to the lack of knowledge about the damping properties of the machine tool structure.

A common way to simulate the FRF of a machine tool is to generate a Finite Element model and to transform the resulting equation of motion into the modal domain [1]. In a first step the system of equations is reformulated as an Eigen value problem. The resulting Eigen values represent the undamped Eigen frequencies. The corresponding Eigen vectors describe the Mode shapes of the oscillating system. Using the Eigen vectors, the stiffness and mass matrices can be diagonalized and the system is transferred into the modal domain. In this domain, a dynamical system with multiple degrees of freedom is described by several single degree of freedom oscillators. The damping can be considered by defining a diagonal damping matrix with modal damping coefficients for each of these oscillators. Such a global damping description is inherent in the specific structure and cannot be used to predict the dynamic behaviour of other structures.

For this reason the damping needs to be defined locally before the system is transferred to the modal domain. A damping matrix where the damping is defined locally cannot be diagonalized. This requires advanced methods for performing a FRF calculation. By the use of these methods local damping forces (F) can either be formulated proportional to the velocity ($\dot{x}$) Eq. (1) or proportional to the stiffness (k) Eq. (2). They are called "viscous" (d) and "structural" (η) damping in common simulation software solutions (ω: circular frequency).

$$F_{\mathrm{visc}} = d \cdot \dot{x} \tag{1}$$

$$F_{\mathrm{struct}} = k \frac{\eta}{\omega} \dot{x} \tag{2}$$

Modelling the local damping is the basic requirement to predict the dynamic behaviour of machine tools. This article describes an extended methodology to identify appropriate local damping models and their parameters for machine tool components of linear axes.

2. Identification of damping in machine tool components

The damping of machine tool components was investigated by several authors. Yoshimura [2] measured the damping of bolted joints. Dietl [3], Kraus [4] and Ophey [5] describe the damping measurement of spindle bearings, Groche [6] deals with the damping of linear guides and Meidlinger [7] measures the damping of machine mounts. There exist no investigations of ball screw nuts or ball screw bearings.

2.1. Identification of local damping models

These former investigations basically use the same method to identify the damping of test objects (comp. Fig. 1 "top"). The test bench is designed in a way that the test object is the least stiff part of the whole structure. There exist Eigen modes of the test bench where the highest relative displacement takes place in the test object itself. The damping of these Eigen modes is mainly caused by the test object and can be qualified by the damping ratio. The damping ratio of a particular Eigen mode can be identified by using the bandwidth method on a measured FRF. This method can be used for systems with linear behaviour and Eigen modes that are clearly separated [8]. The damping ratio corresponds to the global modal damping description in Section 1. Thus, the damping ratio is

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not transferable to other systems as well. By the use of a simulation model of the test bench with local damping models for the material, the bolted gaps and the test object suitable models and parameters can be identified by matching the measured to the simulated damping ratios. One problem of this method is the fact, that the particular Eigen modes are not only damped by the test object but also by the test bench itself. So it is not clear during the matching process which portion of the damping is a result of the test bench and of the test object. Wrong assumptions about the test bench damping adversely affects the identified test object's damping.

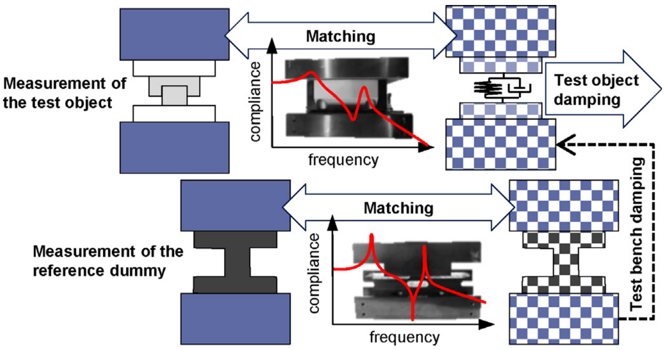

Fig. 1. Extended identification method for local damping models.

2.2. Isolation of the test object's damping

To overcome these problems this method was extended by an additional measuring step to be able to identify the damping of the test bench (comp. Fig. 1 bottom). In this step the test object is replaced by a reference dummy. The reference dummy consists of solid material, has the same interfaces, similar stiffness, but less damping than the test object. The Eigen modes and Eigen frequencies are similar to the setup with the test object. The identified damping ratios only include the test bench damping and the material damping of the reference dummy, whereas the damping of the test object is excluded. With this measurement the test bench damping can be considered more precisely in the simulation model.

To validate the identified damping models the test objects need to be measured under different dynamic boundary conditions. In a first step this can be realized by modification of the mass distribution of the test bench itself. In a second step different test objects can be combined to subsystems to verify if the identified local damping models can predict the measured behaviour.

With this method machine tool components of linear axes have been investigated regarding their damping properties. Damping models for linear guides, ball screw bearings and ball screw nuts were identified with the help of single component test benches. For validation, four linear guides were assembled into a slide system. Analogously, all ball screw components were combined to a ball screw drive. In actual studies this ball screw drive will be merged with the slide system to validate the damping models within an exemplary linear axis system.

3. Damping in linear guides

In this section the test bench for linear guides and the measuring results are presented. The investigated test objects are linear roller guides with three different preload classes (2%, 8%, 12%). To identify the production tolerance, three identical linear guides of each preload class are measured. The linear guides have an installation size of 45, corresponding to the width of the guide rail in millimetres. The linear guides were measured without any additional lubrication.

3.1. Test bench for linear guides

To investigate the damping of linear guides, a test bench has been developed which consists of two masses and is shown in Fig. 2. The first mass (a) is mounted to the guide carriage (b), the second mass (c) to the guide rail (d). Both mass centres are located in the centre of the linear guide. Adapter plates (e) enable the measurement of linear guides with different installation sizes and designs. All test bench components are designed to have square shapes to allow simple FE meshing with linear hexahedron elements. The test bench is mounted on springs (f) which cause free vibration modes with isolated translational and rotational mode shapes.

In the presented studies the test bench was excited with an impact hammer (g), while the response signal was measured with a glued acceleration sensor (h) at the adapter plate of the carriage.

The reference dummy (i) features identical interfaces to the adapter plates, so the damping of the rolling contact as well as the damping of the bolted gaps at the carriage and rail is substituted by the weakly damped reference dummy.

To illustrate the dynamic behaviour of the test bench the FRFs are shown for an excitation in both vertical and horizontal directions with a linear guide and the reference dummy.

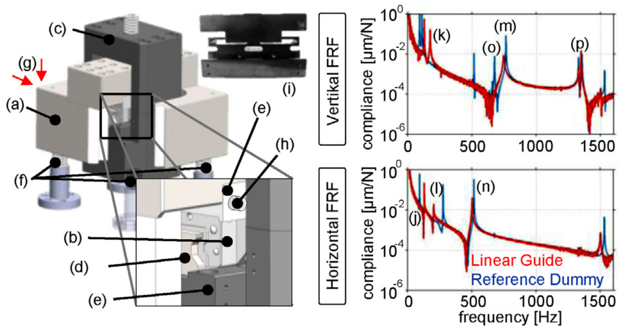

Fig. 2. Test bench for linear guides and its dynamical behaviour.

The test bench has six Eigen modes below 15 Hz, where the entire structure oscillates in its spring mounts. The positioning of the mass centres causes distinct Eigen modes where the linear guide is stressed only rotationally around its three axes (rolling (j), yawing (k) and pitching (l)) and only translationally along its vertical and horizontal axes (push/pull (m) and cross direction (n)). All these Eigen modes are clearly separated and have no superposition of rotational and translational movements. This enables a clear assignment of the damping direction during the identification process. An additional Eigen mode at 680 Hz (o) is an oscillation in the axial direction, which occurs only during the reference dummy measurement. The first Eigen modes of the test bench's structure (p) occur above 1300 Hz. At similar frequencies, the Eigen modes' amplitudes of a reference dummy measurement are substantially higher (vertically: 4–12 times; horizontally: 5–10 times).

3.2. Measuring results and identification of a damping model

Significant differences in the measured damping ratios of the investigated linear guides occur due to different preload classes. To increase the preload, larger rolling elements are used. Thereof results a slight increase in stiffness and reduced damping. In the measurements, this manifests as a higher resonance frequency and a lower damping ratio. Fig. 3 shows the frequency and damping ratio of the Eigen modes in push/pull and cross direction (comp. Fig. 2 (m,n)) of the investigated linear guides with the preload classes C2%, C8% and C12%.

In each case, three identical linear guides were studied. Fig. 3 shows the standard deviation and the coefficient of variation (standard deviation divided by the sample mean) for these