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Modelling vibration transmission in the mechanical and control system of a precision machine

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ABSTRACT

The relative vibration between tool and workpiece factors significantly to the performance of a precision machine. This paper develops a model for predicting the vibration transmission from two major excitation sources, ground vibration and fluid bearing force, to the tool and the workpiece position through the mechanical and control system of a precision machine. We synthesised the frequency response functions obtained from a finite element analysis of the machine to create transmissibility matrices that define the dynamic behaviours of the electromechanical system. The validity of the developed model was checked by comparing the measured relative vibrations to the results calculated from the measured excitations.

1. Introduction

Precision machines such as diamond cutting machines, coordinate measurement machines (CMM) and semiconductor manufacturing equipment often use fluid bearings and frictionless drive mechanisms in order to achieve fine positioning accuracy and smooth motion. One drawback of these systems, however, is that fluid bearings may cause excitations to the system when either the supply pressure fluctuates or the flow of oil or air around the bearing pads is unstable. Furthermore, the position control systems of these machines are more sensitive to disturbances because of low damping in a moving direction. Such a situation complicates achieving good position stability or low relative vibration between the tool and workpiece in precision machines, which is crucial for achieving good machined surfaces in machine tools and good repeatability for measurement machines. In general the position stability of a precision machine can be improved by designing the machine with a control system whose bandwidth is high enough to suppress given disturbances as much as possible and by lowering the level of disturbances by, for example, isolating ground vibration.

Since ground vibration is a major source of disturbance for precision machines, research has extensively investigated its isolation from precision equipment [1]. Also, hydrostatic and aerostatic bearings are key technologies for precision machines, thus their design methods are comfortably established after years of being investigated [2]. Demand for higher position stability of precision machines has led to an investigation on the micro-vibration of aerostatic bearings [3]. Several different kinds of precision machines have been studied by researchers who attest to general design guidelines for achieving high precision [4,5]. Researchers in control engineering interested in minimising the position error of a control system developed a systematic method for decomposing noise sources in position error signals, and a similar method has been implemented in the development of some precision machines [6]. However, it is not clear to what extent relative vibrations between the tool and workpiece are affected by various sources of disturbances and how machine design can reduce such disturbances.

This paper thus seeks to develop a model for predicting vibration transmission from disturbance sources to the tool and workpiece locations in a precision machine. To this end, our study considers two major disturbances: ground vibration and dynamic forces created from fluid bearings. To reach this aim, we constructed an analytical model consisting of matrices of frequency response functions (FRF) to represent the structural and servo dynamic behaviour of the machine tool. Using the computer-aided engineering (CAE) analysis method, we calculated the relative FRFs at the encoder-scale positions as well as the tool-work positions with various sources of excitation. The former FRFs were used to obtain disturbance input to the control system and to calculate the motor forces which became another source of excitation. We obtained the relative vibration at the tool-work position by combining the latter FRFs and the excitation signals. We verified the feasibility of the developed vibration transmission model by comparing the relative vibrations measured and calculated at encoders and the tool-work interface.

The developed model will be beneficial to precision machine designers. Given data of disturbance, the developed model will help to estimate the magnitude and frequency content of relative vibration between the tool and workpiece. This will allow to evaluate design variations in vibrations of a given machine during design stage, as well as provide a guideline for allowable level of disturbance in order to achieve a desired level of vibration.

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2. Vibration transmission in a precision machine

2.1. Main sources of excitations

In this study we investigated major sources of disturbance and their influence on machine vibration by using the precision machine illustrated in Fig. 1. The machine's main purpose is creating fine linear grooves on master moulds for optical components by using single crystal diamond tools. It has three main moving axes in *X*, *Y*, and *Z* directions, which are supported by hydrostatic bearings, driven by linear motors, and position controlled using error signals detected at high resolution linear encoders [7].

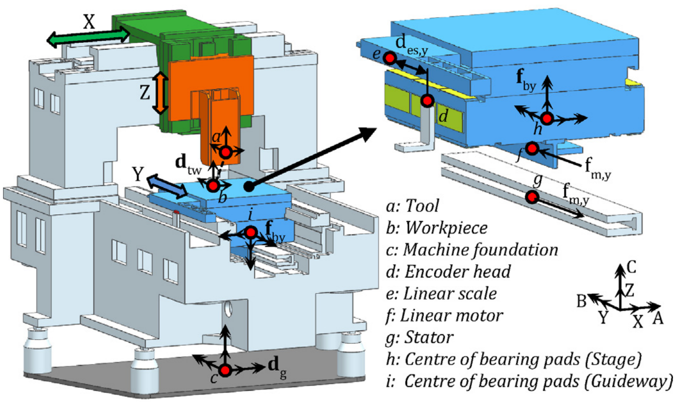

Fig. 1. Test precision machine and primary vibration sources.

This study focuses on relative vibration between the tool and the workpiece with all axis positions on hold while hydrostatic bearings are pressurised and position control is activated, since this baseline vibration exists even while axes are moving. Regarding this situation, this study considers three types of excitations as primary potential sources of vibration as illustrated in Fig. 1:

1. Ground vibration ($\mathbf{d}_g$): usually this can be characterised as a random vibrational motion of the ground with six degrees of freedom (DOF) over a broad frequency range. It can be easily identified using seismic accelerometers and is expressed as a vector with six power spectral density (PSD) components, three translational motions in the *X*, *Y* and *Z* directions and three rotational motions around the *A*, *B* and *C* axes applied at the machine's centre on the ground.
2. Bearing force ($\mathbf{f}_{bi}$): this is created at the hydrostatic bearing pads usually due to continuous fluctuation in supply pressure from its set value or instability in the supply line and bearing pads. Bearing force can also be characterised as a broadband random excitation. In this study, it has been simplified as a vector, $\mathbf{f}_{bi}$, with three force and three moment PSD components applied at the centre of bearing pad set of machine axis *i*. Both the stage and guideway side of a bearing pad set experience the same amplitude of forces and moments in opposite directions.
3. Motor force ($\mathbf{f}_m$): this is determined by the position control system based on the position error detected at the encoders so that the position command can be followed. Motor force helps to suppress relative vibration between the encoder head and the scale up to the control bandwidth, yet at once serves as an excitation to the stationary and moving part of the axis.

2.2. Transmission of vibrations in the mechanical and control system

A combination of all excitations passes through the mechanical and control system of the machine in a different manner. Fig. 2 illustrates the vibration transmission model developed in this study. The vibration transmission characteristics of the mechanical system from one location to another, can be described by frequency response functions (FRF). For example, the dynamic characteristics between ground vibration, $\mathbf{d}_g$, and relative vibration at the tool and work interface, $\mathbf{d}_{tw}$, can be described with a set of FRFs, $\mathbf{G}_{tw,g}$, in Fig. 2. These functions can be obtained from CAE modal analysis under the assumption that the machine can be modelled as a linear system.

The vibration transmission characteristics of each axis in the control direction can be described by a closed-loop response function to disturbance input, which is called the sensitivity function. However, the application of the sensitivity function on the vibration transmission model is not straightforward because the locations of position feedback and force transmission are not identical. The following steps were taken in this study to address this issue:

1. Open-loop frequency responses at the encoders due to excitations, $\mathbf{d}_{eso}$, were calculated using the PSDs of excitations and the FRFs from the excitation sources to the encoders, $\mathbf{G}_{es}$.
2. Given the relative vibrations at the encoders, $\mathbf{d}_{eso}$, the motor forces, $\mathbf{f}_m$, generated by position controllers to suppress the disturbance were calculated in frequency domain.
3. The relative vibration at the tool and work interface due to the motor forces were obtained by combining the motor force signals and the relative FRFs from motors to the tool and workpiece position, $\mathbf{G}_{tw,m}$, then it was combined with the vibration transmitted through the mechanical system to obtain the resulting vibration, $\mathbf{d}_{tw}$.

2.3. Mathematical formulation of the vibration transmission model

This section explains how the relative vibration between the tool and workpiece, $\mathbf{d}_{tw}(s)$, and the relative vibration between the encoder head and the scale, $\mathbf{d}_{es}(s)$, can be mathematically formulated based on the generalised diagram illustrated in Fig. 2. Vector $\mathbf{d}_{tw}(s)$ has three PSDs of vibrations in *X*, *Y* and *Z* directions, $[\mathrm{d}_{tw,x}(s), \mathrm{d}_{tw,y}(s), \mathrm{d}_{tw,z}(s)]$, as components. Similarly, $\mathbf{d}_{es}(s)$, $\mathbf{d}_{eso}(s)$, $\mathbf{d}_g(s)$, $\mathbf{f}_{bi}(s)$, and $\mathbf{f}_m(s)$ are regarded as vectors of PSD functions, $[\mathrm{d}_{es,i}(s)]_{i=x,y,z}$, $[\mathrm{d}_{eso,i}(s)]_{i=x,y,z}$, $[\mathrm{d}_{g,j}(s)]_{j=x,y,z,a,b,c}$, $[\mathrm{f}_{bi,j}(s)]_{j=x,y,z,a,b,c}$ and $[\mathrm{f}_{m,i}(s)]_{i=x,y,z}$ respectively, where index *i* and *j* denotes machine axis and six degrees of freedom of movement, respectively.

For random excitation input signals, the PSD of output signal from a frequency response function of a linear system, $\mathbf{G}(s)$, can generally be obtained by multiplying $|\mathbf{G}(s)|^2$ times the PSD of input signal in frequency domain, and the output PSDs can be added together [6]. Thus, the PSDs of open-loop position error signals at encoders, $\mathbf{d}_{eso}$, can be obtained with

$$\mathbf{d}_{eso}(s) = \left|\mathbf{G}_{es,g}(s)\right|^2 \times \mathbf{d}_g(s) + \sum_{i=x,y,z} \left|\mathbf{G}_{es,bi}(s)\right|^2 \times \mathbf{f}_{bi}(s) \tag{1}$$

where $\mathbf{G}_{es,g}(s)$ and $\mathbf{G}_{es,bi}(s)$ are 3×6 matrices of which elements are frequency response functions from the ground vibration and bearing forces to the relative displacement between the encoder and scale, respectively.

The PSDs of the encoder signals when position control is activated, $\mathbf{d}_{es}(s)$, are calculated by

$$\mathbf{d}_{es}(s) = \left|\frac{1}{\mathrm{I} + \mathbf{G}_c(s)\mathbf{G}_{es,m}(s)}\right|^2 \times \mathbf{d}_{eso}(s) \tag{2}$$

where $\mathbf{G}_c(s)$ and $\mathbf{G}_{es,m}(s)$ are 3×3 diagonal matrices of the control functions of the *X*, *Y* and *Z* axes and frequency responses at encoders to motor excitations, respectively. Note that the inverse of the matrix here means element-wise operation.