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ABSTRACT
The role of a machine tool structure is to support the machine tool and the workpiece under the action of various, mainly cutting, forces. These forces often result in unacceptable levels of vibration of the tool and workpiece. Mechanical joints are the primary source of damping in most passive structures. Therefore, vibration reduction and energy dissipation characteristics of machine tool joints is of great interest. In a large class of machine tools, either the workpiece and/or the tool translates along slideways. Design of these joints is critical to the machining performance. Understanding the effects of the mechanical interface parameters (lubricant viscosity, saddle/slideway clearance and slideway surface finish) is needed to achieve appropriate balance of mechanical performance (friction, energy dissipation), environmental sensitivity (type and amount of fluid) and economic concerns. Experiments were conducted to analyze these parameters. Specifically, four lubricant conditions were investigated, dry (no lubricant) and three commercially available lubricants (spanning a broad range of viscosity). Initial investigation showed minor differences between the three lubricants (damping is primarily viscous and of similar level, but significant difference between dry (primarily structural damping)). The effect of machined versus ground slideway did not show markedly different energy dissipation characteristics. The investigation of clearance showed some effect on the damping behavior.

NOMENCLATURE

- $a_i$: Random noise
- $A$: Area of Crosssection
- $A_k$: Residue of the mode $k$
- $A_0$: Residual Sum of Squares for Higher Order Model
- $A_1$: Residual Sum of Squares for Lower Order Model
- $B$: Back Shift Operator
- $B(s)$: System Matrix
- $[C]$: Damping Matrix
- $E$: Young's Modulus
- $H(s)$: System Transfer Matrix
- $I(x)$: Moment of Inertia Function
- $[K]$: Stiffness Matrix
- $[M]$: Mass Matrix
- $N$: Number of Observations
- $r$: Order of Higher Model of $A_0$
- $s$: Difference in Order between $A_0$ and $A_1$
- $t$: time
- $\{X\}$: Displacement
- $\{\dot{X}\}$: Velocity
- $\{\ddot{X}\}$: Acceleration
- $x_i$: Observation at time $i$
- $\beta$: Eigenvalue
- $\Delta$: Sampling Interval
- $\lambda_i$: Characteristic root of AR model
- $\nu$: Poisson Ratio
- $\rho$: Density of the Material
- $\phi_i$: Auto Regressive Coefficient
- $\sigma_k$: Real Part of the System Pole
- $\omega_k$: Imaginary Part of the System Pole
- $\omega_d$: Damped Natural Frequency
- $\zeta$: Damping Ratio

INTRODUCTION

Customers are demanding ever greater levels of quality for machined components. Dimensional accuracy is a critical attribute in the overall quality of a machine component. Continual pursuit of quality and productivity has placed emphasis on the design of mechanical and precision components in a machine tool.

Machine tool errors may be classified into systematic and random errors. Systematic errors are repeatable and hence predictable, while random errors are stochastic and are described by a statistical distribution. Based on the rate at which they vary with time, systematic errors can be further classified into quasi-static and dynamic errors. Quasi-
static errors vary slowly with time and are related to the machine tool structure itself, while dynamic errors vary rapidly with time. Dynamic errors include errors of spindle motion, vibration of the structure, etc. Dynamic errors are due to those forces that cause the machine to respond/vibrate in a manner which creates an undesirable surface finish on the part.

Dynamic errors are primarily caused by structural vibrations and friction. Increasing structural stiffness decreases the magnitude of structural vibrations but is likely to come at the expense of a larger structural elements and a more expensive machine tool. Generally it is observed that higher the friction in the bearings, the better the damping and the less chance of generating chatter in the tool. However it becomes more difficult to move the tool at a high speed and with high dimensional resolution. Further, high friction causes heat to be more rapidly generated in the bearings. To avoid these errors the machine tool components should possess good damping characteristics at reasonable levels of friction. Beards [1982] reported that 90% of the total damping in a structure originates in the joints. Therefore it is of primary importance to understand the behavior of common joints, to properly design a machine tool to have appropriate damping levels, thereby eliminating the need for expensive and complex add on devices.

Machine tool structures generally consist of two kinds of joints: fixed joints and sliding joints. Research work on joint damping has concentrated on experimental approaches, such as Yoshimura [1976], Tsutsui and Ito [1979], Padmanabhan and Murty [1991], Lanz and Gaul [1995]. Most of the experimental approaches were mainly aimed at measuring hysteresis loops of the joint in loading through sophisticated experimental set-ups. Cao [1996] reported that a tightly bolted joint does not add significant amount of damping to the structure. Ayroma and Funakoshi [1990] combined both rolling guideway and slide-way joints to improve damping capacity in machine tools. Wang and Jiang investigated the influence of inserting damping oil films into machine rolling slideways and found that vibration-proof ability of joints can be significantly improved by damping. The damping capacity of a lap-joint subjected to partial slip has been shown to be maximum when force transference is by friction. Many engineers view dry (coulombic) friction as being detrimental to system performance, others view it as being very beneficial. Dry friction can result in a loss of accuracy in servo mechanisms, a loss of efficiency in power generation units such as automotive engines, and could result in the wear of contacting parts. However dry friction can also enhance system performance due to its damping and isolation properties. This is especially true in lightly damped flexible structures and in seismic isolation systems.

This paper presents a fundamental research characterizing the effects of joint conditions on the damping capacity of sliding joints. Experimental investigation was undertaken to study the impact of factors such as lubrication, clearance, and surface finish on damping.

![Diagram of Saddle Geometry, Guideway Geometry, Assembled Configuration](image)

**Figure 1: Experimental Set Up**

**EXPERIMENTAL METHOD**

Impulse response experiments were conducted to evaluate the damping capacity of the slideways. The sliding joints used in the experiments included two components, the saddle and the base. In this study, the saddle was made of cast iron (mass of 6.8 kg) and the base was made from 1018 Steel (mass of 17.7 kg). The dimensions of saddle, and guideway are shown in the Figure 1.

The saddle and the base are both assembled in an open rectangular configuration. The contacting surface of the first saddle was machined by an end-mill. The surface roughness \( R_a \) was measured to be an average of 1.6 \( \mu m \). For the second saddle the surface was ground and the average surface roughness was 0.2 \( \mu m \).
A piece of polyurathane foam was placed underneath the test setup to simulate free-free boundary conditions for the impulse response test. The experimental configuration and directions of testing are shown in Figure 1. The hammer was struck on the base, while the accelertometer was mounted on the saddle to pick up the signals. To investigate the effect of changing the clearance the saddle was moved along the base to five equally spaced locations. The base was tapered to have a linearly varying width.

Time and frequency domain response signals were collected from the digital signal analyzer. The sampling interval of the signal analyzer was 0.2441 ms. This corresponds to a 4096 Hz sampling frequency. This sampling interval, in conjunction with anti-aliasing filters, assures that the dominant lower frequency modes are retained, while the higher frequency modes do not corrupt the analysis.

**ANALYTICAL METHOD**

To estimate the natural frequency of the saddle and guideway, analytical methods were employed. The analytical method assumes that body is continuous. The guideway has a uniform cross section, the governing equation for free transverse vibration is:

\[
EI(x) \frac{d^4 w(x, t)}{dx^4} + p A dx \frac{d^2 w(x, t)}{dt^2} = 0
\]

Under free-free boundary conditions, the natural frequency (λs) (βs) of the beam are derived from the solution:

\[
\cosh(βs) \cos(βs) = 1
\]

\[
\omega_n = \sqrt{\frac{EI}{pA}} B_n
\]

The natural frequencies for the base were calculated from this relation to be 899 Hz and 2450 Hz for the first and second mode respectively. These values were used as a validity check for the frequency response data and the finite element model. Both the saddle and guideway were also modeled in IDEAS using ‘Guyan Reduction Method’ to identify the natural frequencies. The results indicated that the natural frequencies for the first two modes of the base were 850 Hz and 2201 Hz and that of the saddle were 2206 Hz and 3700 Hz. The fundamental frequency for transverse vibration was obtained as 936 Hz.

**TIME DOMAIN MODELING:**

To estimate damping in the time domain, an Auto-regressive (AR) model was fit to the data. In the AR model, the data is regressed on its own previous values. The AR (n) difference equation, for the signal, X, is the following:

\[
X_t = \phi_1 X_{t-1} + \phi_2 X_{t-2} + \ldots + \phi_n X_{t-n} + \alpha_t
\]

where \( φ \) is the coefficient vector.

Using back shift operator B, the equation (5) can also be expressed as:

\[
X_t (1 - \phi_1 B - \phi_2 B^2 - \ldots - \phi_n B^n) = 0
\]

The operator term can then be rewritten in factored form as

\[
(1 - \lambda_1)(1 - \lambda_2)\ldots(1 - \lambda_n) = 0
\]

The roots can be real or complex conjugate pair. In the case of complex conjugate pair, a sinusoidal decaying series exists. Therefore each conjugate pair represents a potential mode of vibration. Using equations the natural frequency and the damping ratio can be determined for each mode as:

\[
\omega_d = \frac{\sinh(βs)}{2\sqrt{λs} \sinh(λs)}\frac{1}{Δ} = \frac{\sinh(βs)}{2ω_n}\frac{1}{Δ}
\]

where Δ is the sampling interval.

To select an adequate model for the signal, the data were first fitted to a lower order model, then a higher order model was applied to see the difference in the reduction of the residual; sum of squares. If the reduction was significant, the higher order model was adopted. This procedure was continued until there was no significant reduction in the sum of squares. The F-test criterion was adopted to determine the adequacy of higher order model.

\[
F_{calc} = \left(\frac{A_1 - A_0}{s}\right) + \frac{A_0}{N - r}
\]

The result of the calculated F was then compared with the F distribution table statistic with a 95% confidence interval. If the calculated F value was higher than the table value the higher order model was adopted. A MATLAB program has been developed to fit AR(n) model to time data. This program also computes the residual sum of squares, natural frequency and damping ratio for each complex roots. To ensure the program functioned properly simulated data were used to test its accuracy.
FREQUENCY DOMAIN MODELING:
Damping can also be estimated in frequency domain using various curvefitting techniques. The homogeneous equation of motion for a multiple degree of freedom system can be written in matrix form as:

\[ [M][\ddot{x}] + [C][\dot{x}] + [K][x] = 0 \]

This equation can be rewritten as:

\[ [B(s)][x(s)] = 0 \]

where matrix \( B(s) = [[M]s^2 + [C]s + [K] \) is known as system impedance matrix. The inverse of this matrix is system transfer function matrix, \( H(s) \). The system matrix evaluated at \( s=j\omega \) provides the systems frequency response function. In partial fraction form, the frequency response function is represented by equation:

\[ [B(s)]^{-1} = [H(s)] = \sum_{k=1}^{m} \frac{[A_k]}{(j\omega - p_k)} + \frac{[A_k^*]}{(j\omega - p_k^*)} \]

where \( A_k \) is the residue of the mode \( k \).
\( p_k = \sigma_k + j\omega \) gives the values of natural frequency and damping.

Coefficients of a polynomial model of the frequency response function are estimated using built-in module of the STAR system. A root finding solution is used to determine the modal parameters, natural frequency and damping. (The advantage of the polynomial form is that the equations are linear and the coefficients can be solved by a non-iterative process which makes the convergence problem minimal and computer time more reasonable.)

RESULTS
The results from all three different natural frequency and damping estimation techniques were in close agreement. Figure 2 shows the time response of the base and it indicates that damping in the structure is negligible. The damping ratio calculated from FRF and DDS was only 0.0005 as the expected source of damping was only material damping. Impulse response tests on both the milled and ground saddle produced nearly the same damping ratios.

![Base Response](image1)

Figure 2. Impulse Response of Base

![Assembled interference](image2)

Figure 3. Impulse Response of the Assembled, Un-lubricated Sledway

The acceleration response signal in Figure 3 decayed much faster than those obtained individually for base and the saddle. This suggests that assembled sledway possesses a high

<table>
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<th>Table 1: Testing Conditions</th>
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<td>Surface Finish</td>
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<td>Lubricant</td>
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<td>Clearance</td>
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<td>Direction of Testing</td>
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damping property. When the saddle was placed on the base, some of the FRF's indicated that first mode of vibration was the most dominant. It can be also noted that the damping envelope shows a linear decay which is a characteristic of the coulomb or dry friction damping. Figure 4 shows the response characteristics of an assembled slide-way lubricated with Lubecon lubricant. The figure also indicates that at high amplitudes the decay is again linear similar to the response of a un lubricated or a dry joint and exponential at low amplitudes. The response of a lubricated slideway joint (Mobil Gear and Mobil Vactra) shown in Figure 5 follows an exponential decay pattern like in viscous damping.

![Figure 4. Impulse Response of the Assembled, Lubricated Slide-way, Lubricant 1](image)

![Figure 5. Impulse Response of the Assembled, Lubricated Slide-way, Lubricant 2](image)

As mentioned earlier the response characteristics were also studied in the frequency domain using Frequency Response Functions. The coherence function was observed to make sure that there was good relationship between the input signal and the output response. The FRF’s gave a good indication that vibration modes for the assembled structures are heavily damped compared to individual elements. Some of the modes were closely spaced and indicated the presence of transverse modes. Figure 6 shows a sample plot of coherence and frequency response functions for both dry and lubricated slideways at some clearance position used for calculation of damping ratios.

![Figure 6. Frequency Response Function and Coherence for Dry and Lubricated Slide-way](image)

![Figure 7. Damping Ratio Versus Clearance: Transverse Vibration with Ground Saddle](image)

Figures 7-10 show the damping ratios obtained for all the impulse response tests. Dry friction joints exhibited a higher damping ratios than lubricated joints. The addition of lubrication has significant reduction in joint damping. The more viscous lubricants tend to have lower joint damping although
there was some deviation from the observed trend. Figures 8, 9 show that joint damping is independent of lubricant and mainly depended on the clearance (placement) of the saddle on the base.

Surface finish does not seem to have significant influence on the joint damping. The saddle with a milled surface provided a slightly higher friction force than the saddle with a ground surface finish.

Clearance seems to have significant impact on the joint damping irrespective of the lubricating condition. There seems to be an optimum clearance value at which the system possesses good energy dissipation characteristics.

Auto-Regressive models are suitable for lightly damped structures. The AR models fitted to the time domain data gave accurate damping ratio estimates.

REFERENCES


