inter.noise 2000
The 29th International Congress and Exhibition on Noise Control Engineering
27-30 August 2000, Nice, FRANCE

I-INCE Classification: 4.1

NOISE AND VIBRATION ANALYSIS OF MACHINE TOOLS USING INVERSE METHODS

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Keywords:
VIBRATIONS, CHATTER, INVERSE METHODS, MACHINING

ABSTRACT
The vibrations of machine tools create noise problems and have an important influence on the quality of the finished surface particularly when chatter occurs. The exact identification of the cause of vibration which is linked to the non linearity of the response to the dynamic forces at the tool-work piece interface is essential. To help solve this problem an inverse method of indirect identification of the dynamic forces at the tool-work piece interface has been developed which may be applied during machining. The use of previously measured frequency response functions between the tool point and points on the tool support and machine structure associated with the cross spectral matrix of accelerations measured during the cutting process permits these forces to be identified and related to a parametric model of the cutting process.

1 - INTRODUCTION
During the turning process the static and dynamic forces generated by the tool interaction with the work piece may be measured with the aid of specifically designed force dynamometers. At the higher frequencies associated with chatter and other vibratory phenomena the internal resonances of the measurement system mask the response. The true dynamic forces and accelerations at the tool-work piece interface are required in order to correctly identify the appropriate model of the cutting process and predict the levels of vibration generated by the machining with different materials and different cutting speeds and depths. The inverse method of force identification is well adapted to this task and has been applied by us to turning and milling operations on several types of machine.
To find an appropriate model of the machining process we have developed a modified form of the well known model of Marui [1] which has been used successfully to predict chatter frequencies on relatively low frequency cases, but which cannot be used to predict chatter amplitudes.

2 - EXPERIMENTAL METHOD
A matrix of frequency response functions between the three input force directions and multiple output responses is formed from measured data using directly measured force inputs and accelerometer response measurements.
Figure 1 shows the tool and support of a lathe fitted with a Kistler dynamometer and accelerometers. The frequency response functions for the FRF matrix can be measured by excitation with a shaker or shock hammer at the tool point and measuring the acceleration at the output response points or, assuming that reciprocity is respected measuring at the tool point and exciting at the output response points. The direct shock hammer procedure is illustrated in figure 1. The Kistler dynamometer (type 9265B could be used to validate the identified forces at low frequencies before its internal resonances masked the true response.
The dynamometer itself can be calibrated by this procedure to show the frequencies at which resonances occur.

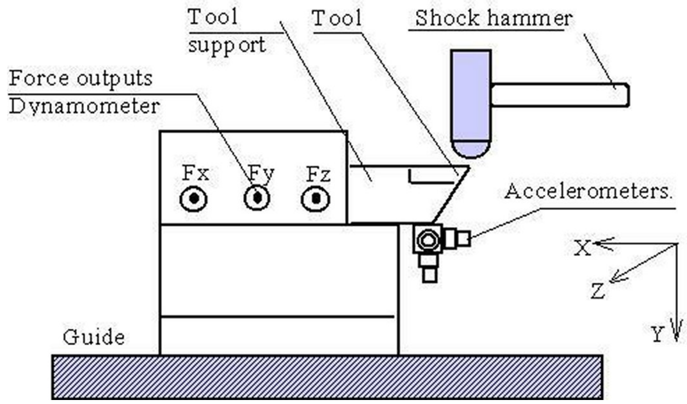

Figure 1: Tool and support and method of acquiring frequency response functions.

3 - FORCE IDENTIFICATION

The calculation of the spectra of the forces at the interface with this method requires the inversion of this frequency response function matrix which may be achieved with the aid of the singular value decomposition. The number of singular values in the frequency response function matrix permits the quality and independence of the output response points to be verified. The spectral matrix of the responses at the output points is then measured during the machining process. The input forces may then be identified by multiplying the response matrix by the inverse or pseudo-inverse of the frequency response functions. Point impedance measurements may also be used to identify the acceleration response of the tool point from the identified force spectra.

The relations between the cross spectral matrix of accelerometer responses $[G_{xx}]$ and the matrix of the forces at the tool point $[G_{ff}]$ may be expressed.

$$[G_{xx}]_{M\times M} = [H]^H_{M\times N}\,[G_{ff}]_{N\times N}\,[H]_{N\times M} \quad (1)$$

where M points are measured. The responses to the N unknown forces are related by $[H]$ the previously measured frequency response function (FRF) matrix with the help of the hermitian transpose operator $[\,]^H$

The solution for the unknown forces can be expressed in spectral matrix form in equation (2) with the aid of the pseudo inverse of the previous expression. The pseudo inverse of the FRF matrix is calculated with the aid of singular value decomposition.

$$[G_{ff}]_{N\times N} = [H]^{H+}_{N\times M}\,[G_{xx}]_{M\times M}\,[H]^{+}_{M\times N} \quad (2)$$

The methods have been applied to several types of lathe in order to obtain a better understanding of the cutting process and a simple model based on the work of Marui was used to identify the cutting parameters under the realistic conditions. The causes and the effects of chatter and the identification of the parameters associated with the vibrations of the system may then be evaluated during the machining process at relatively high frequencies.

Measurements show that the dynamometer response is dominated by its own resonances above 500 Hz. which justifies the use of the inverse method to identify the force spectra at the tool point. Particularly sharp peaks were observed at 900 Hz and 1800 Hz for the measurements in the x or radial direction. An example of the identified forces under sustained chatter conditions is shown in figure 2 where it can be seen that the spectra of all three point forces are dominated by the chatter frequency.

4 - MODEL AND PARAMETER IDENTIFICATION

The expression developed to model the turning process assumes that the effects of any previous periodicity in the machined surface are negligible (primary chatter only) and is developed from Marui's model for a single degree of freedom in the radial direction. It is assumed that the movement of the work piece is negligible, which is true for short and large diameter pieces.

$$m\ddot{x} + \left(c_m + c_a x_0 - \frac{bK_{N0}\sin\theta}{\omega} + \frac{ba_{Na}d_s}{v_0}\right)\dot{x} + \left(K_m + bK_{N0}\cos\theta\right)x = 0 \quad (3)$$

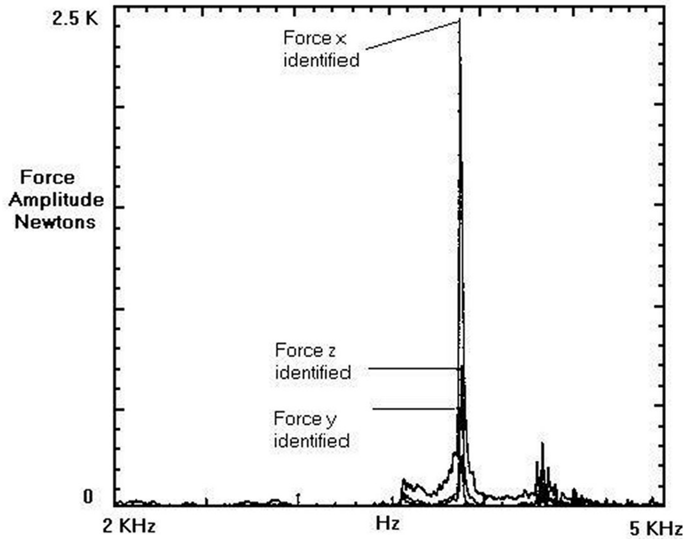

Figure 2: Forces identified in stable chatter conditions in the x, y and z directions.

The equivalent mass, stiffness and damping of the first mode of the tool and its supporting structure m, K_m and c_m were obtained by measurements of the point frequency response function at the tool point in the radial direction. The nominal cutting width and depth are b and d_s respectively, the cutting velocity v_0 and the vibration amplitude is x_0. The values identified for this test were; m=1.98 kg, K_m=1.253 $\times 10^9$ N/m, and c_m=3533 N.s/m.
In order to determine the parameters linked to the machining process, c_a, a damping coefficient proportional to amplitude, K_{N0}, the cutting stiffness per unit width, and $a_{N\alpha}$, the correction factor for the cutting angle the characteristics under stable chatter conditions can be used for three different machining conditions. The "negative damping" induced by the phase difference between the velocity of the tool point and the components of the reaction forces which are expressed parametrically in terms of the last two terms in the first set of brackets in equation 3 exactly balance the sum of the intrinsic damping of the system c_m and the second amplitude proportional damping term c_a which limits the maximum chatter level. Assuming during chatter that the damping terms are reduced to zero and that the parameters are independent of frequency, equation 3 may be rewritten;

$$\left(c_m + c_a x_0 - \frac{bK_{N0}\sin\theta}{\omega} + \frac{ba_{N\alpha}d_s}{v_0}\right) = (K_m + bK_{N0}\cos\theta) - m\omega^2 = 0 \quad (4)$$

Equation 4 can then be expressed in the following form;

$$\left(c_m + c_a x_0 + \frac{ba_{N\alpha}d_s}{v_0}\right)^2 \omega^2 + \left(K_m - m\omega^2\right)^2 = (bK_{N0})^2 \quad (5)$$

There are three unknowns to determine to obtain our parametric model and these can be resolved by performing three experiments using slightly different parameters or machining conditions such as the cutting velocity or the cutting depth. The inverse method is used to identify the forces, frequencies and amplitudes of vibration at the tool point in each case.

Velocity V_0 (m/min)	b (mm)	K_{N0} (N/m^2)	C_a (N.s/m^2)	$a_{N\alpha}$ (N/m^2)	$\omega/2\pi$ (Hz)	d_s (mm)	x_0	θ (deg)
120	0.30	6.34*10^{11}	8.25*10^8	-1.02*10^{10}	3887	2	9.3 μ	112.2
110	0.25	6.34*10^{11}	8.25*10^8	-1.02*10^{10}	3916	2	6.6 μ	109.9
110	0.35	6.34*10^{11}	8.25*10^8	-1.02*10^{10}	3857	2	9.9 μ	118.9

Table 1: Identified and known parameters for three chatter conditions.

The parametric phenomenological model can therefore be expressed in terms of the numerical values obtained from these experiments as follows.

$$1.98\ddot{x} + \left(3533 + 8.245\times10^{8}x_0 - \frac{6.344\times10^{11}b\sin\theta}{\omega} - 1.018\times10^{10}bd_s\right)\dot{x} + \left(1.253\times10^{9} + 6.344\times10^{11}b\cos\theta\right)x = F_x^{(r)} \quad (6)$$

An alternative approach is to calculate the minima of the function G^2 as given in equation 7.

$$G\left(c_a, a_{Na}, K_{N0}\right) = \left(c_m + c_a x_0 + \frac{b a_{N\alpha} d_s}{v_0}\right)^2 \omega^2 + \left(K_m - m\omega^2\right)^2 - \left(bK_{N0}\right)^2 \quad (7)$$

The values obtained are very similar for both approaches. Predictions using the identified parameters give a maximum difference of around 20% between predicted and measured amplitudes and very close agreement in frequency values.

5 - CONCLUSIONS

The use of inverse techniques in association with a modified version of the model of Marui can produce interesting results for the characterisation of the cutting process. The model obtained underestimates the amplitude of chatter by about 20% with the present state of experimental accuracy, but gives an accurate indication of the chatter frequency. This error is considered to be acceptable, but may be reduced when more accurate methods are used to measure the frequency response function matrix. The coupling of the movements with the other modes and directions of movement of the tool support structure are also factors affecting accuracy which are being investigated in the next phase of the work.
As may be expected the limiting cutting width also increases if the intrinsic damping c_m or stiffness K_m of the system is increased. The method may also be transposed to other types of problem such as milling, grinding and the dynamic response of friction forces.

REFERENCES

1. **E. Marui, S. Kato, M. Hashimoto and T. Yamada**, The Mechanism of Chatter Vibration in a Spindle Workpiece System, *Journal of Engineering for Industry, ASME*, Vol. 110, pp. 236-247, 1988