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Experimental Modal Analysis of Machining Centers

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ABSTRACT

Experimental modal analysis is an effective tool to investigate the dynamic behavior of machining centers. This paper presents the measurement system and experimental investigation on the modal analysis of machining centers. The modal analysis shows the weak part of the machining center. An important local model of spindle in our experiment, which influences the dynamics of machining process, is proposed in this paper. The results provide the foundation of structure modification for good dynamic behaviors.

Key Words : Modal analysis, Machining center, Frequency response function

1. Introduction

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The dynamic behavior of machining center influence the machining performance, tool life, chatter vibration and machined workpiece's surface quality. Modal testing and analysis is an effective tool to find out the weak parts of a machining center and can supply necessary data for structure modification and chatter analysis. To obtain the dynamic behavior of a machining center, it is common to use the identification method in the frequency domain based on input and output signals. It is necessary to excite the structure within an interested frequency band. There are several methods used by former researches. Minis *et al.* [1] discussed the impact method in details. Tobias [2] and Koenigsberger *et al.* [3] used a special electromagnetic exciter to apply a sinusoidal force between tool and workpiece. They realized that the state of the vibratory system must simulate as close as possible the state of the tool when testing. Kim *et al.* [4] used an electromagnetic exciter to generate a random input force. Bonzanigo *et al.* [6] used an unequally spaced milling cutter to generate a broadband cutting force that excited all the modes of a milling machine in the bandwidth of interest. Their results are valid only for a particular direction of the cutting force. Opitz *et al.* [7] excited the machine tool's structure by random cutting force generated from the continuous cutting of "random" workpiece. A series of impact tests were also conducted in their study [1] for comparison purposes. They also used a specially modulated workpiece to generate a pseudorandom periodic force signal.

Najeh *et al.* [8] explained that the resulting transfer matrix is not symmetric. This is due to the presence of some gyroscopic phenomena when the structure is excited by chock. It is well known that the static stiffness of the machine tool's joints and guideways display nonlinear characteristics [9]. Such nonlinearities, however, can be minimized with the sufficient preloading of a structure's individual components. In the last several decades, some researchers have studied the dynamic behavior of machine tools. In our research on the dynamic behavior of machining center incorporated in machine tool industry, we found that there are still some aspects concerned with the test, which should be studied.

In this paper, we used an electrohydraulic exciter, which can apply static force in addition to random force. The electrohydraulic exciter has a lot of advantages compared to other type of exciters: It has a great force intensity per volume so that it can be applied to small and large size of machine tools. It shows also a great frequency range for exciting test. Since the machine tool's joints and guideways display nonlinear characteristics, it is necessary to apply an appropriate

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amount of static force to overcome this kind of nonlinearities. Several different kinds of fixtures and dummy tools were manufactured for the experiment, since it is better to simulate the testing state as close as possible to the state of machining.

2. Modal Testing System and Theory

Our modal testing system comprises an electrohydraulic exciter, a dummy tool, accelerometers, a signal conditioner (FFT), a note book computer and modal analysis software. Figure 1 shows the dynamic behavior testing system.

Three Kistler 8774A50 accelerometers were mounted alone in the three orthogonal directions on a magnetic mounting base. which can be conveniently attached to measurement points on the structure. The signals from the accelerometer and force sensor were conditioned by a Siglab 20-42 signal conditioner which has four input channels and two output channels. The output signal of the conditioner is controlled by a digital signal processor. In our testing, a random output signal with a bandwith of 1KHz was used to drive the electrohydraulic exciter. When starting test, firstly we apply a static force from the exciter's head to a dummy tool. Then we apply additional random dynamic force to the dummy tool. The number of data points per channel per average was 4096 in data acquisition. For obtaining good results, the FRF (Frequency Response Function) estimation was finished after 100 times of average.

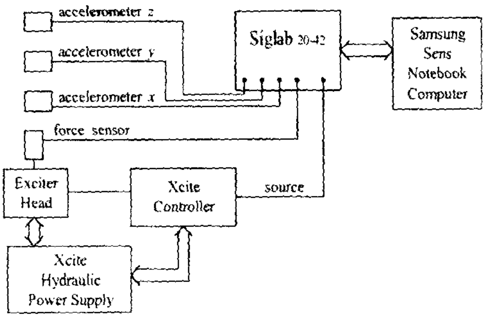

Figure 1 The dynamic behavior testing system

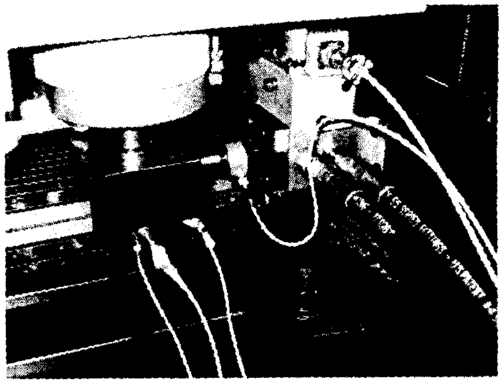

Figure 2 Installation of electrohydraulic exciter

The function of an electrohydraulic exciter is to impart a controlled force into structures such as machine tools. The force generation is accomplished by a closed loop electrohydraulic system which can apply a static preload in addition to dynamic force up to 1200 Hz under either sine or random conditions. Dummy tool is used to substitute a real tool in static and dynamic stiffness estimation of machining center.

The head of an electrohydralic exciter shown in Figure 2 is mounted on the fixture. which is installed on the working table. The fixture should be strong enough for static and dynamic force to be applied.

We use the frequency domain modal analysis method. The system frequency response function can be estimated by the input force and the response accelerations. From the theory of mechanical vibration, the system transfer function can be expressed as:

$$H_{rp}(s) = \sum_{i=1}^{n}\left(\frac{A_{rpi}}{s-\lambda_i} + \frac{A^*_{rpi}}{s-\lambda^*_i}\right) \quad (1)$$

where $A_{rpi} = \lambda_i \phi_{ri} \phi_{pi} / a_i$,

$A^*_{rpi} = \lambda^*_i \phi^*_{ri} \phi^*_{pi} / a^*_i$.

A_{rpi} is the *ith* order modal constant when p is the excite point, r is the response point.

ϕ_{ri} is the *ith* order modal shape coefficients at response point r. In addition, * denotes complex conjugate.

Especially, we rewrite the above equation as a frequency response function, and using the rational fraction polynomial, it follows that

$$H_{rp}(j\omega_k)=\sum_{q=1}^{m}(j\omega_k)^{q-1}X_q/(\sum_{q=1}^{m}(j\omega_k)^{q-1}Y_q+(j\omega_k)^m)$$
$$=\frac{N(j\omega_k)}{D(j\omega_k)} \qquad (2)$$

We can estimate the coefficients of rational fraction polynomial by the measured FRF data. Further more, the modal frequencies and damping ratios can be obtained from the m pairs of poles and modal shapes from the residuals.

3. Modal Test and Analysis

We tested two kinds of machining centers A and B. Modal test took on the position where the spindle head was at the center of the working table. We selected 32 measurement points on the machining center A and 28 measurement points on the machining center B. Each measurement point installed three accelerometers in the three orthogonal directions x, y and z. The dummy tool at the end of spindle was selected as an exciting point since the dummy tool is the input point of machining force at normal operation. From the dynamic stiffness test, we found that the x and y direction stiffness are much smaller than the stiffness in z direction, so we select x direction and y direction as the exciting directions respectively. In modal test, the static force is $1000N$ and dynamic force is $300N$.

In actual machining, the machining force was applied on the spindle head. Thus the exciting force was applied to the dummy tool. In dynamic stiffness, we found that the dynamic stiffness in x and y directions is less than the dynamic stiffness in z direction, so the exciting force was applied from x direction and y direction, respectively. The response points were distributed on the spindle head, column, working base, and the base of machine tool.

Figure 3 shows all measurement points' FRF on machining center B. The overall FRF data is very complex. However, in the measurement point on the spindle head, we found an important local model which has the largest response signal in all measurement points shown in Figure 4. We found the same phenomena on the test of machining center A.

Table 1 shows the modal frequencies and damping ratios on machine tool A under x direction excitation. Figure 5 is the modal shape of the dominant model of dummy tool under x direction excitation.

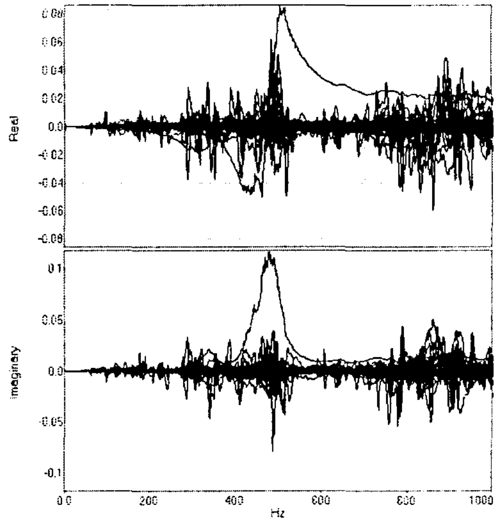

Figure 3 FRF data on machining center B, overlap display format

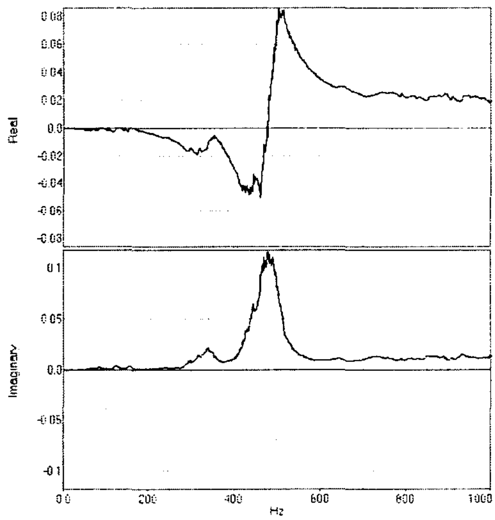

Figure 4 FRF data on machining center B, excitation point $1y$, response point $1y$

Table 1 Modal parameters of machine tool A under x direction excitation.

Modal Number	Frequency (Hz)	Damping Ratio (%)
1	38.464	6.943
2	69.398	3.951
3	93.911	7.131
4	109.244	2.765
5	124.331	3.456
6	155.75	6.859
7	223.132	1.738
8	325.882	2.494
9	361.255	1.064
10	506.975	3.585

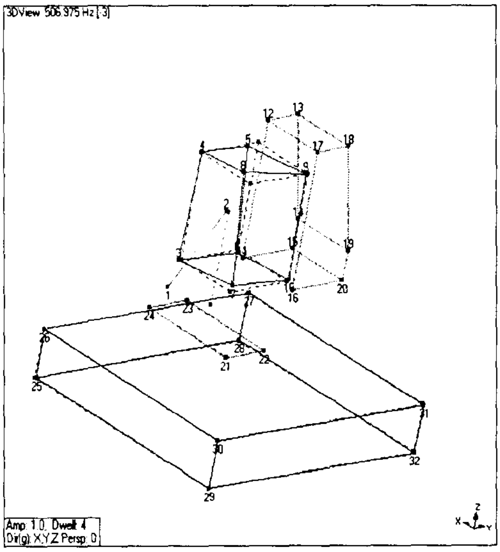

Figure 5 Modal shape, machine tool A, excitation point 1x

Table 2 shows the modal frequencies and damping ratios on machine tool A under y direction excitation. Figure 6 is the modal shape of the dominant model of dummy tool under y direction excitation.

Table 2 Modal parameters of machine tool A under y direction excitation

Modal Number	Frequency (Hz)	Damping Ratio (%)
1	38.067	8.131
2	51.518	0.685
3	102.582	2.075
4	115.172	4.517
5	152.103	3.613
6	185.209	3.476
7	282.45	4.981
8	337.813	3.142
9	497.655	4.32

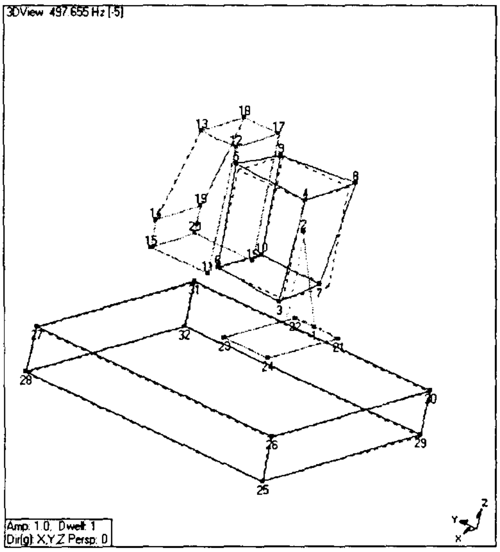

Figure 6 Modal shape, machine tool A, excitation point 1y

Table 3 shows the modal frequencies and damping ratios on machine tool B under x direction excitation. Figure 7 is the modal shape of the dominant model of dummy tool under x direction excitation.

Table 3 Modal parameters of machine tool B under x direction excitation

Modal Number	Frequency (Hz)	Damping Ratio (%)
1	25.109	5.46
2	54.678	1.998
3	85.681	2.882
4	109.222	1.459
5	176.391	1.794
6	255.483	1.42
7	463.22	3.03
8	481.84	0.743
9	502.701	0.334
10	581.87	0.758

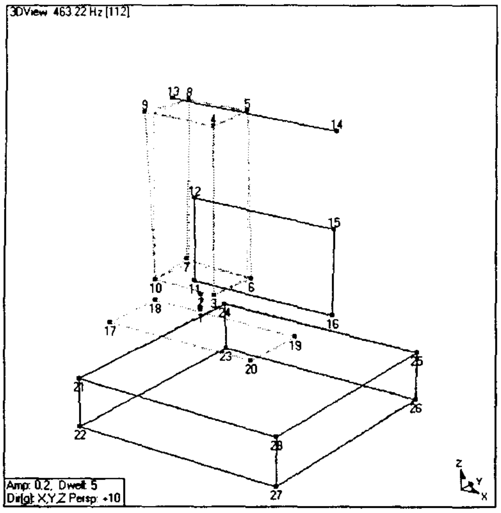

Figure 7 Modal shape, machine tool B, excitation point $1x$

Table 4 shows the modal frequencies and damping ratios on machine tool B under y direction excitation. Figure 8 is the modal shape of the dominant model of dummy tool under y direction excitation.

Table 4 Modal parameters of machine tool B under y direction excitation

Modal Number	Frequency (Hz)	Damping Ratio (%)
1	70.321	2.987
2	78.709	4.025
3	121.313	2.969
4	142.107	1.697
5	149.991	2.405
6	157.292	2.952
7	170.077	1.638
8	204.863	0.641
9	238.704	1.445
10	268.035	0.638
11	361.316	2.256
12	482.247	4.62

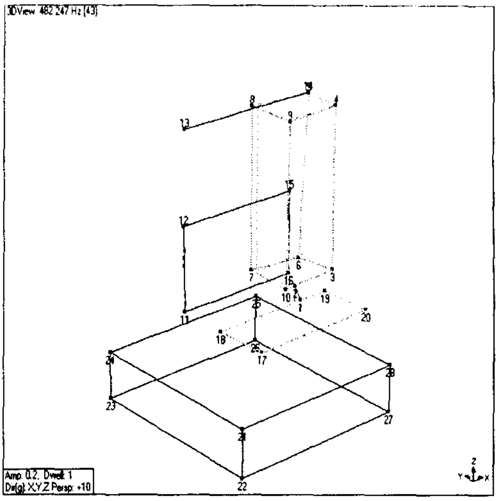

Figure 8 Modal shape, machine tool B, excitation point $1y$

The modal shapes shown in Figures 5 to 8 are the dominant model of dummy tool. Measurement point 1 is at the dummy tool head. Since the other parts of the machining center have no this modal frequency, this modal is a local model of dummy tool. However, the machining force acts on the tool, and this model has the largest modal shape in all modal shapes, which is an important model in machine tool dynamic analysis.

From modal shapes, there is one dominant resonance frequency in the above two machining centers. In the dominant model, the spindle head has much greater vibration level than the other parts of the machining center. The spindle head is the weak part in machining center from the view of dynamic behavior. A machine tool is a complex dynamic system, which has many models in our measurement frequency bandwidth. The modal shape of a spindle has an influence on the machining stability and the machined surface. Unfortunately, the spindle head has the largest coefficient of modal shape and the smallest dynamic stiffness in our test. How to increase the dynamic stiffness of spindle head is important for improving machined quality.

4. Conclusion

Although there are many models in our testing frequency band, the response signal of spindle has one dominant natural frequency in which the response of spindle is much greater than other parts of machine tool and other frequency ranges in most cases. In the mean while, the other parts of the machine tool have no this kind of dominant natural frequency. Hence, this dominant natural frequency indicates a local model.

When the static force is 1000N, with the increase of dynamic force, the resonance frequency decreases slightly, which shows the system is nonlinear.

The total modal parameters should be considered in structure modification. However, this local model influences dynamic behavior significantly. The connection joint between tool and spindle plays an important role in the dynamics of machining.

Acknowledgement

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