UDC 621.9-216
624.072.33.042.7

A COMPUTER AIDED ANALYSIS ON VIBRATION OF ELEMENT AND INTEGRATED STRUCTURE OF MACHINE TOOL BY THE FEM-PART 1

有限要素法による工作機械構造の振動解析

—Study on Bed Type Structures—

—ベッド構造について—

by Hisayoshi SATO*, Yoshio KURODA* and Masataka OHORI*
佐 藤 壽 芳・黒 田 佳 男・大 堀 真 敬

1. Introduction

It has been pointed out that the vibration characteristic of machine tool structure has close relation with occurrence of the self-excited vibration[1),2)] and also affects the machining precision by the forced vibration which is caused by various sort of moving parts such as gear train, spindle, motor and so on[3)].

In order to improve the performance of machine tool from the view point of vibration it is required that the vibration characteristic of machine tool is identified during design procedure. On the other hand, the development of large scale and high speed computer and method of the analysis with aid of the computer makes it possible to perform this. Taylor[4)], Maltbaek[5)] and others obtained the natural frequencies and the normal modes by simulating the machine tool structure with equivalent beam system or lumped mass spring system. However, the configuration of the machine tool structure is generally so complicated that it is sometimes difficult to simulate this by simplified equivalent system. However, the general method giving the equivalent system from the actual system has not been necessarily shown.

As for the structure other than machine tool especially that of airplane the detailed analysis by the finite element method, which will be mentioned as FEM in the following, has been made for obtaining the vibration characteristic. However it needs enormous amount of computation and large scale high speed digital computer compatible to demand of the analysis for such complex structures. As the result very few contributions have been done about static and dynamic problems in the field so far[6),7),8),9),10)].

In this paper it is proposed that the vibration characteristic of each element structure is independently obtained by making use of the FEM as it is thin plate structure[11)]. Then simplified equivalent beam for the element structure is given based on the analysis before the total integrated structure system is built up by the equivalent one. The analysis for the integrated system is made by applying the FEM again to the beam structure[12)].

2. Analysis on the Bed Structures

2.1 Analysis on bed structure of ladder-like shape

Fig. 1 shows general view and dimension of the

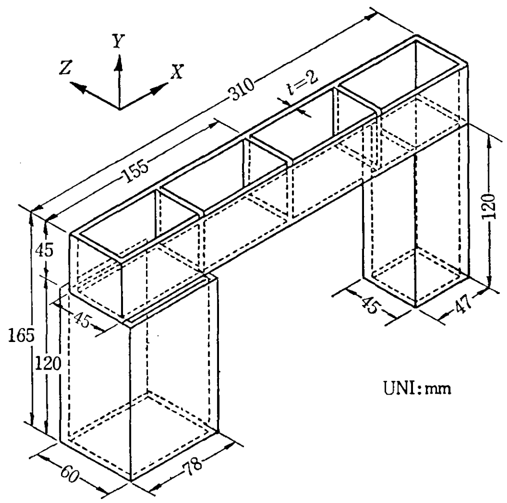

Fig. 1 Perspective of simplified lathe model.

* Dept. of Mechanical Engineering and Naval Architecture, Inst. of Industrial Science, Univ. of Tokyo.

structure which is used for the analysis. The model is made from methacrylic resin perspex plate. Although it does not have gear box, spindle system, tail stock and other auxiliary system, the simplified structural configuration is based on standard type of lathe. The total structural system is integrated by element structures such as a bed and its supporting columns. About these elements the configuration is idealized and simplified, so that slide bearing is removed, complicated shape of panels are exchanged with rectangular plates and so on.

Fig. 3 compares the natural frequencies and the dominant bending modes by the excitation of the perspex model with those by the FEM analysis about the structure shown in Fig. 2 (a). The experiment is carried out by fixed-free boundary condition, which is most feasible to realize for the size of the model.

Standard technique of the FEM is applied for the analysis of vibration characteristics of plate and beam structures. Details should be attributed to the references. Rectangular element is taken for the analysis of plate structure. The eigen value and the eigen vector corresponding to the natural frequency and the normal mode respectively are all obtained by Jacobi's method.

About the vibration modes the first and the last one show very good agreement in Fig. 3. There can be seen some disturbances around the excitation point about the other two modes. This is probably caused by that the model is excited keeping contact with the exciter. This may make a new boundary condition in spite that static force of the exciter is adjusted as light as possible.

The natural frequency also shows good agreement especially for the first one. However, as for the natural frequency accurate comparison seems difficult for some uncertainty of the perspex Young's modulus. It is usually taken as $E=3.3$

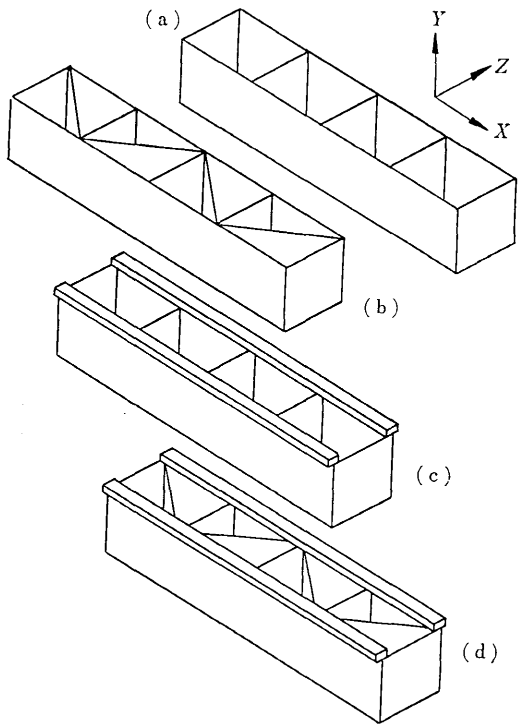

Fig. 2 Perspective of various types of bed structure.

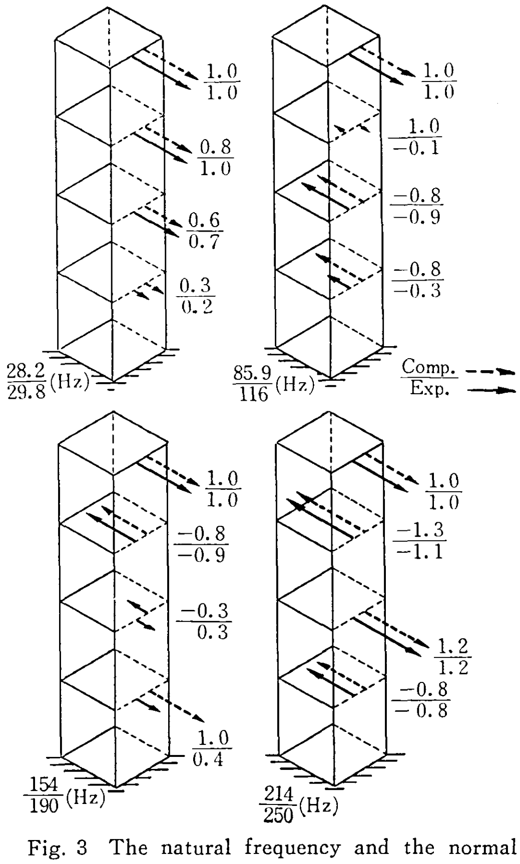

Fig. 3 The natural frequency and the normal mode by the computation and the experiment for the perspex model of the ladder-like bed.

$\times 10^4$ kg/cm^2 for normal condition[13]. Although it is statically calibrated in this investigation, it is not stable due to creep characteristic of the material and the mean value differs from the value described above within 10% at most, so that the value usually recommended is taken standard. It is also necessary to calibrate the relation between Young's modulus and frequency. In this study it is made for the range between 10 and 250 Hz. The measured frequency in the model experiment ranges up to about 1,000 Hz. As for the frequencies beyond the range calibrated[14], the extrapolation is adopted taking the reference into account.

As for the torsional vibration mode the natural frequency by the computation gives a much higher estimation than that by the model experiment as shown in Table 1. This is probably based on the reason that the assumed displacement function is not compatible for the rib plate deformation accompanied by the torsional mode.

2.2 Analysis on various types of bed structure

As the bed model several different types of structure are taken. Fig. 2 shows perspective of these bed models. The diagonal rib plates which are expected to increase torsional stiffness particularly are added in Fig. 2 (b). The models shown in Fig. 2 (c) and (d) have ribs corresponding to slide bearing, the dimension of which is 6 mm width, 4 mm thickness and same length as that of the bed model. It is more or less expected to increase the stiffness.

Table 1 gives the natural frequencies for various types of bed-like structure. For the type of structure Fig. 2 (a) the experiment by a model made from steel plate with 0.6 mm thickness is also carried out. This aims at eliminating the effect of frequency dependent properties of perspex Young's modulus in veryfying the results by analysis. Taking it into consideration that the FEM estimation generally gives upper bound for the theoretical value, both results agree well with respect to bending mode of vibration. The difference gets large and is not systematic for the perspex model, still quite good agreement can be found.

Table 1 also makes it clear that the role of the diagonal rib plate shown in Fig. 2 (b). The natural frequencies of the bending mode of z-direction and the torsional mode become remarkably higher. The effect attaching beams like slide bearing appear as slight rise of the natural frequencies of corresponding modes. Looking at the mode shape for Fig. 2 (c) and (d) in detail, the bending and the torsional mode of vibration are no longer independent, but coupled each other, although the amount of the coupling is still small for these.

At the bottom of the table the natural frequency of the bending mode of y-direction is given. It is difficult to find out this type of mode in the model excitation. The results are to be examined in detail, since it might be possible that the frequency is higher than that for the actual system by the same reason as is seen in the torsional mode of vibration.

3. Conclusions

Aiming at computer aided design of dynamic

Table 1 The natural frequency for the various types of bed structure.

Model in Fig. 3	(a)*		(a)		(b)		(c)	(d)
Dominant Mode	Exp.	Comp.	Exp.	Comp.	Exp.	Comp.	Comp.	Comp.
Bending-z	21.0	25.3	29.8	28.2	155	150.9	39.3	160.7
Bending-z	68.0	77.6	116	85.9	520	586.0	129.0	649.9
Bending-z	117	132.0	190	154.3	—	714.9	235.0	—
Torsion	210	446.4	90.0	176.4	—	653.0	175.7	580.0
Bending-y	—	473.0	—	159.6	—	171.9	167.2	179.5

* Steel plate (Hz)

behaviour of machine tool, the FEM is applied to identify the natural frequency and the normal mode of the element and the integrated total structure. The computation is mode for the element structures such as bed and its supporting columns taking standard lathe machine into consideration. The verification is proved by comparing the results with those from the excitation experiment of small size perspex and steel plate model.

The analysis makes it obvious as for bed-like structures how the regular and the diagonal rib plate, and the slide bearing play roles on dynamic stiffness quantitatively. The computation for the supporting columns also tells us that the normal mode cannot be obtained without the analysis as the plate structure.

(Manuscript received June 13, 1972)

References

1) Tobias, S. A., " Machine-Tool Vibration ", Blackie, (1961).
2) Merrit, H. E., " Theory of Self-Excited Machine-Tool Chatter ", ASME SER. B, (1965-11).
3) Sato, H. and Akutsu, T., " A Study of Indentification of Dynamic Characteristics of Machine Tool by Means of Micro Tremor ", Proc. 13th MTDR (1971-9).
4) Taylor, S. and Tobias, S. A., " Lumped-Constants Method for the Prediction of the Vibration Characteristics of Machine Tool Structures ", Proc. 5th MTDR (1964-9).
5) Maltbaek, J. C., " Classical Beam Method for the Prediction of Vibration Characteristics of Machine Tool Structures ", Proc. 5th MTDR (1964-9).
6) Stephen, A. C. and Taylor, S. A., " Computer Analysis of Machine Tool Structures by the Finite Element Method ", Proc. 9th MTDR (1968-9).
7) Cowley, A. and Hinduja, S., " The Finite Element Method Applied to the Deformation Analysis of Thin Walled Columns", Suppl. 12th MTDR (1971-9).
8) Murakami, A. and Kawasaki, I., " Study on Column Stiffness of Large Horizontal Boring Machine ", Jour. JSME, 628-5 (1971-5) (in Japanese).
9) Sata, T. and Takashima, N., " Analysis on Dynamic Stiffness of Machine Tool by the Finite Element Method ", Proc. Autumn Meeting JSPM, (1971-11) (in Japanese).
10) Takahashi, N. and Ohno, S., " Free Vibration of Lathe Bed ", Jour. 11S, Univ. of Tokyo, (1971-7) (in Japanese).
11) Zienkiewicz, O. C., " The Finite Element Method on Engineering Science ", McGraw Hill, (1971).
12) Martin, H. C., " Introduction to Matrix Mithods of Structural Analysis ", McGraw Hill, (1966).
13) Yasui, T. and others. " Plastic Materials Utilized for Model Test of Machine Tool Structures ", Jour. Mech. Lab., 20-4 (1966) (in Japanese).
14) Stansfield, F. M., " Some Notes on the Use of Perspex Models for the Investigation of Machine Tool Structures ", Proc. 6th MTDR (1965-9).