COMPUTER AIDED VIBRATION TESTING OF MACHINE TOOLS

M. Abdel Salam
Production Engineering Department, Faculty of Engineering,
Ain Shams University, Cairo, Egypt.

ABSTRACT

The dynamic characteristics of machine tools play an active role in chatter and noise problems. The dynamic characteristics are described in terms of transfer functions relating forces as an input and the system displacement as an output. Owing to the complexity of machine tool structures it is not possible to calculate their dynamic characteristics with sufficient accuracy. The usual way of the determinations of these characteristics is by the experimental methods such as impulse and random excitation techniques. The development of (FFT) analyzers make such tests fast and accurate where the transfer function can be calculated and plotted. On the other hand (FFT) analyzers may be too expensive for some research centers to buy it. The aim of the present work is to develop a low cost computerized system based on a Pc computer which can be used for the determination of the transfer function of machine tools in both stand-still or under cutting conditions.

INTRODUCTION

The dynamic characteristics of machine tools play an active role in chatter and noise problems [1]. The dynamic characteristics are described in terms of transfer functions relating forces as an input and the system displacement as an output. Owing to the complexity of machine tool structures it is not possible to calculate their dynamic characteristics. The usual way of the determinations of these characteristics is by the experimental methods [2].

A great deal of efforts has been devoted during the past several years for developing experimental techniques for the determination of the transfer functions such as harmonic, random and impulse excitation techniques [3,4,5]. Although the harmonic excitation test has used as a standard dynamic test for a long time due to its high reliability as well as due to the fact that it does not require any computational efforts, it requires a lot of expensive test equipment and takes a considerably long time for preparation. The random excitation test is faster than the harmonic test, it needs almost the same test equipment and preparations except for the phase angle meter which is not required for such a test. However complicated computational efforts are required [3,4].

Harmonic and random excitation techniques involve the use of a vibrator which creates artificial test conditions that are in general different from actual conditions [3].

The impulse excitation test is the fastest and simplest method of excitation. The required test equipment is much simpler than that used in other methods [6]. The impulse excitation technique gives excitation across a broad frequency range and the upper limit of this range can be controlled to suit the particular test by varying the material, mass and speed of the hammer head. Therefore it is frequently possible to investigate the whole range of interest in a single test[4,8,9,10]. In reference [7] Kwiatkowsky draw attention to the fact that the cutting forces invariably contains a random component in addition to the average force and this suggested to the authors the possibility of using this component for the determination of the transfer function under cutting conditions. On the other hand the random and impulse techniques can be used in standstill or cutting conditions.

The impulse excitation technique can be analyzed using the bases of random signal analysis to eliminate the effect of noise encountered in the measured signals [1,10]. The development of (FFT) analyzers make such tests fast and accurate where the transfer function can be calculated and plotted. On other hand (FFT) analyzers are too expensive for some research centers to buy it.

The aim of the present work is to develop a low cost computerized system based on a Pc computer which can be used for the determination of the transfer function of machine tools in both stand-still or under cutting conditions.

THEORY

As shown in Figure (1), the transfer function (H(f))is defined as the ratio of the Fourier transform of the system output v (t) to the system input u(t) [3,4,5,9,10,13] hence:

$$H(f) = V(f)/U(f) \qquad (1)$$

where :

V(f) : F {v(t)} Fourier transform of the system output.
U(f) : F {u(t)} Fourier transform of the system input.

Figure 1. Transfer function for single input single output system.

The transfer function may be determined directly from equation (1). However owing to the noise which affects the measured signals better results are obtained in practice by computing the transfer function as the ratio the cross spectrum between the input and the output to the power spectrum of the input signal [1,2,9] as follows:

$$H(f) = G_{uv}(f)/G_u(f) \qquad (2)$$

where:
G_{uv} (f): U (f). V(f), cross spectrum between u(t),v(t).
G_u (f) : U(f) U^* (f), power spectrum of u(t).
U^* (f) : complex conjugate of U(f).
Equation (2) is insensitive to extraneous noise on the force signal. From Figure (2). where m(t) and n(t) represent noise at the input and output signals respectively. The measured transfer function H1(f) is given by the following equation [2]:

$$H1(f) = \frac{G_{uv}(f) + G_{un}(f) + G_{mv}(f) + G_{mn}(f)}{G_u(f) + G_{um}(f) + G_{mn}(f) + G_m(f)} \qquad (3)$$

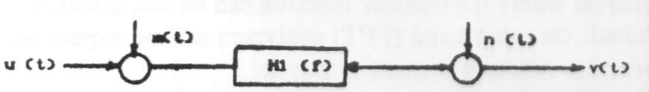


Figure 2. Effect of noise on the transfer function for single input single output system.

In case the measured noise functions m(t) and n(t)are not coherent and not related to each other or to the input signal,then the expected value of the cross spectrum terms involving m,n is equal to zero in equation (3), yielding:

$$H1(f) - \frac{G_{uv}(f)}{G_u(f) + G_m(f)} \qquad (4)$$

$$H1(f) = H(f)/[1 + (G_m(f)/G_u(f))] \qquad (5)$$

The noise to signal ratio is important to determine the degree of accuracy for calculating of the true transfer function H(f). The determination of H(f) using the cross spectrum technique allows to the computation of the coherency function between the input and output signals. The coherency function is defined as follows [1,8,9,11,13]:

$$g^2(f) \quad \frac{[G_{uv}(f)]^2}{G_u(f) \,.\, G_v(f)} \qquad (6)$$

The coherency function will be equal to unity if there is no measurement noise and the system is approximate linear. The minimum value of the coherence function occurs when two signals are totally unrelated is zero therefore the coherency function is a measure of the contamination of the two signals in terms of noise and nonlinear effect [3,9,10,11,13]. The magnitude of the transfer function (receptance) { A(F)} is given by:

$$A(f) = G_{uv}(f)/G_u(f) \qquad (7)$$

and the phase angle between the input and the output signal $\phi(f)$ is obtained by the following equation:

$$\phi(f) = \tan^{-1}\{I_{uv}(f) / R_{uv}(f)\} \qquad (8)$$

where: R_{uv} (f) and I_{uv} (f) are the real and imaginary parts consisting the cross spectrum function.

COMPUTER AIDED VIBRATION TESTING [CAVT] FOR MACHINE TOOLS

The determination of the transfer function of machine tools or any vibrating system implies the measurement of the resulting vibration signal with respect to the exciting force signal whatever it is and analyzing the measured data. The output results of H(f) depend on the used test equipment and the accuracy of data transformed from time domain to frequency domain. The development of the multi channel frequency analyzers (FFT) makes the results more efficient and reliable under different conditions. However due to the high expenses of such equipment it was decided to develop a low cost system which can be used efficiently for the vibration testing based on the new features of Pc computers.

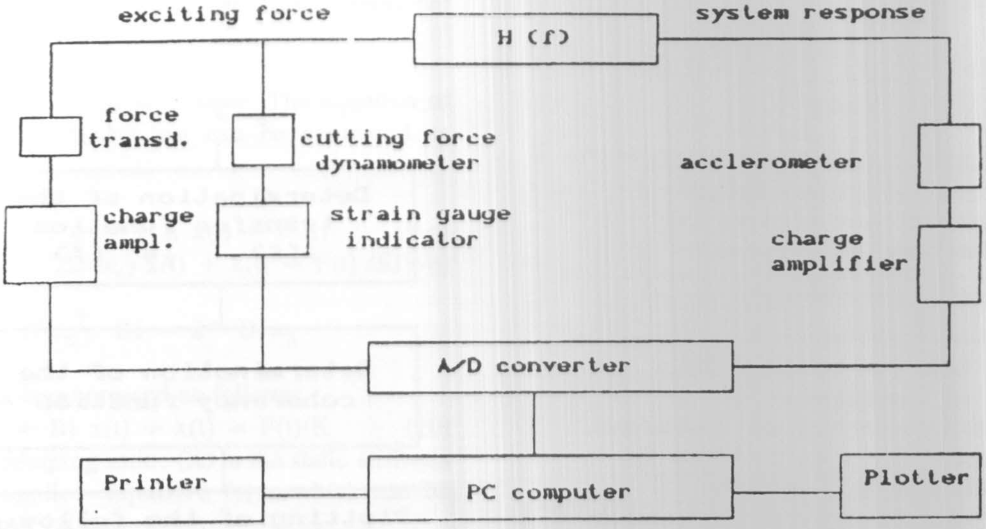


Figure 3. The proposed [CAVT] system.

The developed system [CAVT] consists of the ordinary vibration measuring equipment which can be used for picking up signals such as (force transducers, accelerometers and charge amplifiers) and the analyzing system which consists of Pc computer, a high performance analog to digital converter (A/D converter) to facilitate the conversion of the measured analog signals to digital signals and the associated designed software. Figure (3) shows the proposed [CAVT] system.

CALIBRATION OF THE ANALYZING SECTION

In order to obtain reasonably accurate results using the [CAVT] system, the analyzing section was calibrated to determine relationship between the values of the card counters which control the sampling rates w. r. t. the frequency range of the input signals, as well as the relationship between the input analog signal voltages and the corresponding generated digits.

DETERMINATION OF THE TRANSFER FUNCTION USING THE [CAVT] SYSTEM

As mentioned the [CAVT] system can be used for the determination of the frequency response function for machine tools at standstill or during cutting. Some preliminary tests have to be carried out to determine the system natural frequencies to adjust the sampling rate, the frequency resolution and the recording time.

The exciting force is to be measured using either a force transducer or suitably designed or selected dynamometer corresponding to the type of machining operations. The resulting system response is measured using piezoelectric transducer mounted in the required direction. The force signal is fed to charge amplifier (in case of measuring the force by means of a piezoelectric force transducer) or fed to bridge amplifier (in case of force measurement using strain gauge dynamometers).

The measured signals (force and system response) are fed to two channels of the A/D converter which transform the signals to the corresponding digits required for data in the PC computer. The output results (magnitude and phase) are plotted as functions of the frequency which represents the frequency response function. Figure (4) shows the logic flow diagram.

VERIFICATION OF VALIDITY OF THE [CAVT] SYSTEM

For the sake of [CAVT] system verification for the determination of the transfer function of machine tools, a simply supported beam was selected to determine its dynamic characteristics which can be predicted theoretically. The system under test consists of a beam (diameter 30 mm and 450 mm length) clamped from the rear end in a special fixture. Experimental tests were carried out to check that the system under test can be treated as a single degree of freedom system. The dynamic characteristics of the system under test were calculated theoretically,under the action of impulse and harmonic forces as an exciting forces as well as experimentally using the harmonic excitation test as a standard test. The obtained results were compared with that obtained using the [CAVT] system by means of impulse excitation technique carried out on the same system under test.

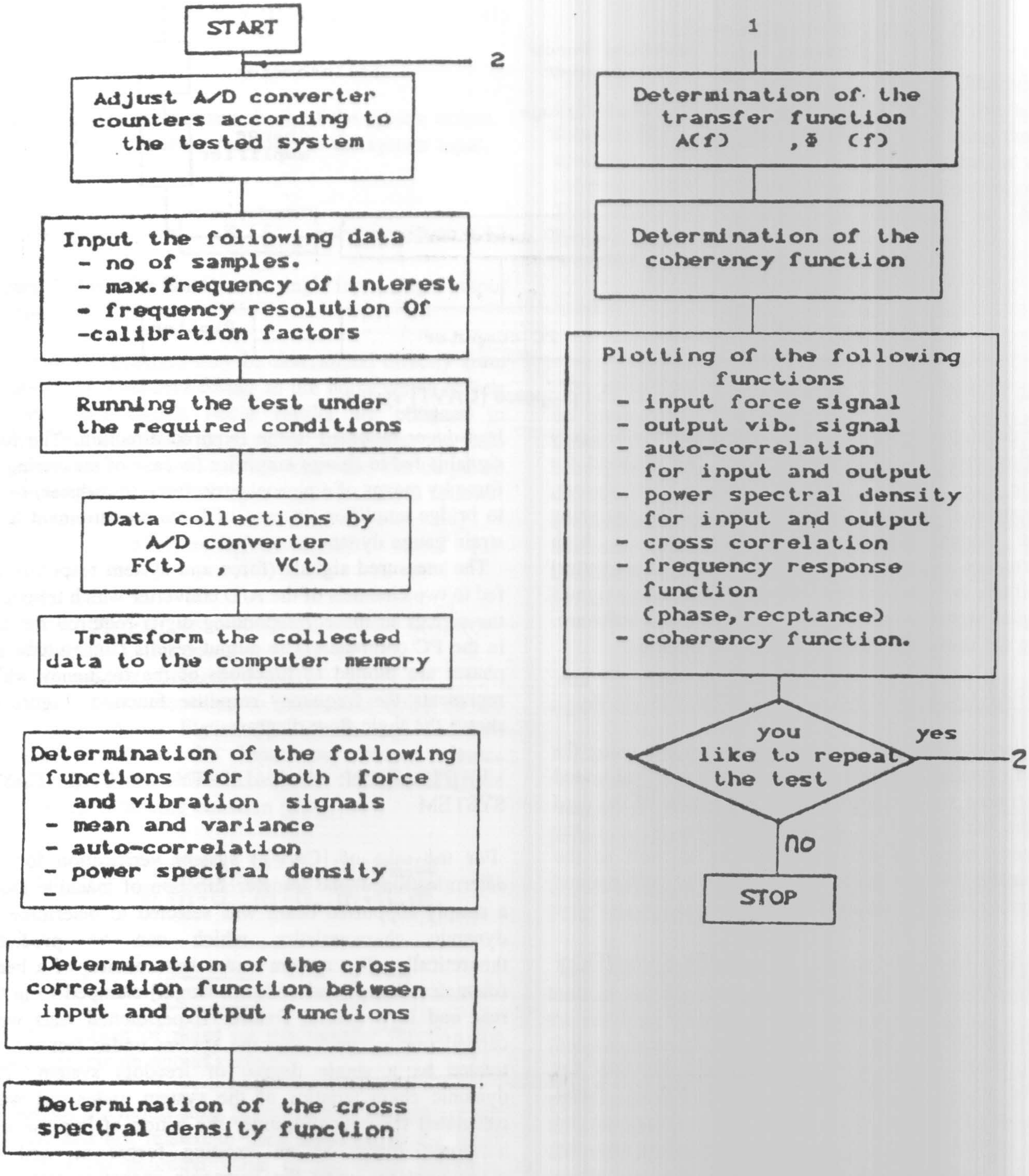


Figure 4. Logic flow chart.

1- *Theoretical determination for the dynamic characteristics*

a) Under the action of impulsive force: The equation of motion of the system under test can be expressed as follows [12,13]

$$m\ x\ (t) + c\ x\ (t) + K\ x(t) = F\ (t) \qquad (9\text{-a})$$
$$(1/w^2_n)\ x\ (t) + (2D/w_n)\ x(t) + x(t) = F(t)\ /K \qquad (9\text{-b})$$

let $\qquad A1 = 1/w_n^2, \ B1 = 2 * D/w_n$

Equation (9-b) can be expressed as follows:

$$A1\ x(t) + B1\ x(t) + x(t) = F(t)/K \qquad (10)$$

where: (D) is the damping ratio, (K) is the static stiffness and (F(t)) is the applied impulsive force which can be expressed as follows [10]:

$$F(t) = 2Fo\ t/T \text{ for } 0 < t < T/2$$
$$= 2Fo\ (1\text{-}t/T) \text{ for } T/2 < t < T$$
$$= 0 \text{ for } t > T$$

where Fo and T are the pulse amplitude and duration respectively which control the frequency content of the pulse, therefore Fo and T must be carefully selected to avoid alyasing and over- sampling problems[9]. Equation (10) can be solved numerically using Runge Kutta method [13] where the second order differential equation is reduced to two first order equations. The calculation interval (Dt), truncation time (Tr) and the number of calculated data (N) are calculated according to the following equations[10]:

$$X(N) = .05\ X(0),$$

$$Dt = 1/\ 2fm, \quad N = T/\ D\,t$$

where (fm) is the maximum frequency of interest = (2.fn) (natural frequency) Figure (5) shows the used impulsive force and the corresponding system response in time domain. In order to calculate the frequency response function for the system under test, the calculated time response is transformed into frequency domain using the random signal analysis [13]. Figure (6) shows the output results.

b) Under the action of harmonic force:In order to obtain the frequency response under harmonic excitation, equation (10)is subjected to a harmonic force { Fo sin (2π f t)}. The solution of this equation consists of the complementary function and particular solution [12]. The magnitude and phase angle with respect to the exciting force may be expressed as follows:

$$A(f) = X(f)/Fo = 1/K\{[1-(f/fn)^2]^2 + (2.D.f/fn)^2\}^{0.5} \qquad (11)$$

$$PHI(f) = \tan^{-1}\ [2\ D(f/fn)/(1-(f/fn)^2] \qquad (12)$$

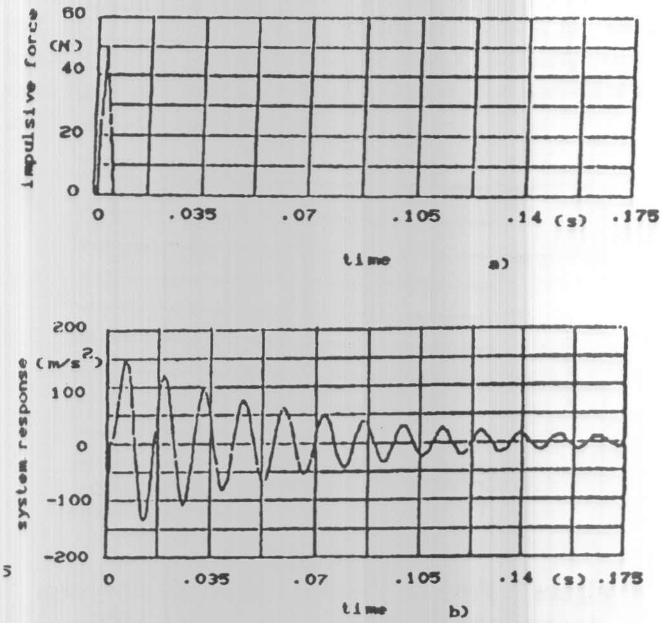


Figure 5. Theoretical time signals using the impulse excitation a) pulse b) system response.

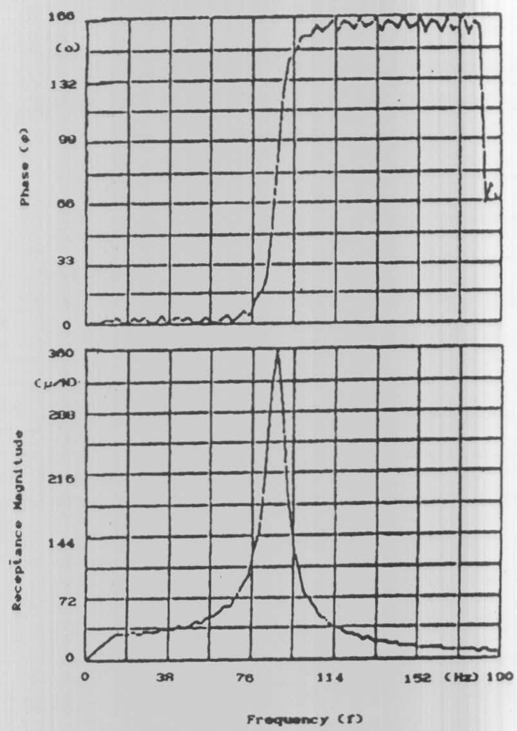


Figure 6. Frequency response curve for the system obtained by impulse excitation technique.

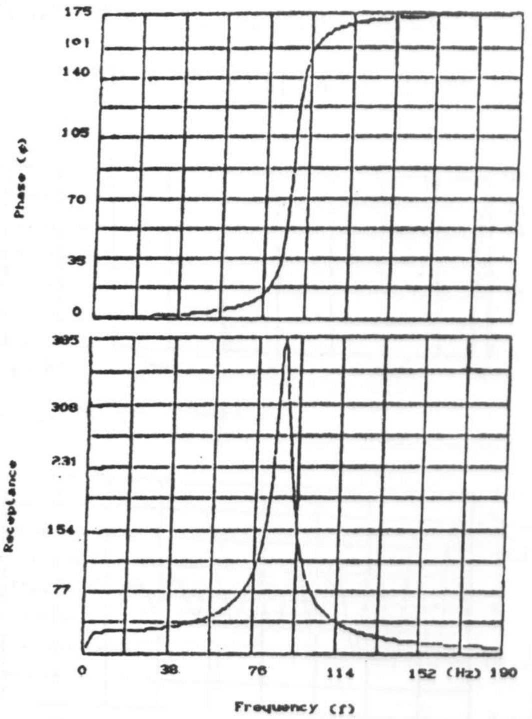


Figure 7. Frequency response curve for the system obtained by harmonic excitation technique theoretically a) Phase b) Receptance.

The exciting frequency (f) is to be swept theoretically up to the upper limiting frequency (fm). The frequency response function obtained according the harmonic force is shown in Figure (7). Before the determination of the frequency response function theoretically under the action of impulsive or harmonic excitation forces, some values such as static stiffness (K), natural frequency (fn)and the damping ratio (D) must be calculated from preliminary test. The tests results reveal that:

$$K = 0.02 \text{ N/um}$$

$$fn = 86 \text{ Hz}$$

$$D = 0.05$$

2- *Experimental determination for the dynamic characteristics*

a) Using harmonic excitation test:A standard excitation test must be carried out on the system under test. The harmonic excitation test represents one of the most traditional technique by which the frequency response function can be directly obtained [3].

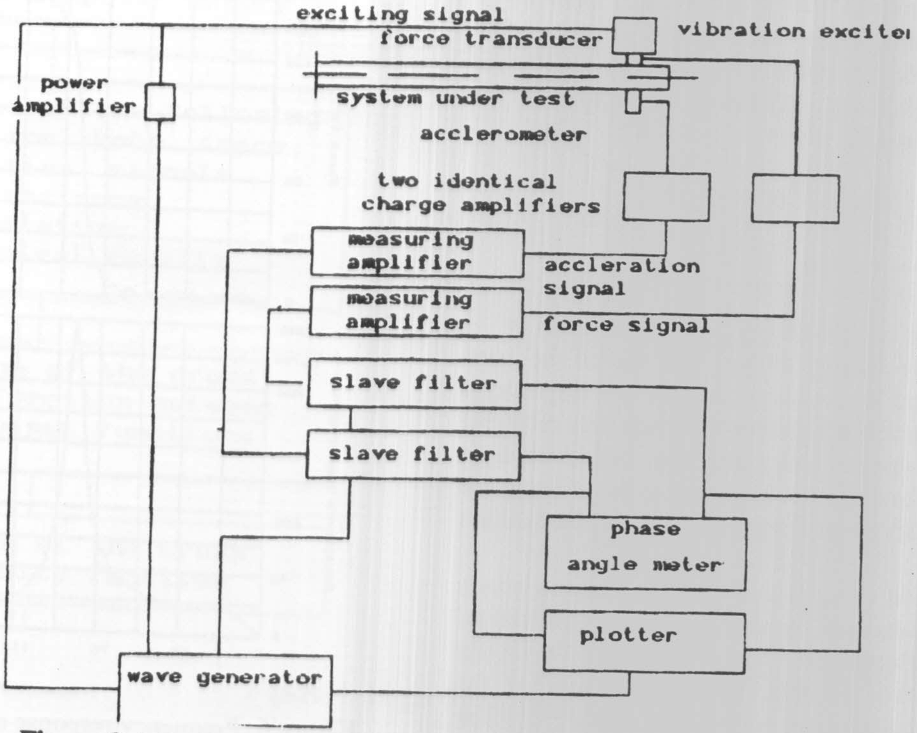


Figure 8. Test equipment used for the harmonic excitation technique.

The test equipment used in the harmonic test is shown in Figure (8). the wave generator generates a sinusoidal signal, the frequency range of which is swept automatically. The signal is fed to shaker. The generated force signal is measured by means of a piezoelectric force transducer. The resulting system vibrations are measured by means of an accelerometer. Both of the force transducer and accelerometer output signals are fed to two identical charge amplifiers and then to two measuring amplifiers. In order to keep a constant amplitude of the exciting force, a direct feed back of the force signal is fed back to the wave generator. Two identical slave filters are used to remove noise from both the force and the response signals to enable phase angle measurement using phase angle meter. The filtered displacement signal is plotted against frequency using an X-Y plotter. Figure (9) shows the obtained frequency response function.

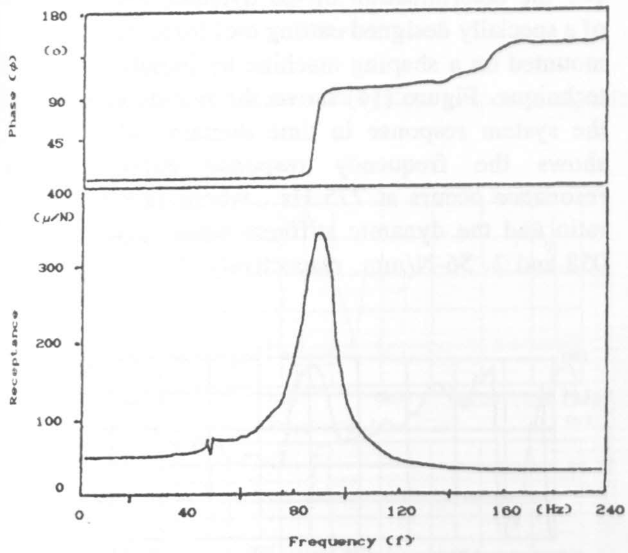


Figure 9. Frequency response curve obtained by the harmonic excitation technique.

b) Using the [CAVT] system:The impulsive force was measured using a piezoelectric force transducer mounted on the hammer,while the resulting vibrations were measured using a piezoelectric accelerometer mounted on the object of the force and vibration signals were fed via two identical charge amplifiers and to the computer system. Preliminary test had been carried out to determine the system natural frequency in order to adjust the sampling rate of A/D converter card. The impulsive force and the system response in time domain are shown in Figure (10) while the corresponding frequency response curve for the tested beam is shown in Figure (11).

COMPARISON BETWEEN THE OBTAINED RESULTS

Table (1) shows the comparison between the obtained results for the dynamic characteristics of the system under test. It is observed that the results are almost the same hence it can be concluded that the [CAVT] system can be used for the determination of dynamic characteristics of machine tools satisfactorily.

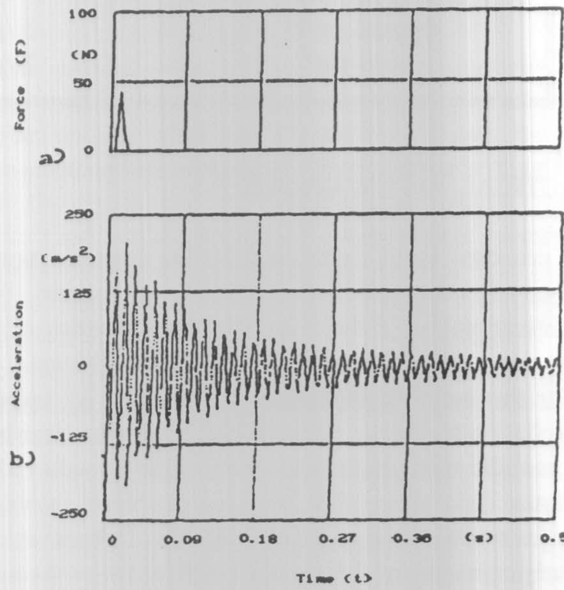


Figure 10. Impulsive force and system response for the system under test. a) impulsive force b) system response.

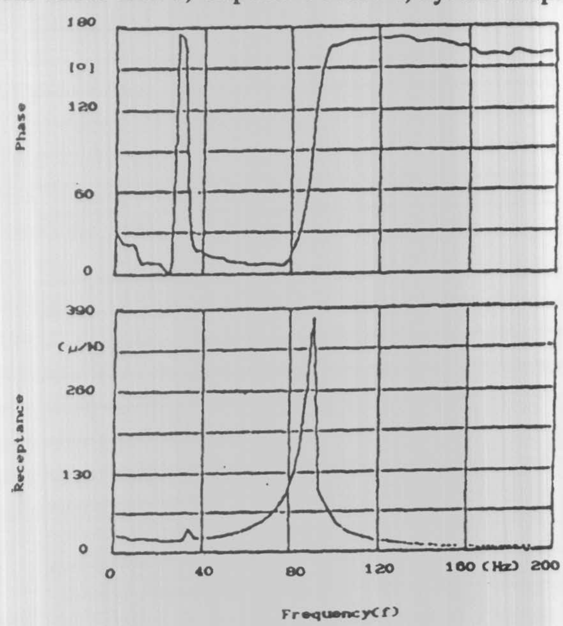


Figure 11. Frequency response curve for the system under test obtained by the [CAVT] system.

Table 1. Comparison between the obtained parameters.

Parameter	Units	Methods of Excitation			
		Theoretical		Experimental	
		harmonic	impulse	harmonic	by CAVT
natural freq.	HZ	86	86	82	85
damping ratio	--	0.053	0.053	0.058	0.55
dynamic stiff	N/μ	0.00273	0.00273	0.0027	0.00268
static stiff	N/μ	0.25	0.25	0.23	0.26

PRACTICAL APPLICATIONS

The proposed system was used for the determination of the dynamic characteristics of the following cases:

1- Determination of the dynamic characteristics of the machine tool elements in stands tell conditions:

a- For the main spindle of a column drill by impulse excitation technique.: Figure (12) shows the impulsive force and the system response in time domain. while Figure (13) shows the frequency response curve,the main resonance occurs at 350 Hz., where as the damping ratio and the dynamic stiffness were found to be 0. 032 and 1. 69 N/mm. respectively.

b- For the determination of the dynamic characteristics of a specially designed cutting tool force dynamometer mounted on a shaping machine by impulse excitation technique. Figure (14) shows the impulsive force and the system response in time domain. while Fig 15 shows the frequency response curve,the main resonance occurs at 775 Hz., where as the damping ratio and the dynamic stiffness were found to be 0. 058 and 2. 56 N/mm. respectively.

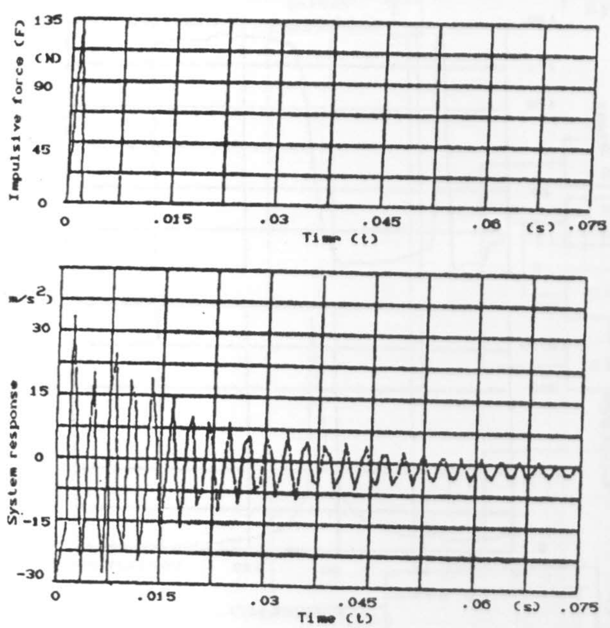


Figure 12. Impulsive force and system response signals.

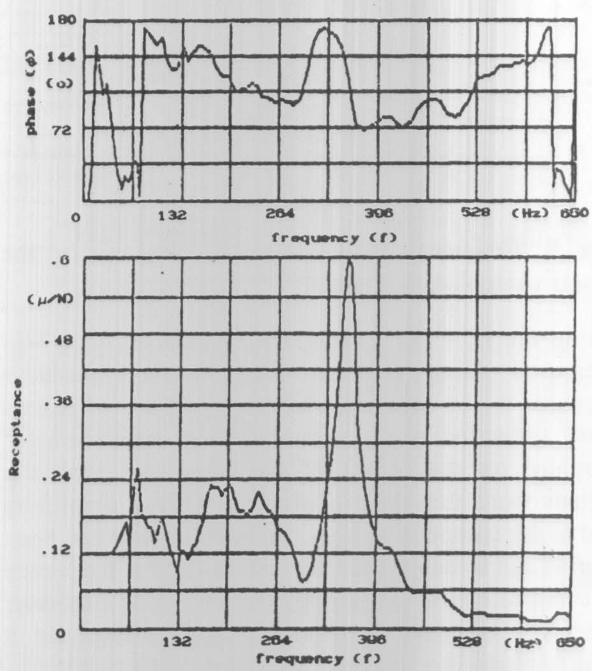


Figure 13. Frequency response curve for the main spindle of the column drill.

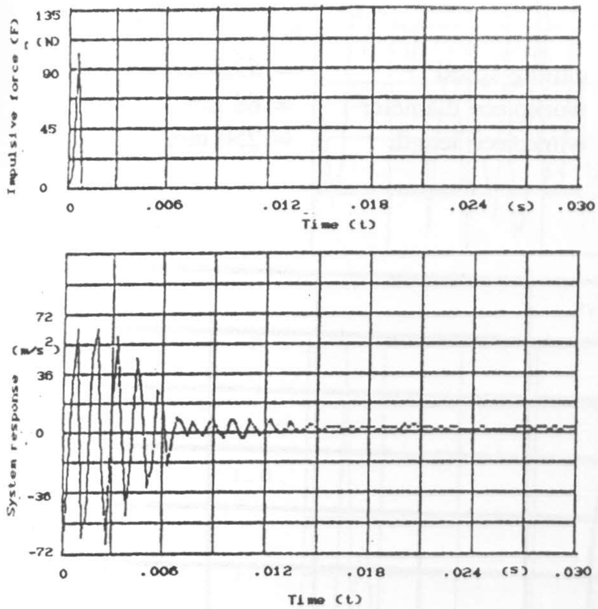
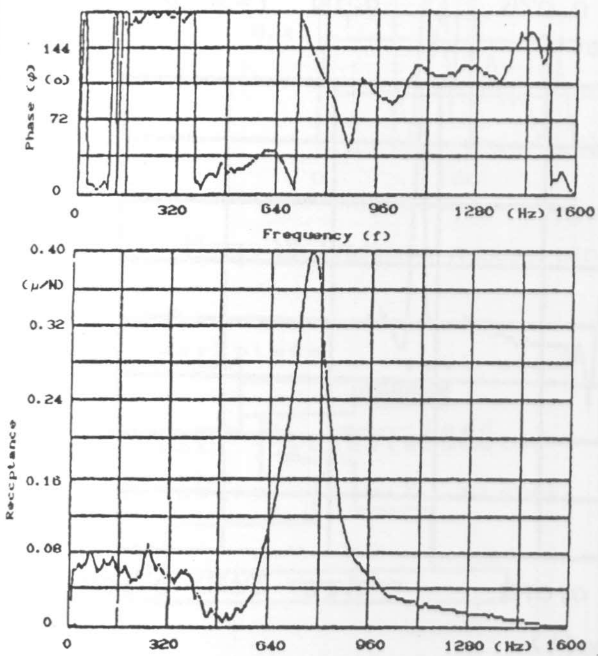


Figure 14. Impulsive force and system response signals.



Frequency (f)

Figure 15. Frequency response curve. for the dynamometer used in shaping machine.

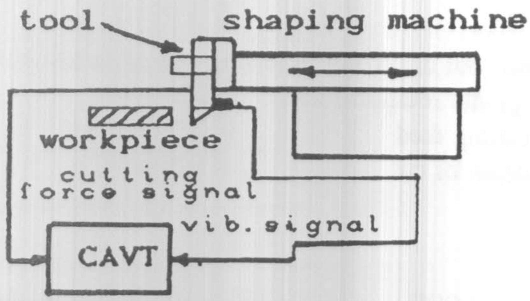


Figure 16. The used test equipment.

2- The determination of the dynamic characteristics of the machine tool elements under cutting conditions;

a- For shaping machine: Figure (16) shows the used measuring test equipment. The exciting cutting force in the direction of cut was measured using a suitably designed cutting force tool dynamometer mounted in place of the cutting tool on the machine tool holder. The resulting vibration in the same direction was measured using piezoelectric accelerometer. The cutting force signal was fed to strain gauge bridge amplifier while the accelerometer signal was fed to charge amplifier. Both of the force and vibration signal were fed to the [CAVT] system. Figure (17) shows the cutting force and the system response signals, while Figure (18) shows the frequency response curve. The test was carried out under the following cutting conditions:

Ram speed	= 48 stroke/min
Cutting speed	= 8. 6 m/min.
Feed	= 0. 25 mm/stroke
Depth of cut	= 1 mm
Tool overhang	= 120 mm
Tool cross sect.	= 16 x 16 mm

b- For turning machine Figure (19) shows the used measuring test equipment. The exciting cutting force in the direction of cut (in case of longitudinal turning)was measured using a suitably designed strain gauge cutting force tool dynamometer mounted in place of the cutting tool on tool holder. The resulting vibration in the same direction was measured using piezoelectric accelerometer. The cutting force signal was fed to strain gauge bridge amplifier while the accelerometer signal was fed to charge amplifier. Both of the force and vibration signal were fed to the [CAVT] system. Figure (20) shows the cutting force and the system response signals, while Figure (21) shows the frequency response curve. The test was carried out under the following cutting conditions:

spindle rotational speed	= 250 rpm	cutting speed	= 53.4 m/min
cutting feed	= 0.2 mm/rev	workpiece diameter	= 68 mm
depth of cut	= 1.5 mm	workpiece length	= 250 mm

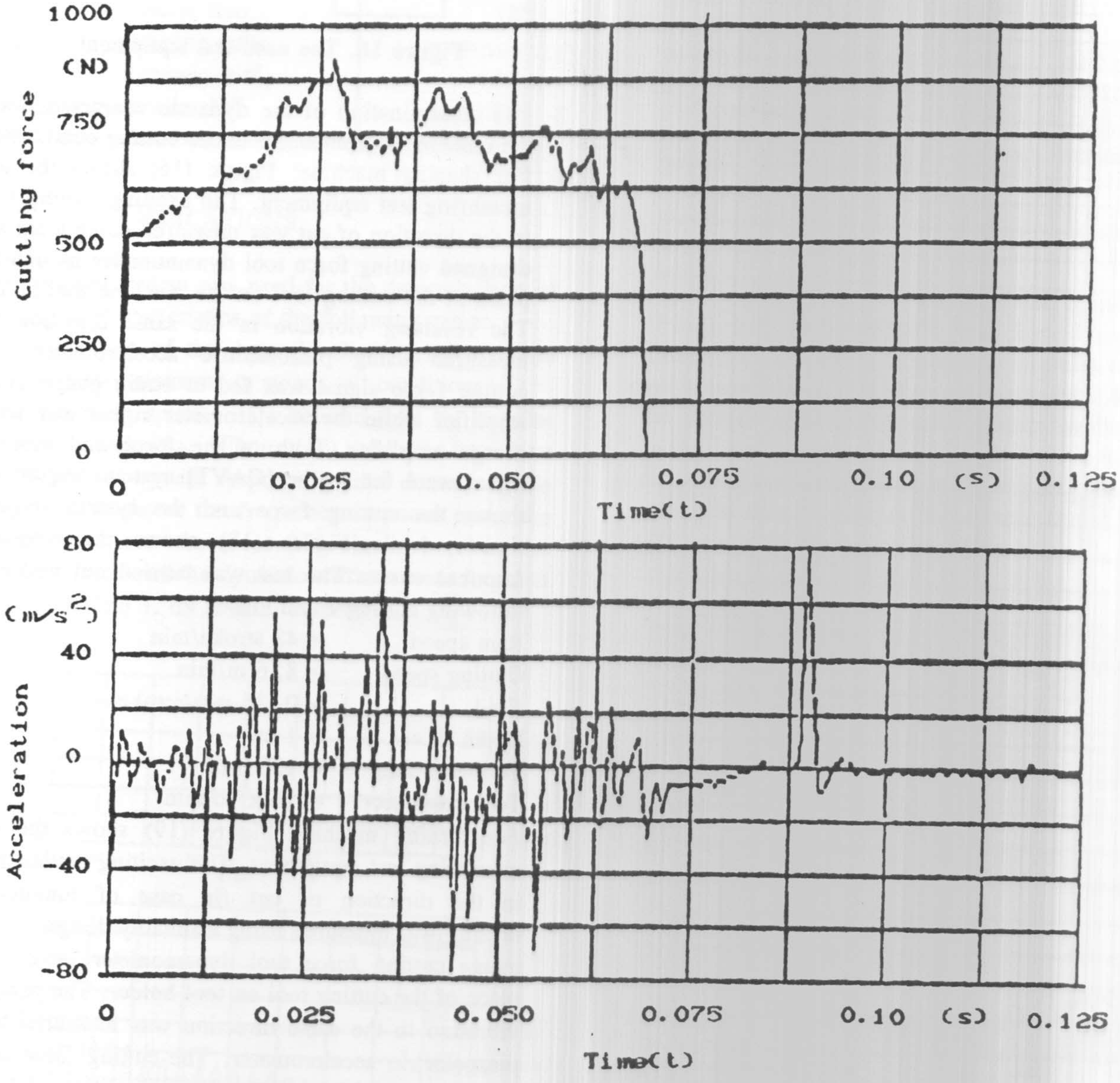


Figure 17. The cutting force and the vibration signals.

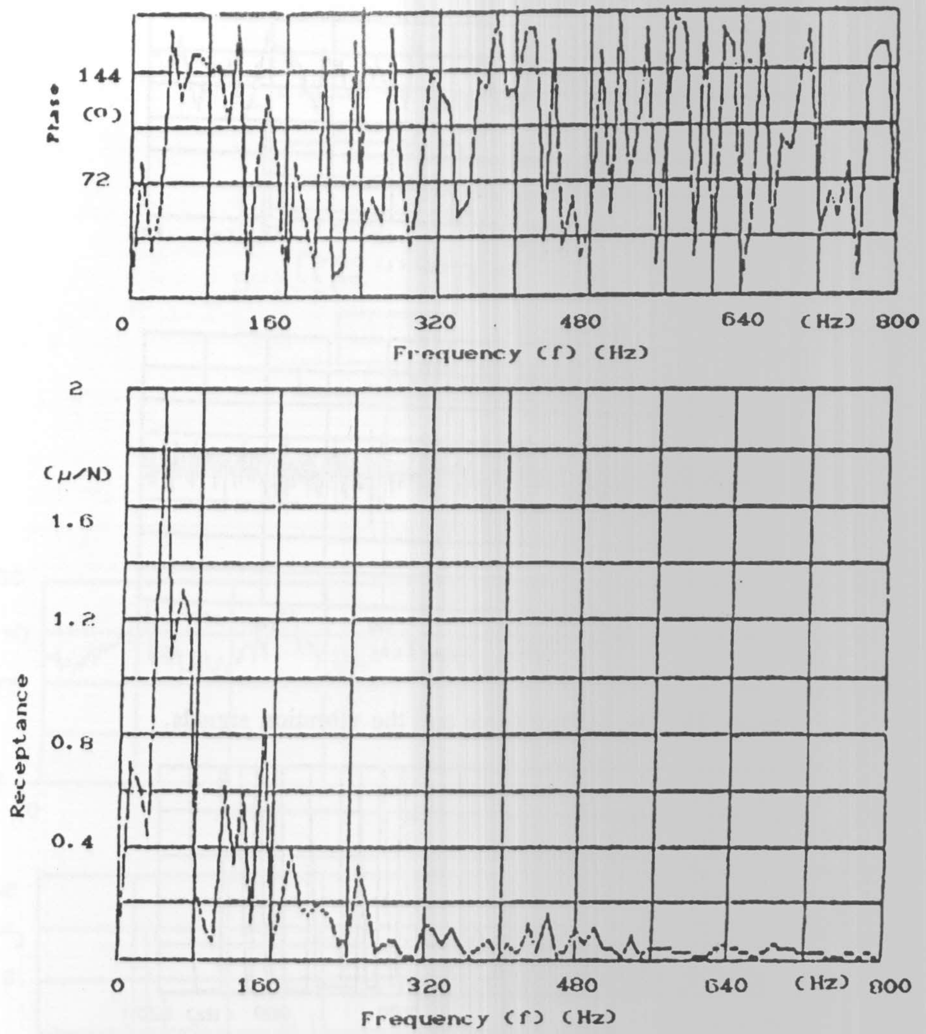


Figure 18. Frequency response curve under cutting conditions in shaping machine.

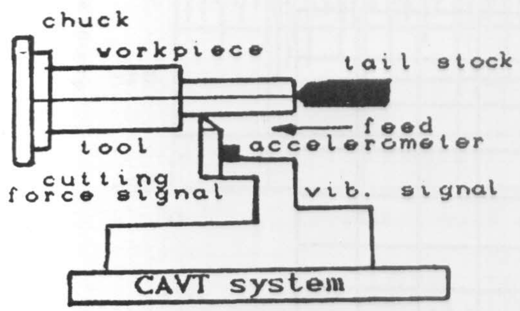


Figure 19. The used test equipment.

c- For turning machine Figure (22) shows the used measuring test equipment. The exciting cutting force in the direction of cut (in case of cross turning)was measured using a suitably designed piezoelectric cutting force tool dynamometer mounted in place of the cutting tool on tool holder. The resulting vibration in the same direction was measured using piezoelectric accelerometer. The cutting force signal and the accelerometer signal were fed to two charge amplifiers. Both of the force and vibration signal were fed to the [CAVT] system. Figure (23) shows the cutting force and the system response signals, while Figure (24) shows the frequency response curve. The test was carried out under the following cutting conditions:

spindle rotational speed	= 250 rpm
cutting feed	= 0.2 mm/rev
depth of cut	= 1.5 mm
cutting speed	= 53.4 m/min
workpiece diameter	= 68 mm
workpiece length	= 250 mm

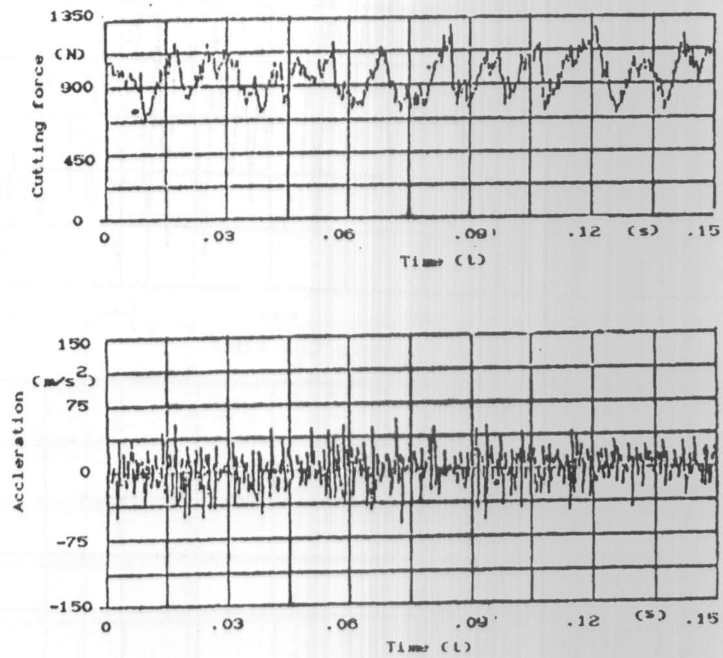


Figure 20. The cutting force and the vibration signals.

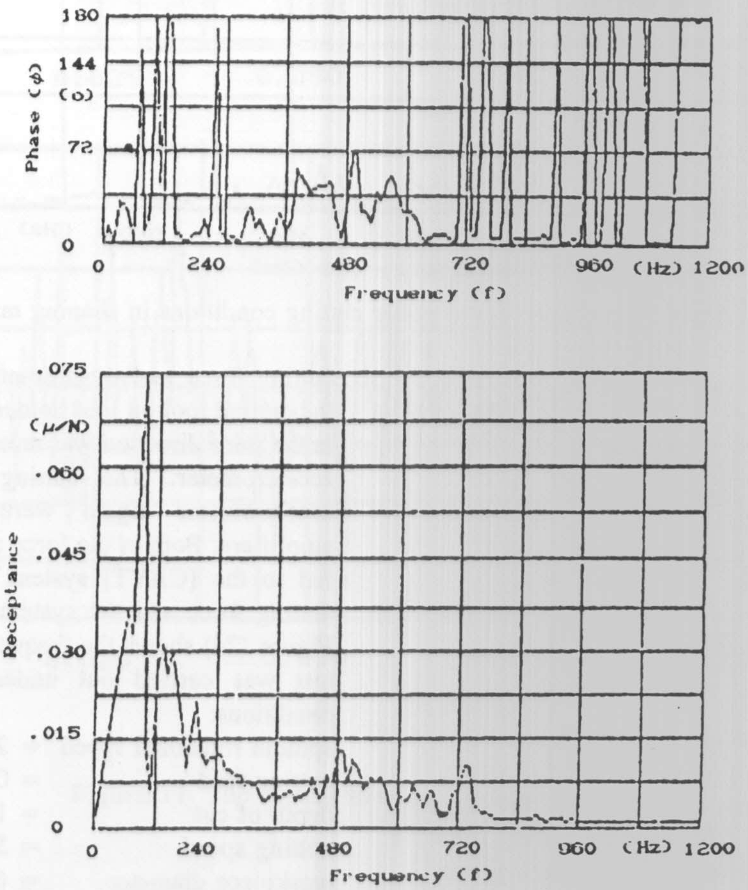


Figure 21. Frequency response curve. under cutting conditions in cross turning.

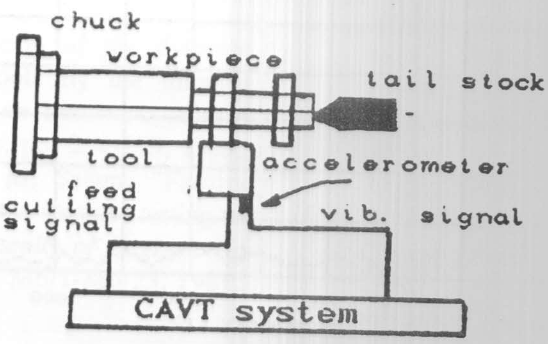


Figure 22. The used test equipment.

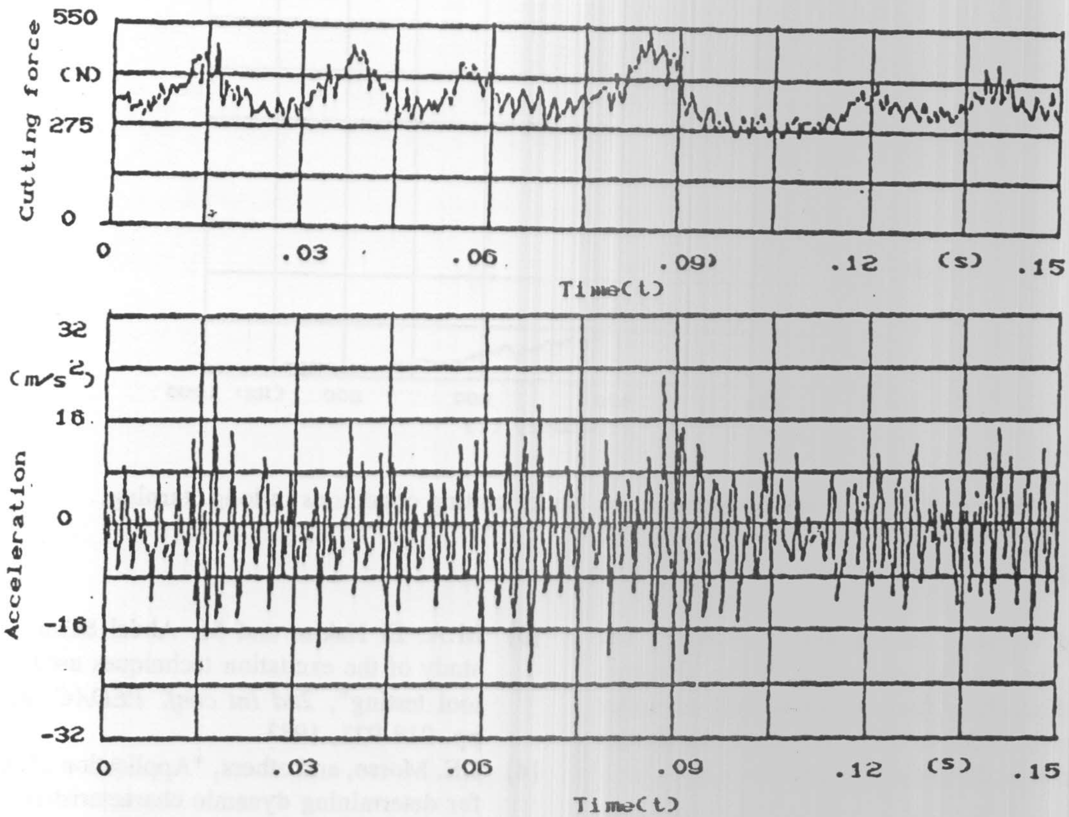


Figure 23. The cutting force and the vibration signals in case of long. turning.

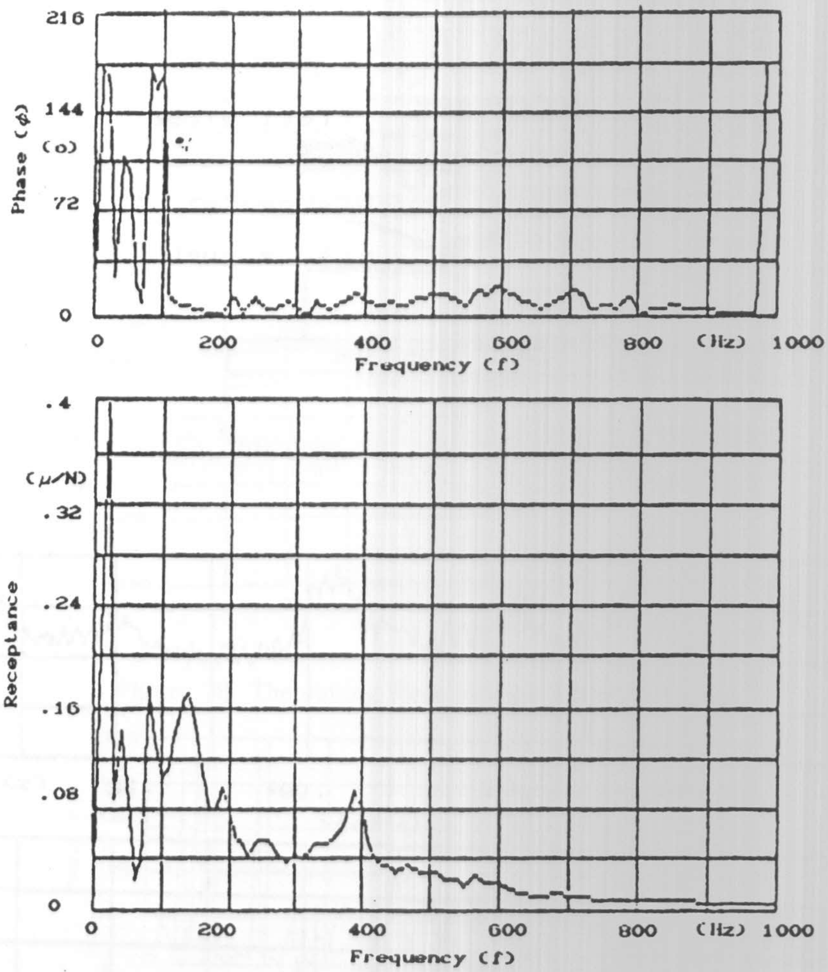


Figure 24. Frequency response curve. under cutting conditions in long. turning.

CONCLUSIONS

A computerized vibration testing system [CAVT] has been developed for the determination of the dynamic characteristics of machine tools in stand still or under cutting conditions in order to solve vibration and chatter problems encountered during machining.

REFERENCES

[1] W.G. Halverson and D.L. Brown, "Impulse technique for structural frequency response testing", *Sound and vibration*, pp. 8-21, 1977.

[2] P. Cowley, "The accuracy of frequency response function measurements using FFT - based analyzers with transient excitation", *Trans. ASME*, Vol. 108, pp. 44-49, 1986.

[3] M.A. El Hakim and M. Abdel Salam, "A critical study of the excitation techniques used for machine tool testing", *2nd Int conf. PEDAC*, Alex. Univ., pp. 918-923, 1983.

[4] I.E. Morse, and others, "Application of pulse testing for determining dynamic characteristics of machine tools", *MTDR conf.*, Vol. 13, pp. 1-11, 1972.

[5] C.F. Field, "Cross correlation and milling machine dynamics, A case study", *Int J. MTDR*, Vol 9, pp. 81-96, 1969.

[6] N.H. Hanna and A.W. Kwiatkowsky, "Identification of machine tool receptances by random force excitation", *Int. J. MTDR*, Vol. 11, pp. 309-325, 1971.

[7] A.W. Kwiatkowsky, and others, "Identification of milling machine receptances from random signal

during cutting", *Annals of CIRP.*, Vol XVI, pp. 81-96, 1969.

[8] M.A. El Hakim, "Identification of the dynamic characteristics of machine tools by the impulse technique", *Proc. 5th. conf. on automatic control and system engineering*, Cairo, pp. 264-275, 1979.

[9] M.A. El Hakim and Abdel M. Salam, "Factors affecting the exciting force in the impulse testing of structures", *Bulletin of the Faculty of Engineering Ain Shams Univ.*, No. 12, pp. M6/1-M6/23, 1981.

[10] M.A. El Hakim and M. Abdel Salam, "Some application notes on the impulse testing of machine tools", *Bulletin of the Faculty of Engineering Ain Shams Univ.*, No. 12, pp. M7/1-M7/20, 1981.

[11] *Utilization of HP Fourier analyzer for impulse testing*, Application notes, Hewlett Packard co., AUG., 1972.

[12] T.W. Thomson, *Vibration theory and applications*, George Allen and Unwin, LTD, London, 1966.

[13] D.E. Newland, "An introduction to random vibrations and spectral analyses", Longman 1976.