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Modal Analysis and Experimental Determination of Optimum Tool Shank Overhang of a Lathe Machine

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Abstract: Vibration of Tool Shank of a cutting tool has large influence on tolerances and surface finish of products. Frequency and amplitude of vibrations depend on the overhang of the shank of the cutting tool. In turning operations, when the tool overhang is about 2 times of the tool height, the amplitude of the vibration is almost zero and dimensional tolerances and surface finish of the product becomes high. In this paper, the above statement is verified firstly by using a finite element analysis of the cutting tool with ANSYS software package and secondly, with experimental verification with a piezoelectric sensor. *Copyright © 2008 IFSA.*

Keywords: Modal testing, Tool shank, Piezoelectric sensor, Signal conditioner, Analyzer

1. Introduction

Vibration is a common problem of almost all types of machining, and particularly when the tool overhang exceeds about 2 times the tool depth. It becomes significant when the requirements of dimensional tolerances and surface finish are high. A stable tool holder is of the utmost importance for increasing the precision of the machining process.

Generally in turning operations there will be some level of relative dynamic motion between cutting tool and work piece. Energy from the chip formation process excites the mechanical modes of the machine-tool system. Modes of the work piece may also influence tool vibration. The dynamic properties of the excitation, i.e. the chip formation process are correlated to the material properties and the geometry of the work piece. The vibrations may lead to unwanted noise, degraded surface finish and reduced tool life.

The dynamic performance of mechanical systems is mostly represented by their modal parameters. The modal parameters which govern the dynamic performance of the mechanical systems can be identified by modal testing which is usually used to study the mechanical systems. The majority of traditional modal testing techniques include excitation of the structure via a function of known frequency characteristics (impulse, random, periodic) using, e.g. an impact hammer or an exciter, and measuring the associated response using force or motion sensors. Finally, a description of system dynamic characteristics is obtained by establishing frequency response functions (FRF).

In this paper both theoretical and experimental analyses of the tool shank have been performed. Theoretical investigation deals with the designs and characteristics of the tool shank. The first step is to develop a mathematical model using Finite Element Method (FE-Model) of the problem using reliable finite element software, ANSYS to get an understanding of the dynamics involved. In order to verify the result of the model an Experimental Modal Analysis (EMA) is done. Based on the experimental results the FE-model is adjusted and when the FE-model and the experimental model correlates accurately the FE-model of the tool holder shank is accepted. The agreement between FE-models and the experimental models gives opportunity to use the FE-models for more extensive studies without further extensive experimental measurements.

2. Literature Review

The theory of chatter vibrations is known for a long time. The first [1] studies about self-excited vibrations have been started in the second half of the past century. Tlusty and Polacek [2] and Tobias [3] determined the most important source of self-excitation which is associated with the structural dynamics of the machine tool and the feedback between the subsequent cuts on the same cutting surface resulting in regeneration of waviness on the cutting surfaces, and thus modulation in the chip thickness [2].

Although chatter stability has been studied in detail in last half a century, chatter vibrations still continue to be one of the most important limitations in production operations. Shi and Tobias [4] showed that the boundaries of the stability increases as the feed rate increases until a nominal value. Important contributions about chatter stability came from Budak and Altintas [5] and, Jensen and Shin [6] in recent years.

Tlusty [7] pointed out that the flank wear flat is critical in positive damping in the occurrence of self-excited vibrations. Chiou et al. [8] demonstrated that chatter instability is delayed to a greater overhang distance as a result of flank wear, and chatter limit increases especially at lower cutting speeds, as the tool wear increases. Chiou and Liang [9] demonstrated the effect of tool wear on chatter stability in turning. The chatter stability increases as the tool wear flat of the cutting tool enlarges.

Chiou and Liang [10] analyzed the acoustic emission in chatter vibration with tool wear effect in turning. Fofana et al. [11] investigated machining stability in turning by using worn tool inserts. Cutting forces varying with depth of cut and feed rate and cutting force coefficients are investigated as the tool wear progresses and it is demonstrated that tool wear and dynamic instability are both contributed by the combined effect of the contact and friction mechanisms between work piece-tool, tool-chip and work piece-tool-machine tool interactions.

3. The Tool Shank

The tool shank that is used for the experimental procedures has a cross section of 20 x 20 mm and the length is 140 mm. The geometry of the tool shank is presented in Fig. 1 and Table 1. Particularly for this experiment we used carbon steel as shank materials and the tip is tungsten carbide.

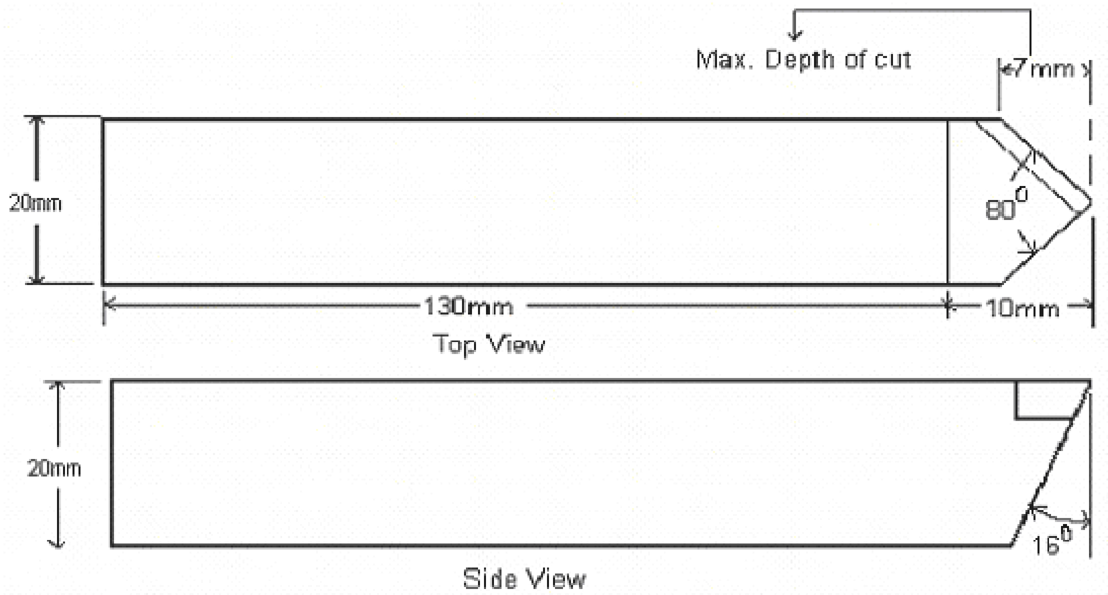

Fig. 1. The geometry of the tool shank.

Table 1. Specification of tool shank.

Tool No.		Max. Depth of cut	Tip No.	Tool Dimensions			
0^0 Top Rake	R.H. Lipped			Height	Width	Length	Nose radius
60*	62	7 mm	60*	20 mm	20 mm	140 mm	1.5 mm

4. The Theoretical Model

The shank of a cutting tool is generally analyzed for strength and rigidity. This is assumed to be loaded as a cantilever by tool forces at the cutting edge as shown in Fig. 2. The main design criterion for shank size is rigidity. The deflection at the cutting edge is limited to a certain value depending on the size of the machine, cutting conditions and tool overhang. The Tool overhang (Le) is related also to the shank size as well as to end fixing conditions. It is seen that only below (Le/H) of 2, the amplitude is practically zero. The recommended value of (Le/H) is between 1.2 and 2.

Knowing the chatter frequency f and the shank deflection can be calculated from the well-known equation (12),

$$f = 15.76/\sqrt{\Delta} \text{ c.p.s.}, \quad (1)$$

where Δ is deflection, in mm.

The shank deflection is given by the equation,

$$\Delta = P_z L_e^3 / 3EI , \quad (2)$$

where P_z is applied force in Newton;
L_e is overhang length in mm;
E is modulus of elasticity;
I is the area moment of inertia of the shank.

The force P_z for a given size of lathe, is given by

$$P_z = s \times t \times k , \quad (3)$$

where s is feed in mm,
t is depth of cut in mm,
k is cutting force constant.

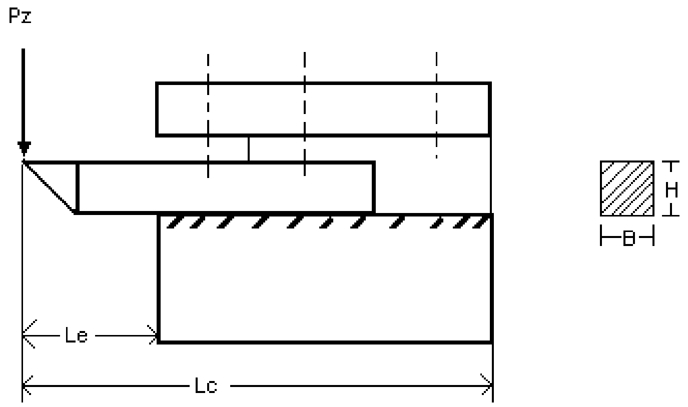

Fig. 2. Cutting tool overhang.

One of the most important choices when making a finite element model is which element type to use. There are many different types of element that we can choose. These elements are beam element, shell element and solid element. In this paper, finite element modeling is done by using ANSYS software and the modeling was done by using solid element. Beam elements can be used to make a very efficient finite element model to predict overall deflection and bending moments but will not be able to predict the local stress concentrations at the point of application of a load or at joints. A beam has three transitional degrees of freedom and three rotational degrees of freedom. The most general elements are the solid elements. The disadvantage is that more elements and nodes are usually required. However, many devices designed by engineers have highly three-dimensional geometry, which makes it necessary to use three-dimensional elements.

4.1. FE-model

The boundary condition includes loads, temperature restraints, and displacement restraints. The FE-model is built up with solid elements. To simulate the boundary conditions the FE-model is clamped at 100 mm, 80 mm, 60 mm and 50 mm hang out respectively. The finite element models show the natural frequencies of different overhang length (Fig. 3-6).

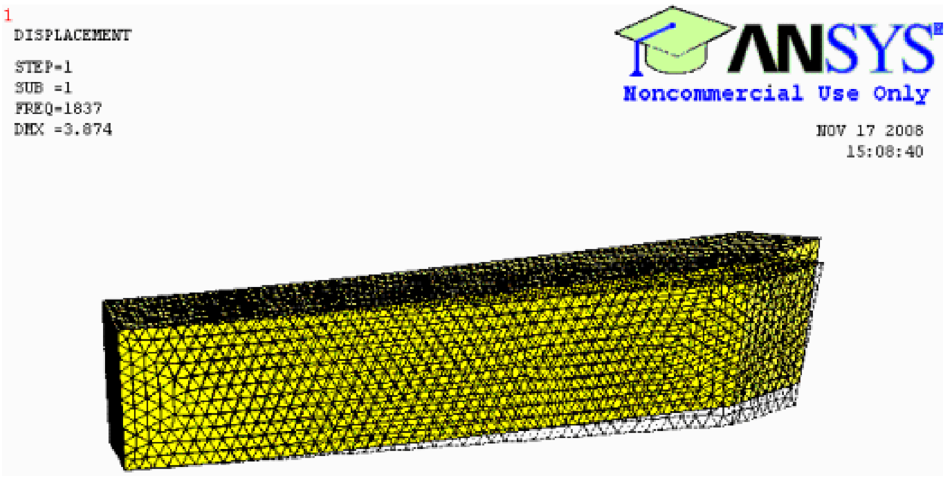

Fig. 3. Tool shank overhang 100 mm.

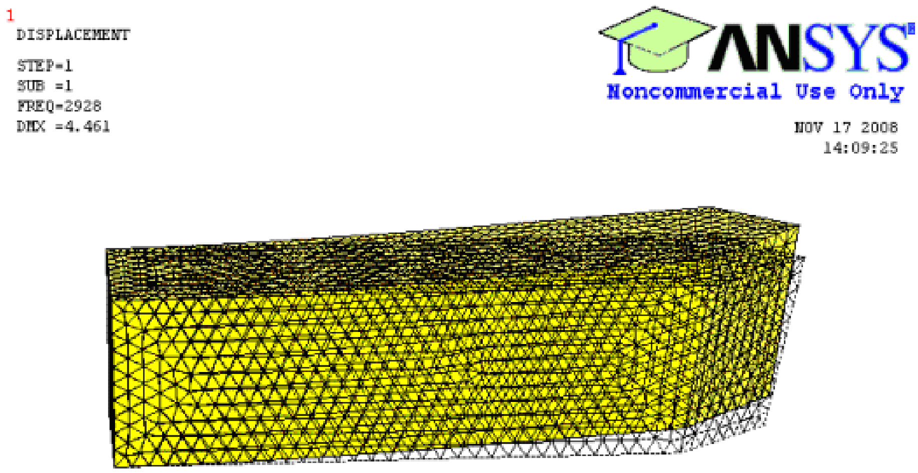

Fig. 4. Tool shank overhangs 80 mm.

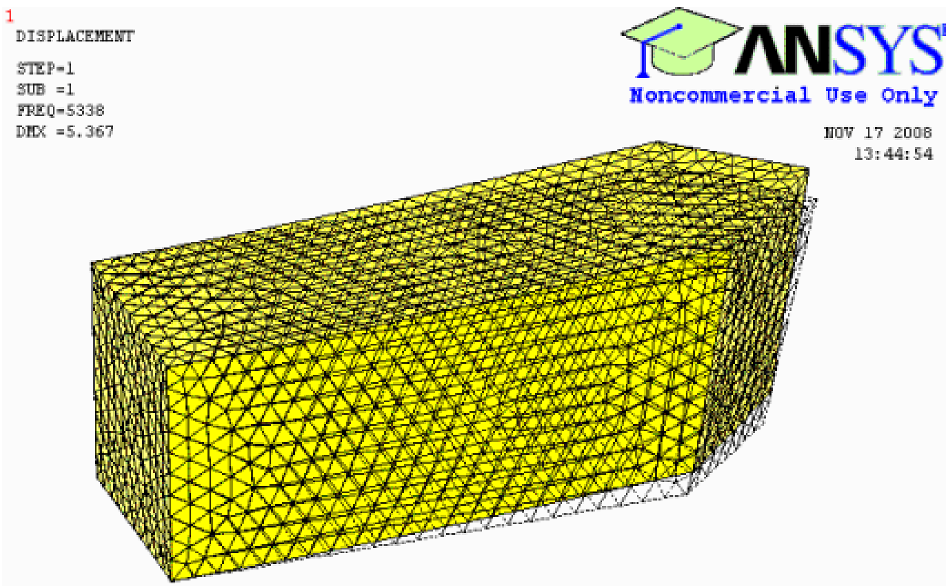

Fig. 5. Tool shank overhang 60 mm.

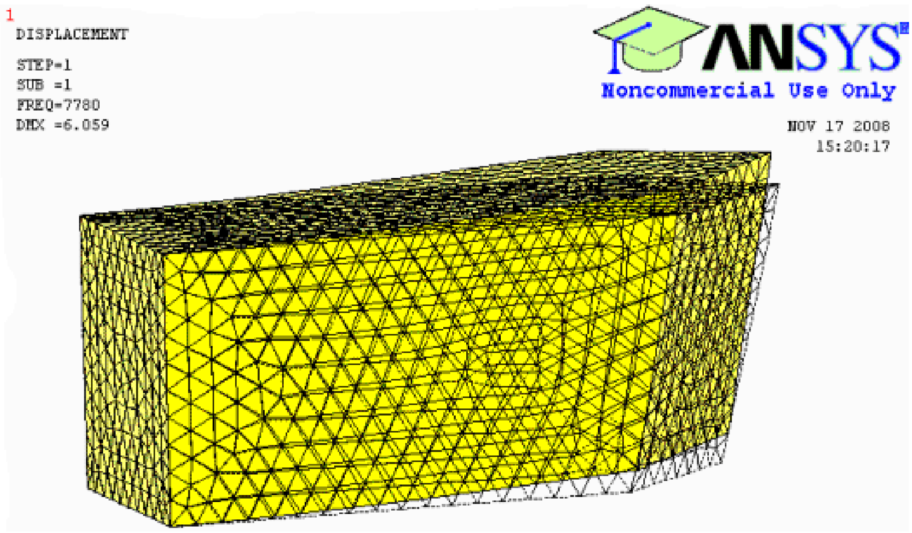

Fig. 6. Tool shank overhang 50 mm.

5. Experimental Setup

5.1. Vibration Exciters

The vibration exciter or shaker can be used in several applications such as determination of the dynamic characteristics of machines and structures and fatigue testing of materials. The vibration exciters can be mechanical, electromagnetic, or hydraulic type.

One of the most commonly used equipment in producing excitation forces is the electromagnetic exciter. Fig. 7 depicts the schematic diagram of an electromagnetic exciter. It consists of a coil over a central core of permanent magnet. The coil can be moved freely with respect to the magnet. When a sinusoidal signal is applied to the coil, because of electromagnetic interaction, the coil will move in a sinusoidal fashion with respect to the coil. If the coil is attached to any structure then the structure will in turn be forced to move sinusoidally.

Fig. 7. Electromagnetic Exciter.

5.2. Piezoelectric Accelerometer

An accelerometer is an instrument that measures the acceleration of a vibrating body. Accelerometers are widely used for vibration measurements and to record earthquakes. The accelerometer can be used for measuring acceleration, velocity, and amplitude of an oscillating body. Accelerometers are secured as firmly as possible to the body of which the movements are to be investigated. The most commonly used type of sensor for vibration measurements is the piezoelectric acceleration sensor. As the spring constant is very high, the resonant frequency of such devices is of the order of 10000 Hz or higher. The force to which the crystal is subjected is a linear function of the position of the moving body relative to the fixed body, as the crystals are deformed elastically.

Piezoelectric accelerometers as shown in Fig. 8 operate on the principle that a mass moves will stresses a piezoelectric crystal. Acceleration forces cause the amount of applied force on the crystal to vary cyclically and this, by the very nature of the piezoelectric effect, produces an electric output signal proportion to compression. Since this stress is it directly related to acceleration, such a signal is proportional to the vibratory acceleration of the surface on which it is mounted. Since the electric output is charges, a charge amplifier has to be used to convert the signal to voltage signal.

Fig. 8. Piezoelectric accelerometer.

Specification of the piezoelectric accelerometer is given below:

Vibration Sensor:	Piezoelectric accelerometer 100 mV/g with magnetic base, probe and 5 foot cable to BNC
Display:	LCD 3.5 digit, measurement, hold, low battery indicator
Measurement:	Acceleration 0.01-19.99 g (RMS)
Range:	Velocity 0.01-19.99 in/sec., 0.1-199.9 mm/sec (RMS)
	Envelope 0.01-19.99 ge (peak)
Frequency Range:	Overall 10 Hz - 10 kHz
	Envelope 0.5 kHz-10 kHz
Output:	Sensor Excitation: 12 Vdc @ 2 mA (BNC)
	Audio Out: 3.5 mm mini plug; 250 mW into 8 Ohms, 150 mW into 32 Ohms;
	Adjustable volume control with off position
Power:	(2) "AA" cells
Operating Time:	20 hours continuous without headphones
Weight: Instrument:	7 oz. (0.19 kg); Complete Kit: 2.85 lb (1.30 kg)
Dimensions:	6.3 x 3.3 x 1.25" (152 x 83 x 32 mm)
Operating conditions:	-14° to 122°F (-10° to 50°C)

5.3. Signal Conditioner

Signal conditioning means manipulating an analogue signal in such a way that it meets the requirements of the next stage for further processing. More generally, signal conditioning can include amplification, converting, and any other processes required to make sensors output suitable for conversion to digital format. Since the output impedance of accelerometers is not suitable for direct input in the signal analysis equipment, signal conditioners, is in the form of charge or voltage amplifiers, are use to match and amplify the signals before signal analysis. It is primarily utilized for data acquisition, in Fig. 9 which sensor signals must be normalized and filtered to levels suitable for analog-to-digital conversion so they can be read by computerized devices.

Fig. 9. Signal conditioner.

Specification of signal conditioner is given below:

Model 480E09,
1-channel, battery-powered, ICP® sensor signal conditioner, gain x1, x10, x100, BNC input/output connection.

Frequency Range (-5 %)(x1, x10 Gain)	0.15 to 100,000 Hz
Voltage Gain (± 2 %)	1:10
Excitation Voltage (To Sensor)	27 to 29 VDC
Internal Battery (Type)	9V
Electrical Connector (Input, sensor)	BNC Jack

5.4. Analyzer (Pico Scope 2202)

An ***analyzer*** (Fig. 10) is used to perform the tasks of signal processing and modal analysis using suitable software. The response signal, after conditioning, is sent to an analyzer for signal processing. A commonly used analyzer is called a Fast Fourier Transform (FFT) analyzer. Such an analyzer receives analog voltage signals (representing displacement, velocity, acceleration, strain, or force) from a signal conditioning amplifier, filter, and digitizer for computations. The analyzed signals can be used to find the natural frequencies, damping ratios, and mode shapes either in numerical or graphical form. The analyzer converts the analog time-domain signals, x(t), into digital frequency-domain data using Fourier series relations given by Eqs. (4) to (7), to facilitate digital computation. Thus the analyzer accepts the analog output signals of accelerometers or force transducers, x(t), and computes the spectral coefficients of these signals a_0, a_n and b_n using above equations in the frequency domain. The process of converting analog signals into digital data is indicated in figure below for two representative signals. In figure x(t) denotes the analog signal and $x_i = x(t_i)$ represents the corresponding digital record, with t_i indicating the ith discrete value of time. This process is performed by an analog-to-digital (A/D) converter, which is part of a digital analyzer.

If x(t) is a periodic function with period τ , its Fourier's Series representation is given by

$$\begin{aligned} x(t) &= \frac{a_0}{2} + a_1 \cos \omega t + a_2 \cos 2\omega t + ... + b_1 \sin \omega t + b_2 \sin 2\omega t + ... \\ &= \frac{a_0}{2} + \sum_{n=1}^{\alpha} \left(a_n \cos n\omega t + b_n \sin n\omega t \right) \end{aligned} \tag{4}$$

where the digital spectral coefficients a_0, a_n, b_n are given by

$$a_0 = \frac{2}{N} \sum_{i=1}^{N} x_i \; ; \tag{5}$$

$$a_n = \frac{2}{N} \sum_{i=1}^{N} x_i \cos \frac{2n\pi t_i}{\tau} ; \tag{6}$$

$$b_n = \frac{2}{N} \sum_{i=1}^{N} x_i \sin \frac{2n\pi t_i}{\tau} \tag{7}$$

Fig. 10. PicoScope.

Specification of PicoScope 2202:

Vertical Resolution:	8 bits
Analog Bandwidth:	2 MHz
Maximum Sampling Rate:	Single-channel-20MS/s Dual-channel-10 MS/s per channel
Buffer Size:	Dual-channel-16k samples per channel
Inputs:	2 channels with BNC sockets.
Voltage Ranges:	± 50 mV to ± 20 V ranges in 9 ranges.

Accuracy:	3 % (voltage) 100 ppm (time)
Operating Environment:	Temperature range- 0°C to 70°C. Humidity-25% to 75% RH
Overload Protection :	± 100 V.
PC Connection:	USB 2.0 Compatible with USB 1.1.
Power Supply:	From USB port- 4.6 to 5.25 V; 500 mA.
Maximum Dimensions:	140 mm x 190 mm x 45 mm.

The choice of experimental excitation method depends on several factors, such as the geometry and shape of the structure. It took several tests to establish a method to use for our application. The structure excited by using an exciter connected with an impedance head by a flexible drive rod**,** Fig. 11. The exciter gives rise to a vibrating excitation signal that is transferred into the structure by an impedance head. The impedance head is an accelerometer and it measures the acceleration in the defined point. The requirements for the excitation method were that the frequency range amount to 10 kHz and that sufficient energy are still in the system at least up to 2.4 kHz. These requirements were thought to be fulfilled by both excitation methods at the beginning, but after several tests it became clear that the shaker excitation only has sufficient energy up to approximately 1 kHz. The problems when measuring with the exciter is that it is very important to mount the impedance head in the centre of the tool shank to avoid undesired movements and moments into the structure. Another problem is the mass spring system of the impedance head that could affect the natural frequencies of the system.

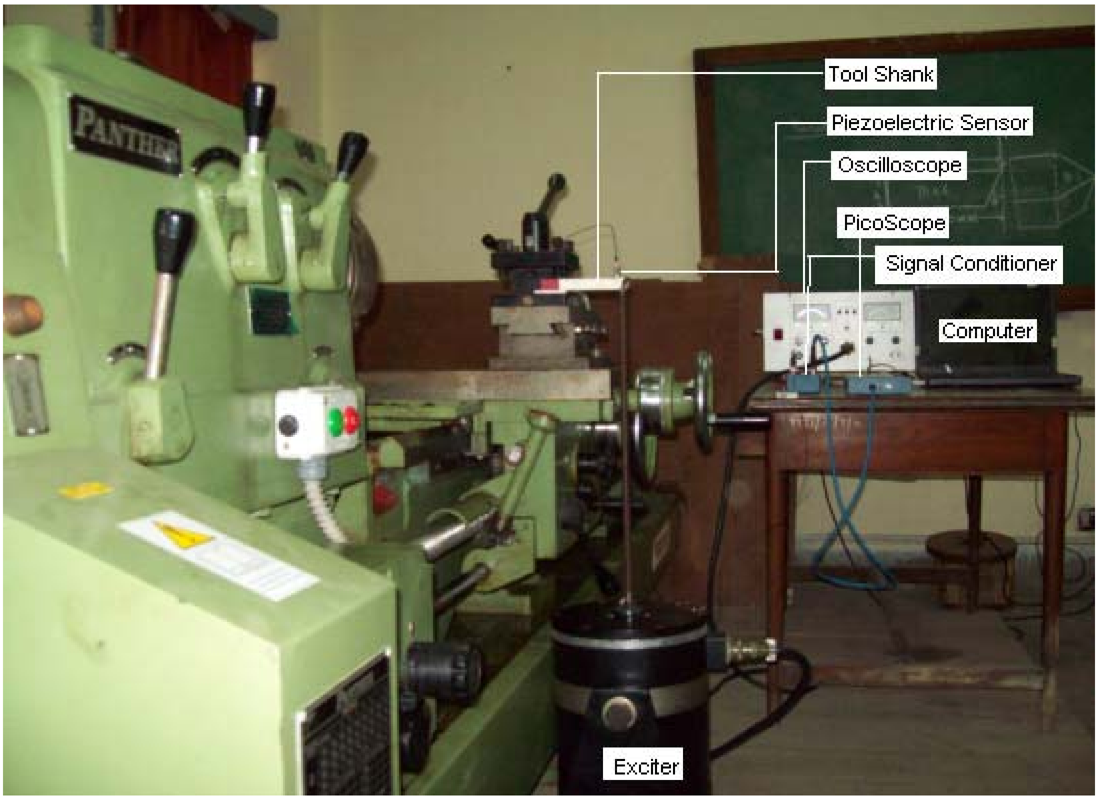

Fig. 11. Experimental set-up.

Frequency Response Function Curves at different overhang length are shown in Fig. 12-15.

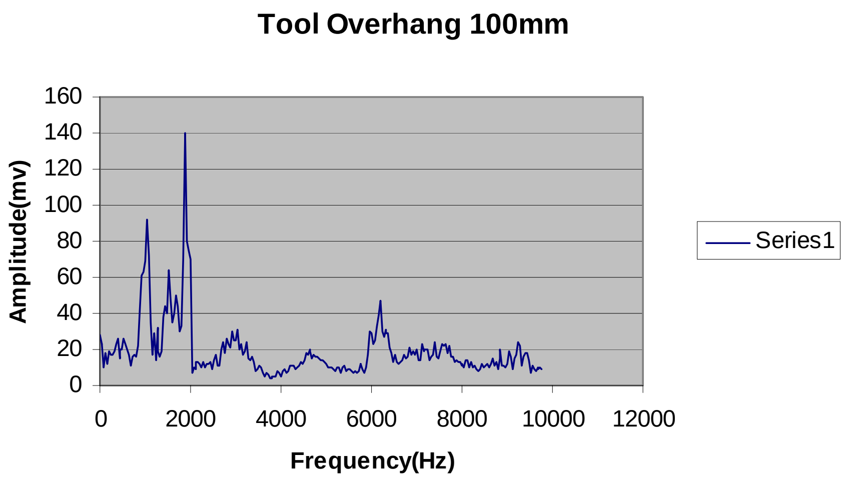

Fig. 12. Tool overhang of 100 mm.

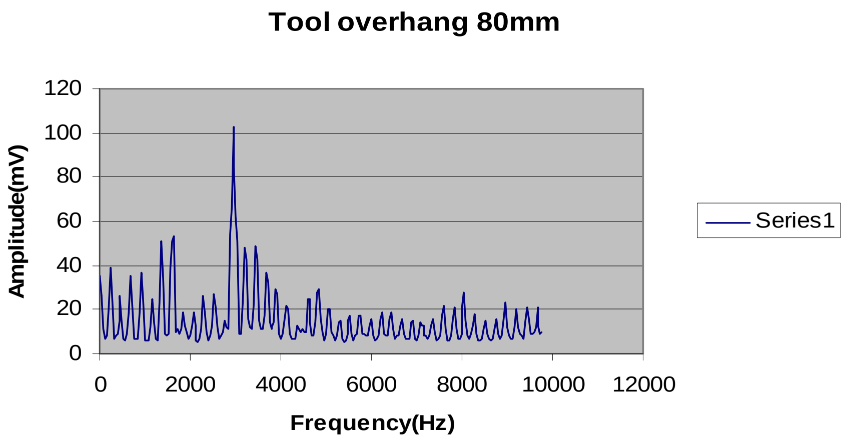

Fig. 13. Tool Overhang of 80 mm.

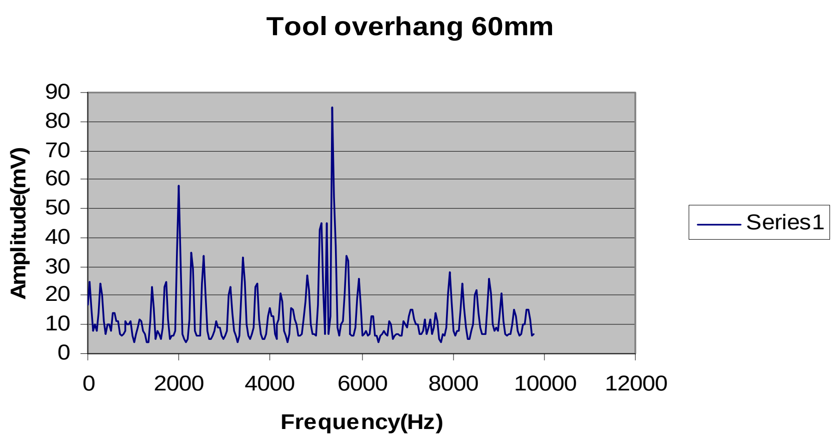

Fig. 14. Tool overhang of 60 mm.

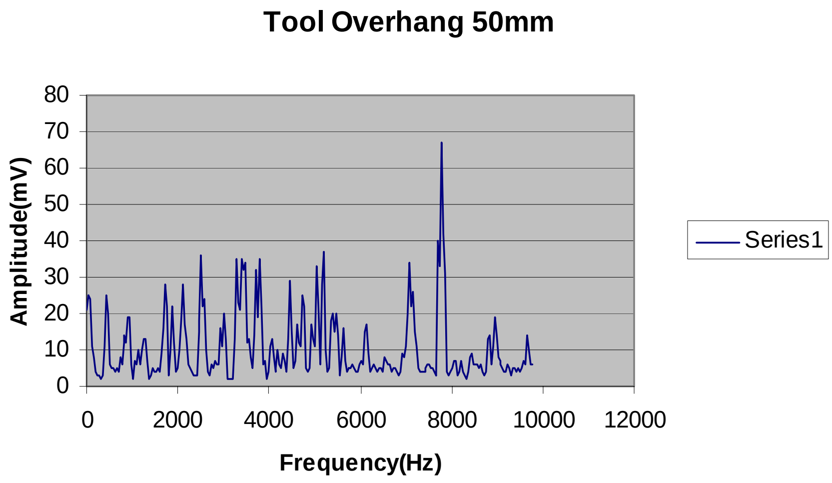

Fig. 15. Tool overhang of 50 mm.

6. Results and Conclusions

Tools shanks were positioned in different overhang, 100 mm, 80 mm, 60 mm and 50 mm respectively. Firstly, Finite Element Analysis of the Tool shank was performed using ANSYS software and frequency and Amplitude were found out in different overhang positions. Secondly, Experimental measurement of frequency and amplitude of the Tool shank were taken at defined overhang position. In Frequency Response Function (FRF) curves, in which the vertical y-axis shows the magnitude of the out put signal in volts (mV) and in the x-axis is the frequency interval f (Hz) shown in Figs. 12-15. The greater the amplitude, the stronger is the effect of vibrations. Fig. 16 shows the curve at different overhang length with the corresponding natural frequency of tool shank.

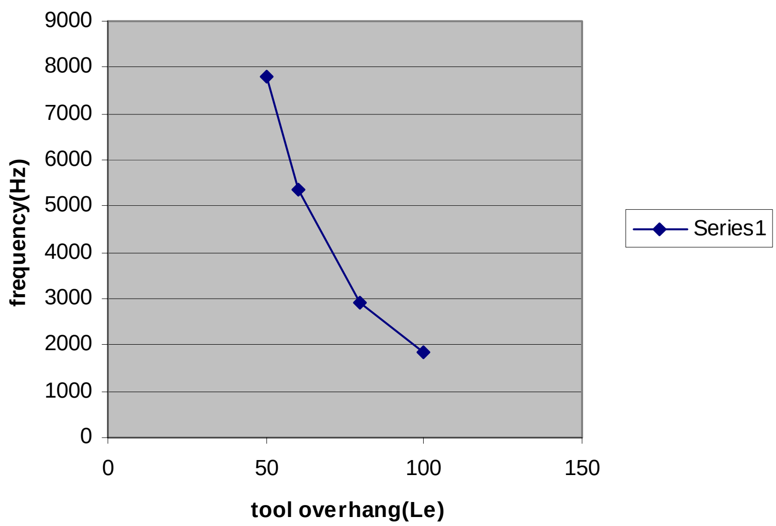

Fig. 16. Relation between frequency with tool overhang.

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Guide for Contributors

Aims and Scope

Sensors & Transducers Journal (ISSN 1726-5479) provides an advanced forum for the science and technology of physical, chemical sensors and biosensors. It publishes state-of-the-art reviews, regular research and application specific papers, short notes, letters to Editor and sensors related books reviews as well as academic, practical and commercial information of interest to its readership. Because it is an open access, peer review international journal, papers rapidly published in *Sensors & Transducers Journal* will receive a very high publicity. The journal is published monthly as twelve issues per annual by International Frequency Association (IFSA). In additional, some special sponsored and conference issues published annually.

Topics Covered

Contributions are invited on all aspects of research, development and application of the science and technology of sensors, transducers and sensor instrumentations. Topics include, but are not restricted to:

- Physical, chemical and biosensors;
- Digital, frequency, period, duty-cycle, time interval, PWM, pulse number output sensors and transducers;
- Theory, principles, effects, design, standardization and modeling;
- Smart sensors and systems;
- Sensor instrumentation;
- Virtual instruments;
- Sensors interfaces, buses and networks;
- Signal processing;
- Frequency (period, duty-cycle)-to-digital converters, ADC;
- Technologies and materials;
- Nanosensors;
- Microsystems;
- Applications.

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Articles should be written in English. Authors are invited to submit by e-mail editor@sensorsportal.com 6-14 pages article (including abstract, illustrations (color or grayscale), photos and references) in both: MS Word (doc) and Acrobat (pdf) formats. Detailed preparation instructions, paper example and template of manuscript are available from the journal's webpage: http://www.sensorsportal.com/HTML/DIGEST/Submition.htm Authors must follow the instructions strictly when submitting their manuscripts.

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