Study Dynamic Characteristics of Milling Tool Holder by Experimental and Theoretical Work

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***Abstract*: Vibration problems often occur in manufacturing processes during metal cutting. vibration level depends upon many parameters such as design of machine and tool structure, material of work piece and tool and fixture of (tool – tool holder) structure. As a result of vibration, the tool life is reduced, noise appears, and the surface is poorly finished. In this paper an assessment of dynamic characteristics was carried out by experimental modal analysis and Theoretical modal analysis by simulated data using ANSYS program. A comparison was made between the two methods and the difference in the results of both of them was clarified.**

Keywords: Vibration -modal analysis -dynamic characteristics – finite element method – machine tool.

I. INTRODUCTION

In order to obtain a high surface quality and precision of the workpiece, the problems of vibration caused during the cutting operations in the manufacturing operations must be overcome. Hung J et al. Suggested that the machining performance of a machine tools jointly determined by the structure of the machine tool, the tool spindle and the dynamic characteristics of machine tools are expressed in different forms and with different frequency ranges, such as the low-frequency structural mode and the high-frequency instrument mode [1-3]. Occurrence of chattering has an initial occurrence condition. Chatter generates during cutting and its values increase continuously Therefore, it is feasible to discover and control the chatter in the initial period of its occurrence. The machine tool has multiple degrees of freedom vibration system and chatter occurs at the natural frequency of one of the two final execution parts. the system is transformed into a single-degree-of-freedom chatter model with executing components to simplify the model. One or two chatter active bodies can be analyzed according to specific conditions in actual engineering applications. The cutting chatter system can have two chatter active bodies. With different cutting parameters, chatter can occur at different natural frequencies. Therefore, combined with the actual situation at the scene, we should pay enough attention to the optimization of machine vibration resistance and chatter monitoring [4].The holder is an affective parameter in cutting process so that Fleischel et al estimated the dynamic behavior of the whole cutting system consisting of spindle, tool holder, tool and work piece. Therefore modal and operational vibration analyses were performed to describe the damping and operational characteristics of two competing tool holder technologies, namely heat shrink (HS) and hydraulic expansion (HE). A modal and operational vibration analysis on three different machining centers showed that HE technology has better damping properties which implies lower operational vibrations and reduced noise emission compared to HS [5].

By the way Benattia et al analyzed the influence of the perpendicularity of the spindle of the milling machine on the machined surface. The study was conducted by computer simulation tests and experimental part using surface condition monitoring instruments, and showed that the presented model could thus be integrated into systems computer-aided design and computer-aided manufacturing. Finally, the physical and statistical parameters of roughness during milling at position 90 confirmed that when the defect of the perpendicularity is eliminated to the maximum, the best surface conditions are obtained. They Recommended that it is better to use the vertical spindle milling machines as the effect of the angle of inclination of the spindle axis influences the surface quality generated and the roughness of the surface obtained by this process must be improved. The change in run out sizes should be limited to a short interval. The modeling of the vibrations is also possible from these data and the dynamic behavior of the active part (tool, tool holder of the spindle, and angle of inclination) can be analyzed. The vibration effect of the machine can be explained by superimposing the vibratory motion on the nominal peripheral of the milling operation [6]. In order to estimate the dynamic characteristics of system, modal analysis technique can be effectively used. Estimation of machine tool's modal parameters is very important in service. the parameter estimation is carried out by experimental Modal Analysis (EMA), under artificial excitation using an impact hammer or shaker tests, at rest to control machining stability more efficiently [7-8]. The input and output data are measured from the frequency response functions (FRFs) of the structure, which represents the relationship between the excitation and vibrational response of the structure, Conditional parameters are specified [9-11]. Maamar et al developed a robust modal identification

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procedure of a machine tool, during machining. In service, structural modes are weakly present and masked by strongly dominant harmonic components, which can induce a large error in the estimated values. The transmissibility Function-Based (TFB) method has been carried out in order to identify modal parameters of a machine tool, in machining conditions. Results demonstrate that the TBF method is a particular OMA approach by its original ability to eliminate spurious poles and to provide an accurate prediction of modal parameters in the presence of preponderant harmonic components. The modified Enhanced Frequency Domain Decomposition method (EFDD), which is based on white noise excitation assumption, is considered for a comparison purpose. The presence of harmonics is reduced when considering information about the applied cutting force and the distinction between harmonics and structural poles is performed using three selection criteria [12]. Recently, Devriendt et al proposed the Transmissibility Function-Based (TFB) method. The authors demonstrated its original ability to eliminate harmonic components and its independence from the excitation nature [13-15]

From the theoretical stand point finite element method plays a vital rule in modal analysis. The FEM can be applied in solving the mathematical models of many engineering problems, from stress analysis of truss and frame structures or complicated machines, to dynamic responses of automobiles, trains, or airplanes under different mechanical, or electromagnetic loading. The finite element method (FEM), or finite element analysis (FEA), is based on dividing a complicated object into smaller and manageable pieces. computer-aided design (CAD) using computer graphics is used to design a product instead of hand drawing and computer simulations using computer-aided engineering (CAE) software is also nowadays used to analysis the design instead of hand calculations. Many commercial programs have become available for conducting the FEA like ANSYS work bench [16-19].

II. EXPERIMENTAL PROCEDURE AND SET UP:

To get the experimental data, an accelerometer of type B&K Tri-Axial Piezoelectric Type 4506 was used. the system was excited though an impact hammer of type B&K Type (8202). The function FRF was obtained by data acquisition B&K Data Acquisition Type 3160 using enhanced domain decomposition technique to estimate the dynamic characteristics of the tool holder (natural frequency and damping ratio). The measurements were made for holder only and also (tool- tool holder) structure.

A. Experimental Set up for the Holder only:

The measurement was carried out on the MT3 ER32Taper holder which consists of three components (main shaft, collar and stopper) as shown in fig 1 according to the following procedure:

•The tool holder was hanged by a spring on a stand Let the system fixed from the holder shank and free from tool direction

•An accelerometer was fixed on the stopper.

•An impact hammer is used to give the excitation to the system.

•Measurements were obtained by data acquisition.

Fig. (1) Experimental setup and data acquisition system for holder only.

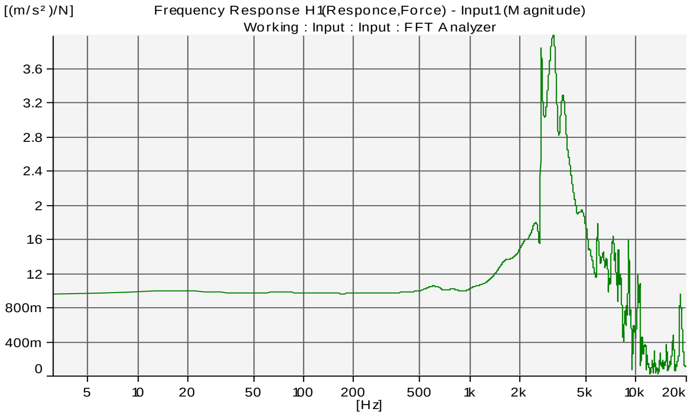

Fig. (2) FRF Results for holder only.

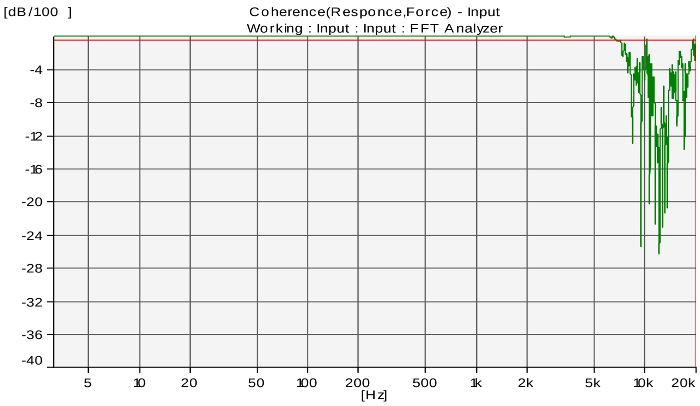

Fig. (3) Coherence diagram for the holder only.

Table (1) Damping natural frequency results and vibration amplitude for holder only from experimental work.

Modal test result			
Mode no	Damped frequency (Hz)	Vibration amplitude $[(m/s^2)/N]$	Damping ratio %
1	2750	1.7	0.82
2	3050	3.8	0.71
3	3800	4	0.412
4	5150	3.3	0.518
5	5600	1.9	0.432

B. Experimental Procedure for the (Tool – Tool Holder) Structure:

• The measurements were taken like the holder only procedure where (the tool-tool holder) structure was hanged

Let the system fixed from the holder shank and free from tool direction as shown in fig 4.

- The accelerometer was fixed on the stopper.
- System excitation by the impact hammer.
- Results taken by data acquisition.

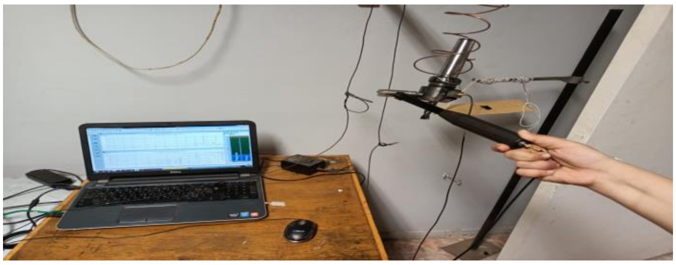

Fig. (4) experimental setup and data acquisition system for (tool-tool holder) assembly.

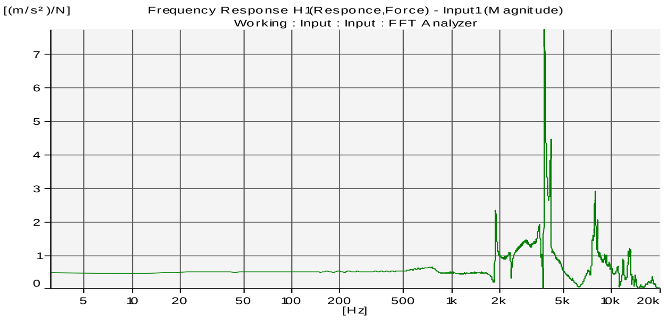

Fig. (5) Frequency vs Amplitude for tool- holder assembly

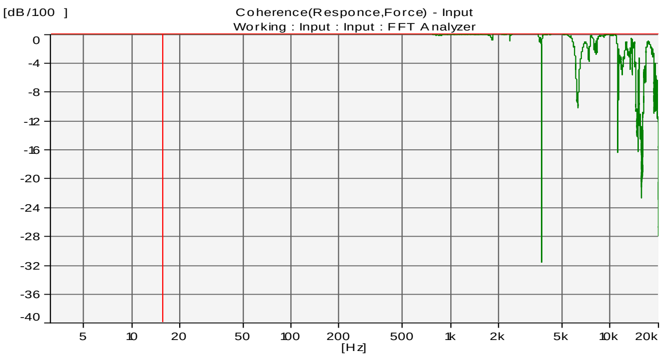

Fig. (6) Coherence measurements for tool- holder assembly.

Table (2) Damping natural frequency and vibration amplitude for (tool-tool holder) assembly from experimental work.

Modal test result			
Mode no	**Damped frequency (Hz)**	**Vibration amplitude $[(m/s^2)/N]$**	**Damping ratio%**
1	1900	2.5	0.78
2	2500	1.3	0.67
3	2750	1.9	0.528
4	2900	7.5	0.517
5	3050	4.5	0.804

III. FINITE ELEMENT MODELING USING ANSYS SOFTWARE:

The modelling was carried out for the holder only and tool-tool holder assembly

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A. Modeling for Holder Only:

For the used tool holder type is MT3 ER32Taper. alloy steel was used for toolholder component material.

1. Modeling Procedure:

- Construction 3D model by geometry using solid work software as fig. 7.
- Input Material properties of holder components as table 4
- Selection Boundary conditions (fixed top of main shaft).
- Automatic Mesh selection for all component of tool holder.
- Give solution order to the program

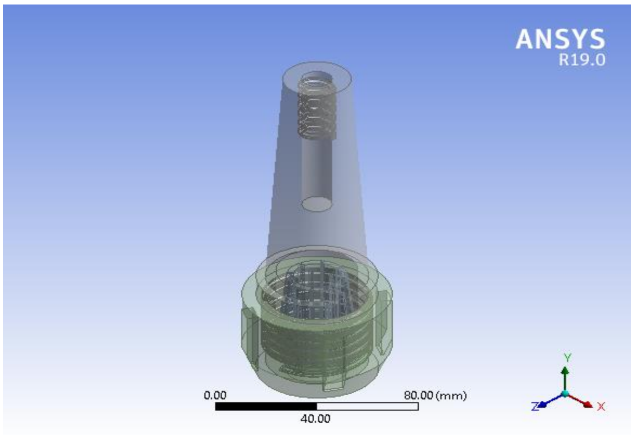
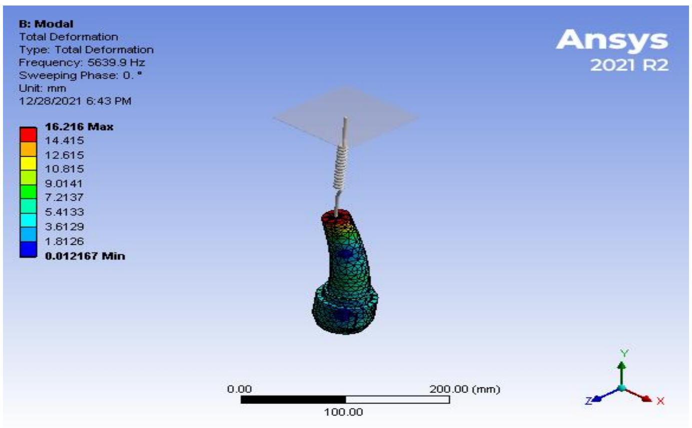


Fig. (7) tool holder geometry.

The modal analysis has been done to estimate natural frequencies to find time consuming when damping ratio is constant and equal 0.09. A sample of mode shape result from the model is shown in fig. (8).

Table (3) Mechanical properties of alloy steel.

Mechanical Property	Value
Density (kg/m^3)	7600kg/m^3
Modulus of elasticity(MPa)	233000
Poisson's ratio	0.3
Bulk modulus (Mpa)	194100
Shear modulus(Mpa)	896000



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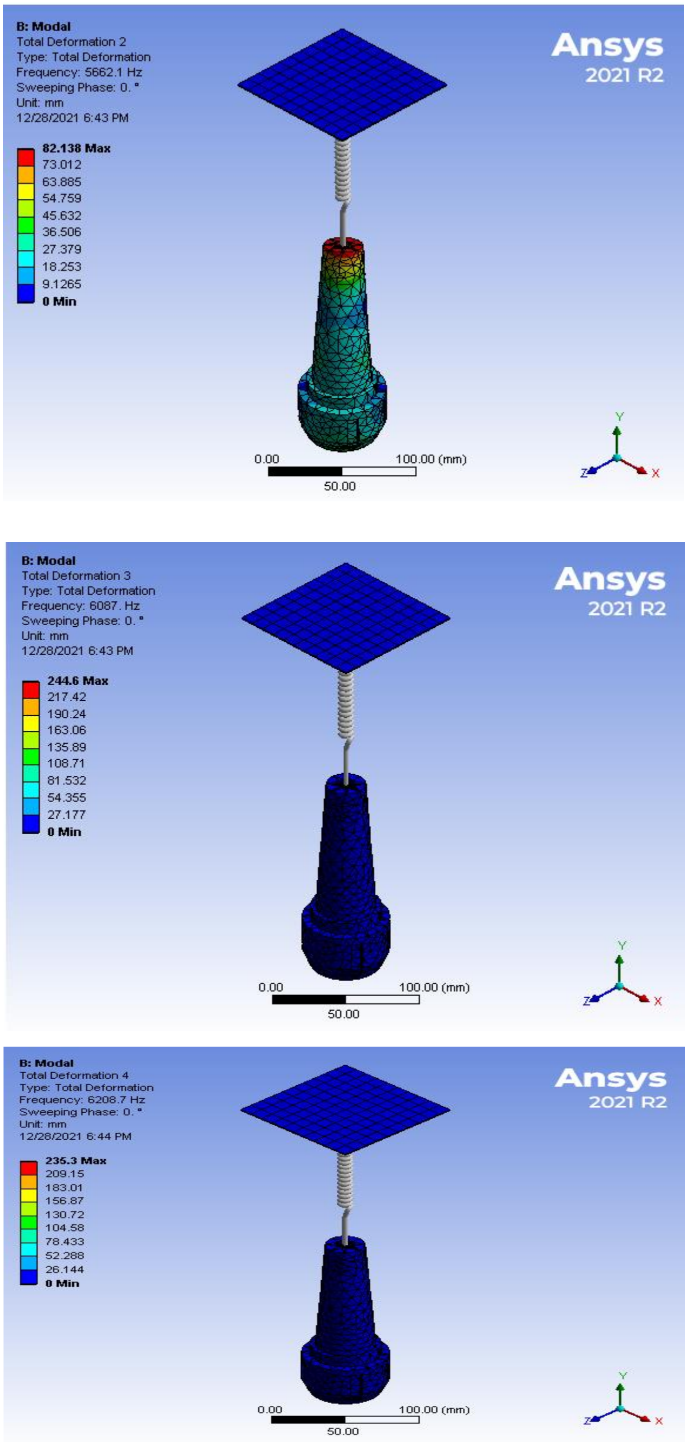

Fig. (8) sample of mode shapes of holder.

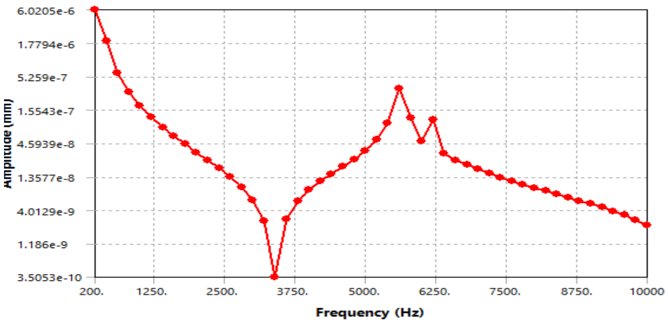

Fig. (9) Amplitude of the first ten modes shapes for holder.

Table (4) damped natural frequency, damping ratio and maximum deflection of holder only.

ANSYS results			
Mode No.	Damped Frequency (Hz)	Damping Ratio%	Maximum deflection (mm)
1	5639.9	9.9877e-003	16.216
2	5662.1	9.9876e-003	82.13
3	6087.	9.9661e-003	235.3
4	6208.7	9.9668e-003	244.6
6	6230.8	9.9537e-003	409.1
7	6250.4	9.9498e-003	392.5
8	6317.7	9.9119e-003	385.79
9	6344.	9.9389e-003	109.51
10	6369.2	9.925e-003	251.02

B. Modeling for Tool - Tool Holder Assembly:

By adding the tool to the last system and repeat the same procedure. Geometric model for the tool-tool holder assembly is shown in the fig. (10). The used material for the tool was high speed steel (HSS). Tool material properties are shown in table (5).

Table (5) Mechanical properties of highspeed steel.

Mechanical Property	Value
Density (kg/m^3)	7700kg/m^3
Modulus of elasticity (MPa)	210000
Poisson’s ratio	0.28
Bulk modulus (Mpa)	159000
Shear modulus (Mpa)	820000

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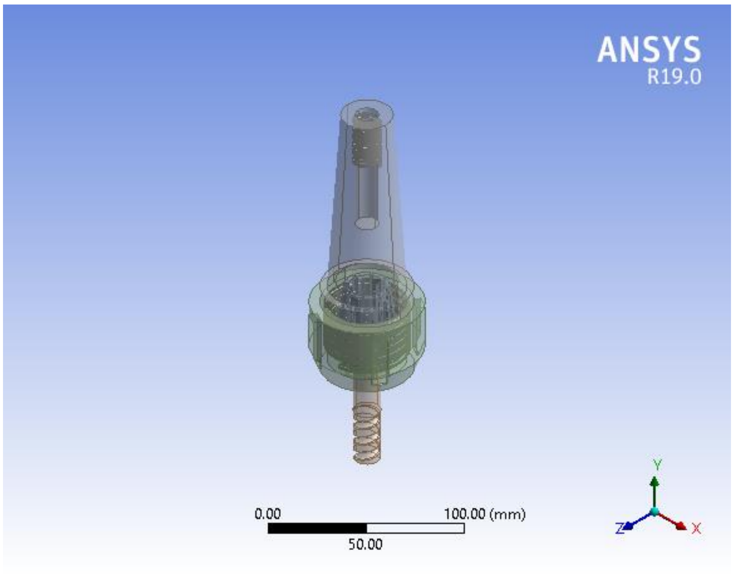
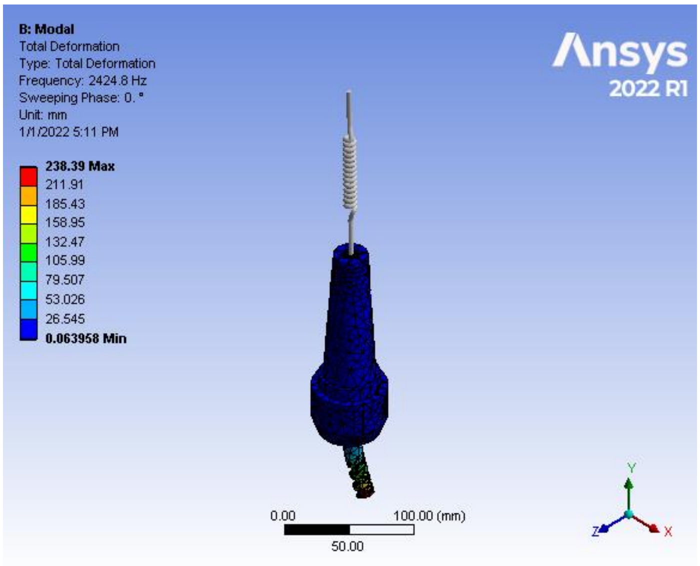
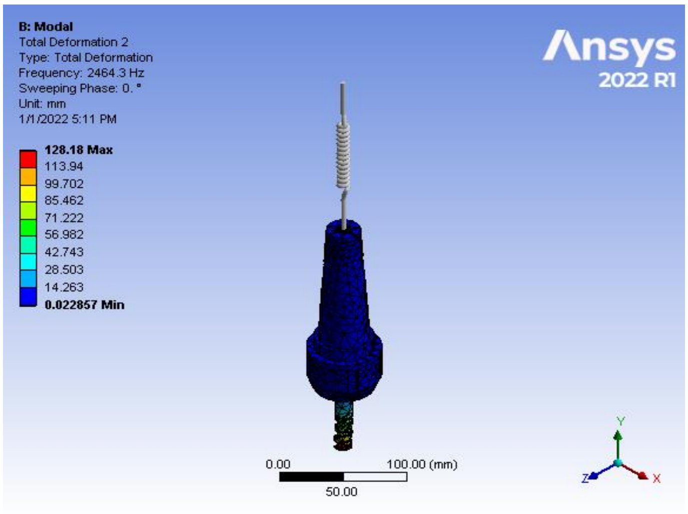
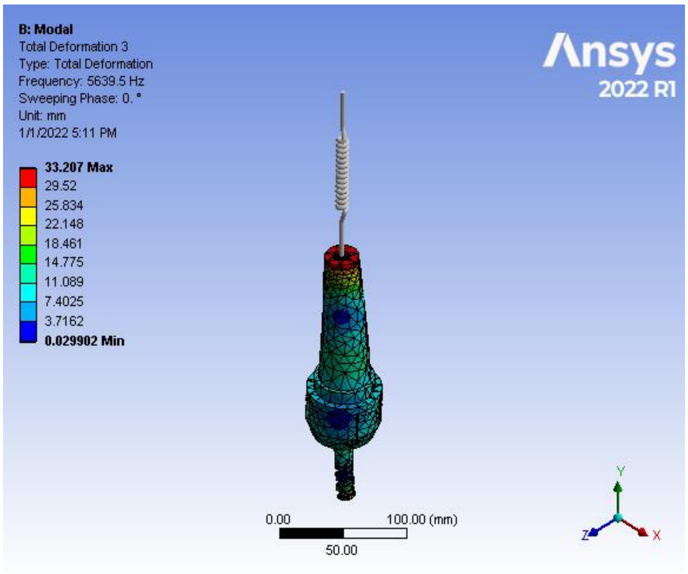


Fig. (10) tool holder assembly.

After carrying the model analysis at constant damping ratio equal 0.09 to estimate natural frequencies and damping ratio as in table 4 and a sample of mode shapes as in fig 11.







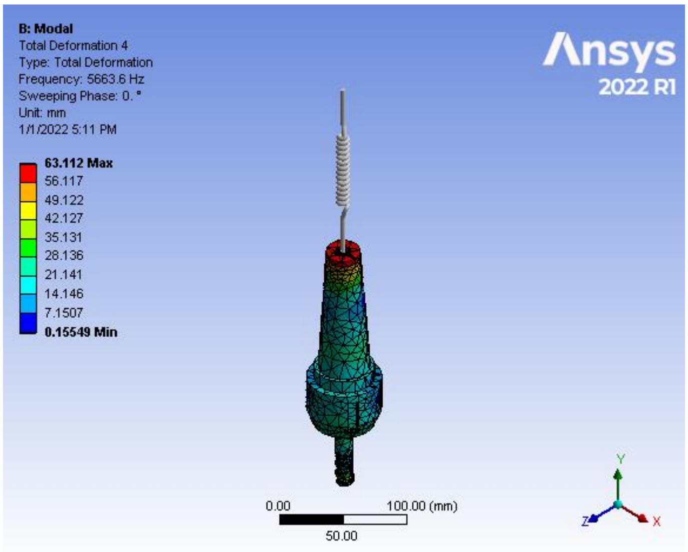


Fig. (11) A sample of mode shapes of tool-holder system

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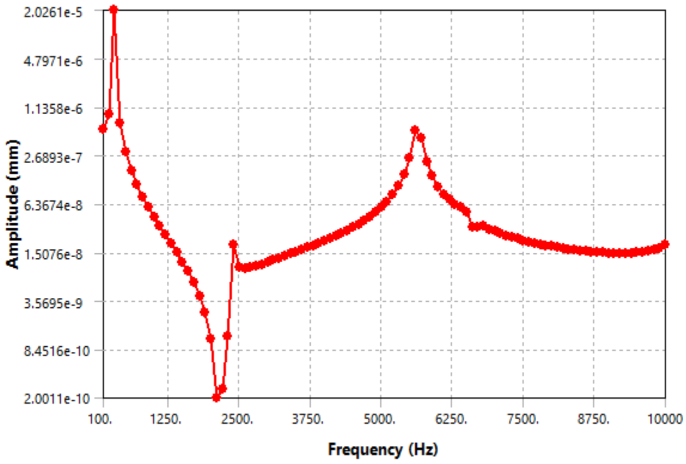

Fig. (12) Amplitude of the first ten modes shapes for Tool – holder assembly.

Table (6) Damped natural frequency, damping ratio and maximum deflection for tool holder assembly.

ANSYS results			
Mode No.	Damped Frequency [Hz]	Modal Damping Ratio	Maximum deflection (mm)
1.	2424.8	9.9625e-003	238.39
2.	2464.3	9.9591e-003	128.18
3.	5639.5	9.9731e-003	33.207
4.	5663.6	9.9733e-003	63.112
5.	6515.7	9.9432e-003	193.4
6.	6561.3	9.8961e-003	119.24
7.	6567.6	9.8959e-003	329.32
8.	6589.5	9.897e-003	327.10
9.	6606.6	9.8976e-003	43.716
10.	6615.7	9.9003e-003	427.29

IV. RESULTS AND DISCUSSION:

- From experimental modal analysis results it's found that in the case of carrying the analysis out on the holder only, a group of peaks and frequencies values appear as in the fig 2, and the damping ratio was found to be equal as in the table 1. And when assembling the holder and the tool, it was found that the values of frequencies decreased, as in the fig 5, as well as the decrease in the existing damping ratios, as in the table 2. And all mods are about bending modes.
- It was also observed that at the frequency of the fourth mode which equal 2900 Hz has an amplitude of 7.5mm. this high amplitude indicates that machining must be avoided at this frequency.
- When comparing the dynamic characteristics of the holder only and the tool- holder assembly, it was found that the natural frequencies decreased approximately from 2750 Hz to 1900 Hz for the first mode, as well as the damping ratio decreased in the case of the holder and the tool more than the holder only.
- The high damping ratio of the holder is considered an influential factor, and this is expected from the characteristics of the holder that it has an effective effect on cutting operations.
- From the theoretical point of view, when the analysis was carried out using the ANSYS software for both the holder only and the tool- holder assembly, the values of the natural frequencies and the damping ratio for the holder as in a table 4. and also for the tool- holder assembly as in a table 6. and also a sample of the mod for each of the holder and the tool- holder assembly appeared as in the fig. 8 and fig 12 respectively. Mode shapes showed the places maximum deformation for each mode, where the red region indicates the maximum deformation that occurs.
- It was observed that There is a difference in the number of mods resulting from the experimental and the theoretical analysis, because the experimental work measure bending only, but the theoretical measures both of bending and torsion modes so the comparison was made between the mods that have bending only in the two cases.
- A comparison between the modal test and ANSYS results for the holder only was carried as in table7 and for tool-holder assembly as in table 8. Deviation error was estimated by this relationship :

error (%) = (ANSYS result -modal test result)/ ANSYS result.

Table (7) comparison bet. Modal test and ANSYS results and error percentage for holder only.

Mode No.	Natural Frequency (Hz) (Modal Test)	Natural Frequency (Hz) (Theoretical - ANSYS)	Deviation Error (%)
1	5150	5639.9	8.6%
2	5600	5662.1	1.09%
3	-	6087.	-
4	-	6208.7	-
5	-	6213.	-
6	-	6230.8	-
7	-	6250.4	-
8	-	6317.7	-
9	-	6344.	-
10	-	6369.2	-

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Table (8) comparison bet. Modal test and ANSYS results and error percentage for tool-holder assembly.

Mode No.	Natural Frequency (Hz) (Modal Test)	Natural Frequency (Hz) (Theoretical - ANSYS)	Deviation Error (%)
1	1900	2424.8	21.6%
2	2500	2464.3	-1%
3	-	5639.5	-
4	-	5663.6	-
5	-	6515.7	-
6	-	6561.3	-
7	-	6567.6	-
8	-	6589.5	-
9	-	6606.6	-
10	-	6615.7	-

- From table 7 it's observed that there is a small difference between the data obtained by modal test and ANSYS and error percentage is illustrated above in case of holder only when comparing the bending modes from the two methods and in tool-holder assembly as table 8. this difference due to many different factors such as surroundings conditions, measuring accuracy and instruments sensitivity.

  - This approach in results indicates to the validity of experimental model analysis.

V. CONCLUSION:

- Results show that good agreement between experimental and analytical data from ANSYS.
- Theoretical analysis using ANSYS program help in identification dynamic characteristics (damped ratio natural frequency mode shapes).
- ANSYS software is a powerful tool for modal analysis that can rely on in cases that no instruments available and can calculate tool holder damping.
- Damping in holder is an affecting parameter in improving cutting process.
- Modal analysis technique is avital method for estimating dynamic characteristics of system.

VI. RECOMMENDATION:

It's recommended to pay attention to damping and adding materials that help to increase the damping of the holder in order to obtain a high surface finish and high accuracy in cutting operations.

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