UDK 621.824:534.1

Scientific paper

TRANSIENT VIBRATION ANALYSIS OF A TURNING MACHINE, S SPINDLE

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Abstract

Spindle is one of the most important member in turning machines, so that its precision determines the dimensional tolerances of machined parts. In this paper, the spindle's response to standard input parameters has been analyzed via Finite Element Method (FEM).From this point of view, the spindle of LZ330VS turning type machine is modeled with Solid Works software. Then, the model is meshed via Msc-Patran software and analyzed by ANSYS. Four values which are the same for all inputs are applied to each one to study the affection of input parameters on the model, simultaneously. The results show that increasing in applied loads increases the oscillating dominations. The value of applied forces has also more affection than the applying time and input type on the oscillating dominations.

Key words: *Turning, Spindle, Modeling, Vibration, Machining.*

1. INTRODUCTION

Production systems are going to rapid automations and manufacturing elements with very high accuracies. Therefore, the precision of various machine elements from statically, dynamical and thermal points of view are very important. In the selection of process parameters, production-planning engineers are conservative in order to avoid undesirable results such as spindle errors, chipping, cutter breakage or over-cut due to excessive cutter deflection [1]. Recently, the number of high speed spindles and high speed machining systems introduced on various machine tools has shows a steady increase. With increasing popularity in high speed machining, optimal design and operation based on spindles errors are becoming more important [2]. Vibration analysis is concerned to the behavior of oscillation parts and their related forces. In other words, vibration is a dynamic phenomenon, so that, many machine's structures are affected by this phenomenon [3]. Although, vibration in machine tools is almost undesirable but, sometimes its creation is non-avoidable. Undesired vibrations controlling and applying them when necessary are the target of designers [4]. The mechanical systems may be malfunctioned and damaged under these undesirable vibrations. Vibration in machine tools on operation is very important. These vibrations can be analyzed from two points of view [5]; 1) chatter which is very important and related to the edges of tool on the surface of the workpiece [6] and 2) the vibrations related to workpiece's supports, mainly spindle and tailstock. In critical conditions, these vibrations have seriously affections on machining output parameters and accuracy of machine parts [7].

Among various machine tools' parts, spindle which directly dictates the quality and accuracy of machined parts is under closed attention [8]. This subject is more serious and obvious in machining production with high spindle speed and good surface finishing [9]. Although a lot of works has been done by many researches but, most of them as designed documentations, belong to manufacturing companies. The accuracy of spindle is the main point that determines the quality and precision of final products [10]. Long lasting and continuation of this accuracy with time and against various shocks are more important [11]. Since this accuracy elongation causes uniform production which is basic condition of economically production, unsuitable design with miss-considering effective parameters causes decreasing in spindle accuracy due to applied shocking [12].

2. MODELING PROCEDURES

In this research the spindle of a turning machine from Leinen Co. in Germany is analyzed. After measuring various machine parts with 0.05 mm accuracy a schematic model is produced in Solid works area and is assembled with assembly software (See Fig. 1). Spindle, driver and rotor gears, two gear pins, bush and center are the elements of the model. This is modeled elementally with Cosmos/ Design Star format and the result is analyzed via ANSYS Solid 92 with three positioning and three rotational degrees of freedom. Density (7800 Kg/m^3), Poison coefficient (0.28) and Young Modulus (210×10^9 Pa) as physical characteristics are considered in the model. In order to determine the force effects, the amounts of 300N, 400N, 500N and 600N are considered. Boundary conditions are the function of bearings and their dimensions. Spindle is bounded with roller bearing and ball thrust-bearing. The roller bearing cancels Rx, Ry, Ux and Uy degrees of freedom from surfaced nodes whereas, the ball thrust-bearing cancels these points with Uz degree of freedom(U and R are the positioning and rotational symbols respectively) (See Fig.2).

3. ANALAYSIS OF SPINDLE FORCES

Spindle shock has various sources and its amplitude depends on numerous parameters such as; cutting depth, material removal rate lubricant / no lubricant fluid usage conditions, the kind of machine tool, tool cutting angle and so on. To calculate the force applied on the tip of the center, the general following equation with pre-suggestions can be used [11]:

$$K = K_e.K_\phi.K_\gamma.K_m \quad , \quad P_y = C_p.t^x.S^y.K \qquad (1)$$

These presuggestions are as follows:

1. There is no lubricant fluid used, then K_e=1

In using or not using lubricant fluid conditions, there is not any noticeable different of Ke value between these two conditions since, the difference value of K_e in these two conditions is something about 10٪.

2. The tool's tip angle respect to workpiece K_ϕ=0.92 then; $\phi=90^\circ$
3. The scanned angle (the angle between tool's surface head and vertical plane) $\gamma=90^\circ$ then; $K\gamma$=1.13
4. Workpiece material : It is supposed that the ultimate stress of workpiece material is in the range of 400 Mpa<σ_{ult}<500 Mp then; K_m =0.79
5. Machine tool: Turning then; x =1, y = 0.75, C_p=225

Substituting the mentioned value into Eq.1, the value of force can be calculated as follows:

$K = K_e.K_\phi.K_\gamma.\ K_m = 0.82$

The variations of this force respect to the cutting depth and feeding rate is shown in Figure 3.

4. TRANSIENT ANALYSIS

To do the transient analysis of center, it is necessary to obtain the first natural frequency of the system [12]. The time step, from this frequency, can be determined to do transient analysis. According to ANSYS guidance, this time step is $\Delta t=1/20f_n$

This is an undetermined system since, the natural frequency at the first step is zero and the spindle is not rotationally fixed around Z axis. The result of natural frequency analysis shows that the first non-zero frequency is **fn**=5.048 Hz or Δt=0.01 sec.

This is the least value of time step in transient analysis which is used as standard input value in analyzing of impact, step, ramp and ramp-step (See Fig. 4).

5. DISCUSSION

The result analysis is as follows:

5.1 Impact

The mathematical function of impact is defined as follows:

$$F(t) = \begin{cases} a & t = t_1 \\ 0 & t \neq t_1 \end{cases} \qquad (2)$$

The minimum step time Δt=0.01Sec. is used since, it is not possible to apply any force in zero duration time. So that from t = 0 to t =.01 the force reaches to its maximum value. In this case, the deflection of the tip of the center reaches to its maximum value. Due to the omission of the force, the tip of the center starts oscillating around its equilibrium (no loading) point. Since the oscillation amplitude against primary displacement is very small, this amplitude is not clear in analytic figure (see Fig. 5 and Table 1).

5.2 Step

The step mathematical equation is as follows:

$$F(t) = \begin{cases} a & 0 \le t \le t_1 \\ 0 & t > t_1 \end{cases} \tag{3}$$

As the previous section, the applied load reaches from zero to its maximum value at time step Δt=0.01sec and remain constant till $t = t_1$. In this research it is supposed that t_1= 0.5 Sec. The load is cancelled at the end of this time duration. Afterwards, the center tip oscillates around its equilibrium point (See Fig.6 and Table 1).

5.3 Ramp

The mathematical function of slope is as follows:

$$F(t) = \begin{cases} a.t & 0 \le t \le t_1 \\ 0 & t > t_1 \end{cases} \tag{4}$$

In this case the load reaches to the maximum value at the time range of 0-t_1, so that the displacement of the center tip finds its maximum value. Then, due to load canceling, the center tip oscillates around its equilibrium point. The time t_1=0.5 sec and therefore a=F/0.5=2F is considered in this case (See Fig. 7 and Table 1).

5.4 Ramp - Step

The mathematical slope – step function is defined as:

$$F(t) = \begin{cases} a.t & 0 \le t \le t \\ a.t & t_1 < t \le t_2 \\ 0 & t > t_2 \end{cases} \tag{5}$$

At t = 0- t_1 time duration, the load and afterwards, the displacement of the center tip reaches to its maximum value. At t = t_1-t_2 time duration, the value of load and displacement remains constant and at t = t_2 the load is cancelled. So, the center tip starts the oscillation around its equilibrium point. In this research t_1=0.5Sec and t_2=1Sec are considered (See Fig.8 and Table 1).

6. RESULT

In this research the tip center vibration of a special turning machine under machining load is analyzed. The results are as follows:

- According to harmonic analysis of obtained graphs, it is defined that the third, sixth and ninth frequencies among the first ten are excited. Therefore, $f_{n,3}$ is considered as desired natural frequency.
- The displacement contour in this analysis has two figures; a) the applied load is affected mostly on the tip of the center and there is not any noticeable affection on the other positions. b) The combination of the center and the end part of spindle are oscillation affected. In this manner, the affected zone is more in comparison with the previous case.
- The oscillation figures resulted from applied standard inputs shows that the response is the combination of several harmonics. Therefore, in addition to the third mode, the sixth and ninth modes are also excited with input applications.
- According to the results, the vibration amplitude increases with time increasing. But, the value of load has more affection respect to the kind of input and applied time (Fig.9).
- The resulted analysis of ANSYS, MSc-Patran and Cosmos/Design Star softwares show the same and confirm each others (Table 3).

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Tables and figures

Table 1. Natural frequencies obtained from modal analysis

Fn,1=568.25	Fn,4=675.68	Fn,7=723.04	Fn,10=788.69
Fn,2=648.27	Fn,5=683.35	Fn,8=763.65	
Fn,3=652.12	Fn,6=720.13	Fn,9=768.16	

Table 2. Maximum displacement with various inputs during and after shock

Input	**Displacement Amplitude on Shock Duration (m)**				**Displacement Amplitude After Shock (m)**			
Force	F=300 N	F=400 N	F=500 N	F= 600 N	F=300 N	F=400 N	F=500 N	F=600 N
Shock	0.550e-6	0.550e-6	0.577e-6	0.577e-6	0.396e-6	0.528e-5	0.573e-5	0.791e-5
Step	0.733e-5	0.733e-5	0.796e-5	0.796e-5	0.419e-6	0.559e-5	0.698e-5	0.838e-5

Table 3. Comparison between three natural frequencies obtained from ANSYS ، Msc-Nastran and Cosmos/Design Star software

Natural Frequencies	ANSYS	Msc-Patran	Cosmos/Design star
$F_{n,1}$	568.25	578.10	583.61
$F_{n,2}$	648.27	637.30	654.12
$F_{n,3}$	652.12	654.01	660.06
$F_{n,4}$	675.68	669.85	681.76
$F_{n,5}$	683.35	674.53	689.48
$F_{n,6}$	720.13	711.04	727.53
$F_{n,7}$	723.04	730.71	740.07
$F_{n,8}$	763.65	770.00	772.43
$F_{n,9}$	768.16	773.09	781.29
$F_{n,10}$	788.69	783.42	796.51

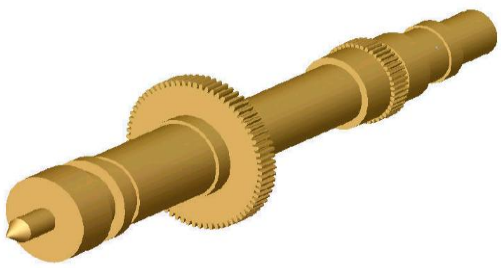


Fig.1. Solid Works model of center

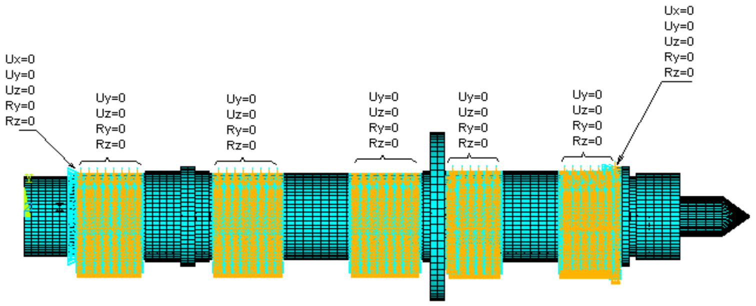


Fig.2. Boundary conditions applied in ANSYS

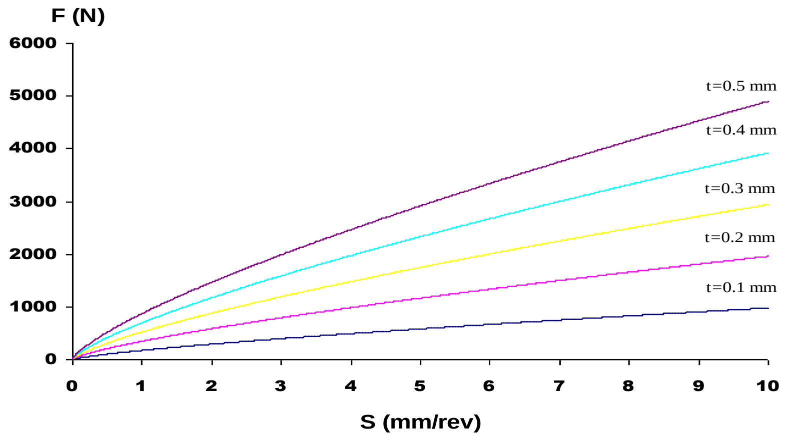


Fig.3. Center tip forces variations with feed rate in various depths of cut

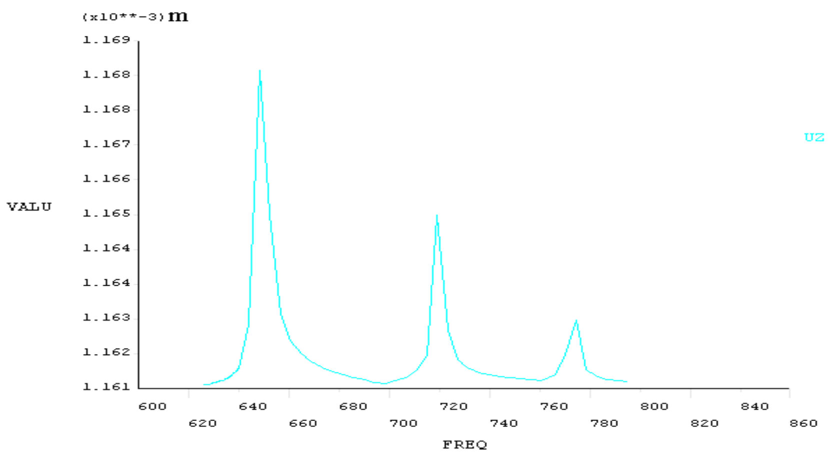
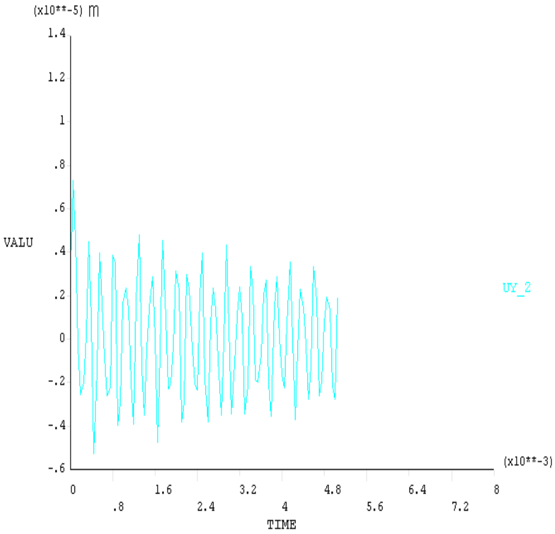
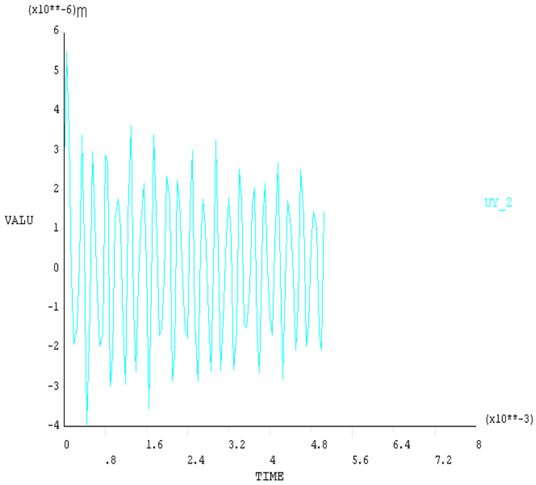


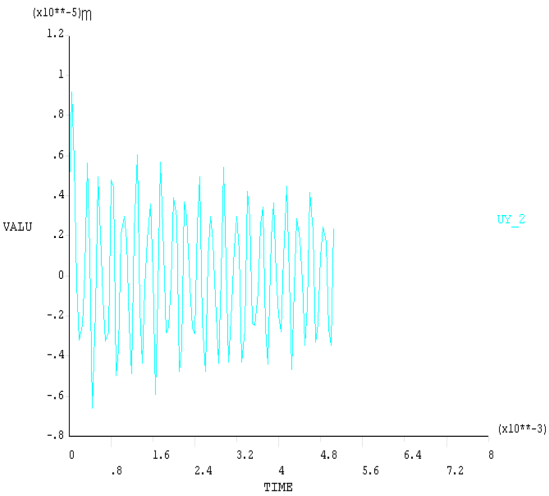
Fig.4. Displacement of center tip with shock frequency



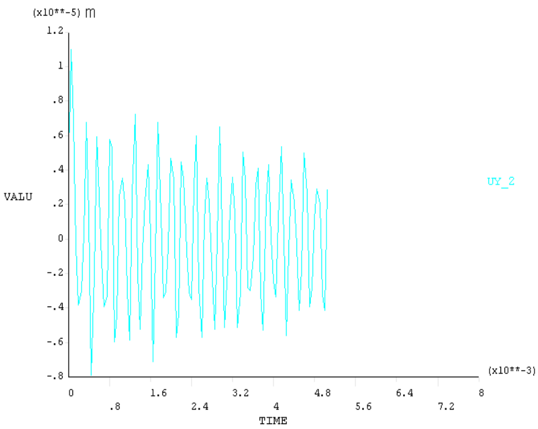
(a)



(b)

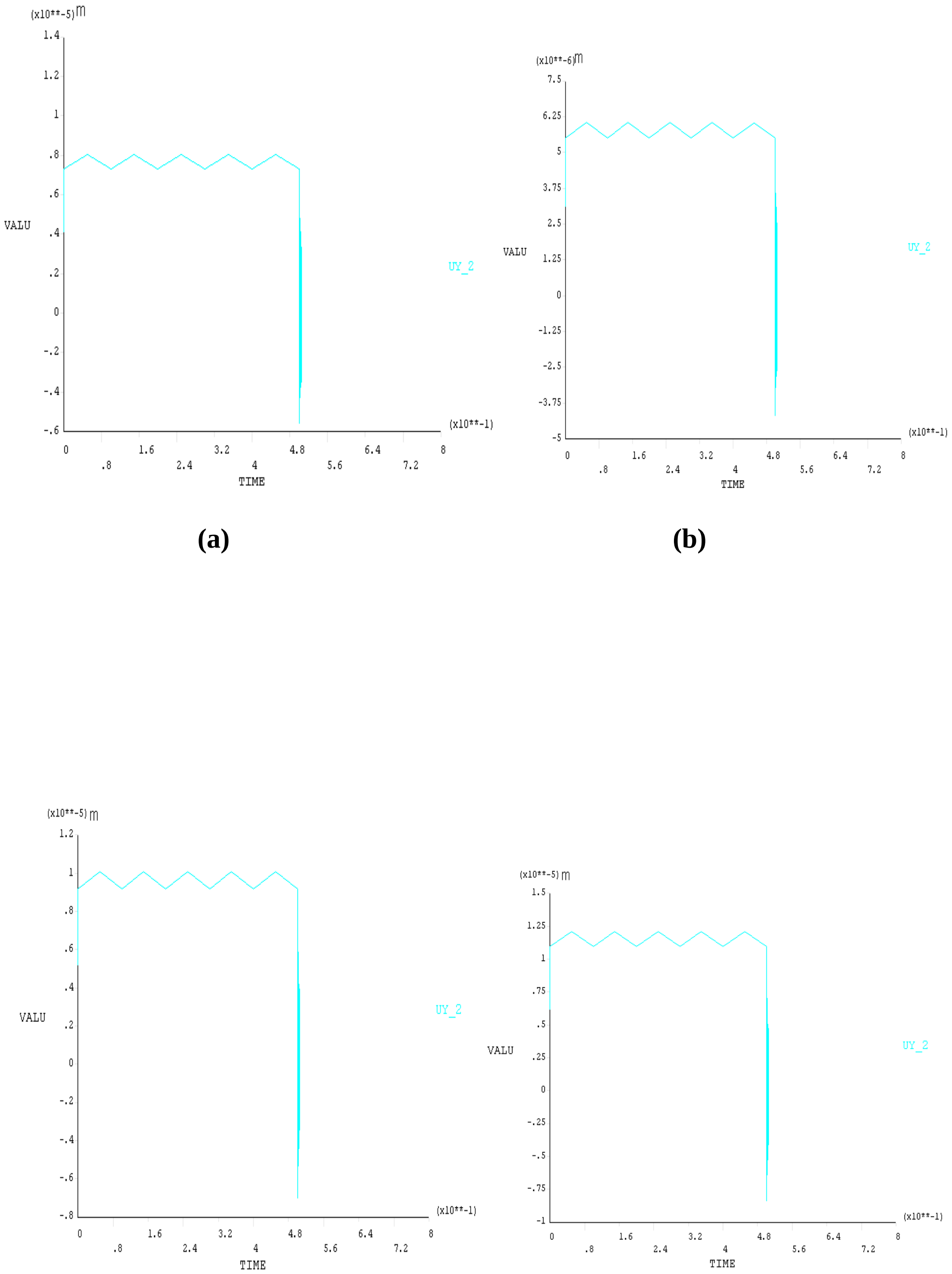


(c)



(d)

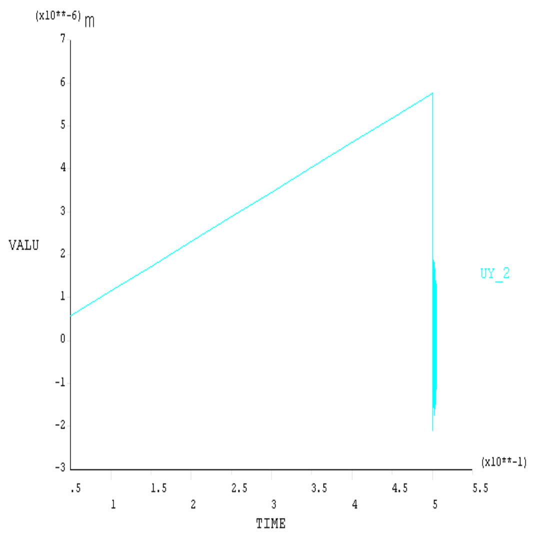
Fig.5. Displacement of center tip against impact with; (a)-F=300 N, (b)- F=400 N, (c) F=500 N and (d)-F=600 N



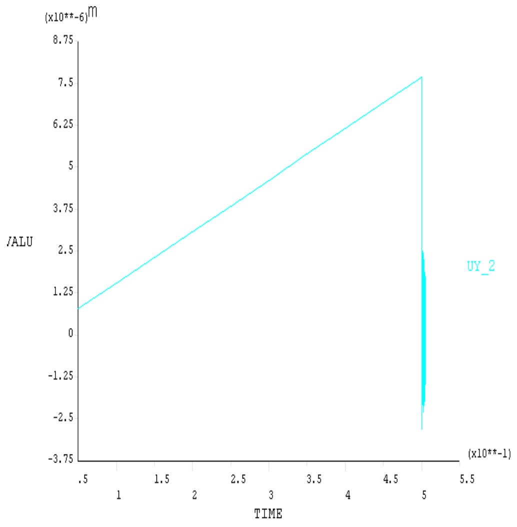
(a) (b)

(c) (d) --

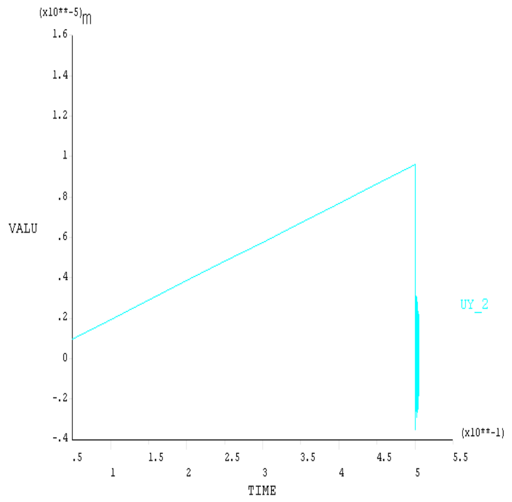
Fig.6. Displacement of center tip against step with; (a)-F=300 N, (b)- F=400 N, (c) F=500 N and (d)-F=600 N



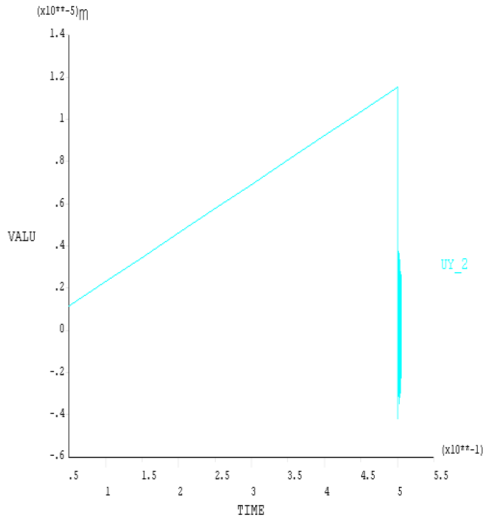
(a)



(b)

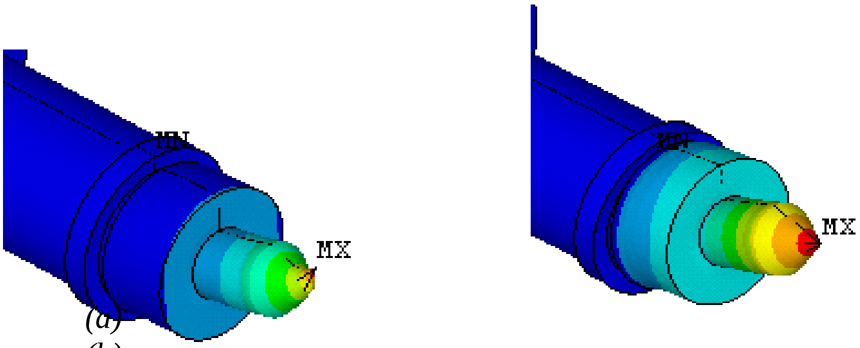


(c)



(d)

Fig.7. Displacement of center tip against ramp with; (a)-F=300 N, (b) - F=400 N, (c) F=500 N and (d)-F=600 N



(a)
(b)
(c)

(a) (b)

Fig.8. Displacement of center tip contour in transient analysis on; (a)-loading continuation and (b)-oscillation after load cancellation

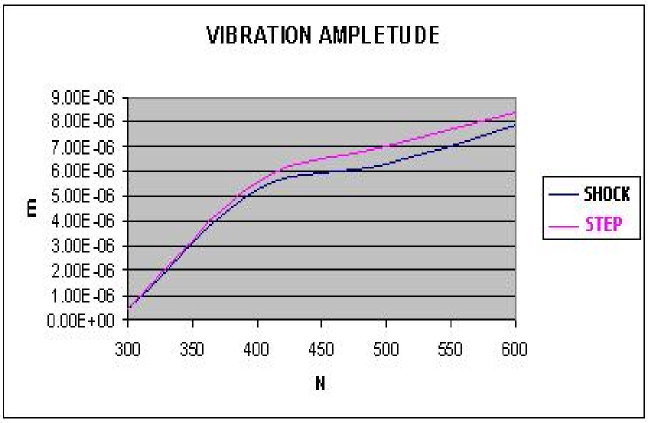


Fig.9. Vibration amplitude variations with shock force

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