Machine Tools Foundation Static and Free vibration analyses

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Abstract: Machine tools foundations are very important element of these machine structures. Deformations for foundations under static load are studied in present work, the effect of changing stiffening plates shape and cutting conditions such as type of cutting process and angle of cutting forces on these deformation are investigated using finite element program (ANSYS5.4) also effects of these parameters on the foundation natural frequencies are obtained.

Keywords:machine tools, foundations, free vibration, static analyses, structure, finite element method.

1.Introduction:

To perform satisfactorily a machine tool must be both statically and dynamically rigid. Its static stiffness determines its ability to produce dimensionally accurate parts and its dynamic stiffness affects the quality of the component's surface finish and the maximum metal removal rates that can be achieved[1].
It is essential therefore to be able to specify and subsequently measure the static stiffness of machine tool foundations, in order to ensure that the correct level of support is provided and that the machine tool alignment accuracies are achieved [2, 3].
For a satisfactory machine installation the foundation stiffness must first be specified based upon the required alignment tolerances for the machine, as specified in the appropriate ISO standard for the particular machine configuration e.g. ISO 3070 Part 2 for a large Moving Column, Horizontal Ram Type milling machine and ISO 8636 Part 2 for a Moving Gantry Vertical Ram Type milling machine. The stiffness specification for the foundation must state a number of criteria and the associated tolerances [1].

The following requirements should be satisfied from the design point of view [4]:

1. The foundation should be able to carry the superimposed loads without crushing failure.
2. The settlements should be within the permissible limits.
3. The combined center of gravity of machine and foundation should as far as possible be in the same vertical line as the center of gravity of the base plane.
4. No resonance should occur, hence the natural frequency of foundation-soil system should be either too large or too small compared to the operating frequency of the machine. For low-speed machines, the natural frequency should be high, and vice-versa.
5. The amplitudes under service condition should be within permissible limits. The permissible limits are generally prescribed by machine manufacturers.
6. All rotating and reciprocating parts of a machine should be so well balanced as to minimize the unbalanced forces or moments. This is generally the responsibility of the mechanical engineers.
7. Where possible, the foundation should be planned in such a manner as to permit a subsequent alteration of natural frequency by changing base area or mass of the foundation as may subsequently be found necessary.

In principle machine foundations should be designed such that the dynamic forces of machines are transmitted to the soil through the foundation in such a way that all kinds of harmful effects are eliminated [5]

Z. Huang and S. Hinduja, show how the cost of an existing foundation for a large machine can be minimized by optimizing the parameters which define its shape whilst maintaining the required stiffness. The parameters considered include the thickness of the concrete in the machine pit and the platform, the cross-sectional area and the amount of reinforcement in the piles, the number of piles and the spacing between them. The foundation is optimized using two- and three-dimensional models. Both these models have been optimized for different loading conditions and varying stiffness.A non-linear unconstrained optimization technique combined with the finite element method has been used to obtain the optimal designs[6].

A. Myers et al present a novel technique for accurately measuring the static stiffness of a machine tool concrete foundation using various items of metrology equipment. The foundation was loaded in a number of different ways which simulated the erection of the machine, traversing of the axes and loading of the heaviest component. The results were compared with the stiffness tolerances specified for the foundation which were deemed necessary in order that the machine alignments could be achieved. This paper is a continuation of research previously published for a FEA of the foundation [1].

SHAMSHER PRAKASH VIJAY K. PURI discusses the methods of analysis for determining the response of foundations subjected to vibratory loads. The design of a machine foundation isgenerally made by idealizing the foundation- soil system as spring-mass –dashpotmodel having one or two degrees of freedom. Most machine foundations aretreated as surface footing and the soil spring and damping values are determined usingthe elastic-half space analog. The spring and damping values for response of embeddedfoundations can also be determined from the elastic half space concept as per Novak‟s work. The soil spring and damping values can also be obtained following the impedance–compliance function approach. The paper also presents a brief discussion of thepredicted and observed response of machine foundations[7] .

LOREDANA THEODORA PAUN et al, studied a structure for a multifunctional machine toolcomposite materials based on analysis, in static regime, of. The multifunctional machine is designed for processing by turning, milling, drilling,boring, mortising, toothing and grinding. However, the analysis was done only for turning.One of the news brought by this machine is that its structure is made mostly, of composite materials. The simulation was madein order to determine the total strain, maximum tension, the equivalent tension for shearing and displacements[8].

K.G. Bhatia, highlights need for a better interaction between foundation designer and machinemanufacturer to ensure improved machine performance. The paper also describes the designaids/methodologies for foundation design. Various issues related to mathematical modeling andinterpretations of results are discussed at length. Intricacies of designing vibration isolation system forheavy-duty machines are also discussed. Influences of dynamic characteristics of foundation elements,viz., beams, columns, and pedestals etc. on the response of machine, along with some case studies, arealso presented. The paper also touches upon the effects of earthquakes on machines as well as on theirfoundations. Use of commercially available finite element packages[9].

NICOLAE–DORU STĂNESCU and STEFAN TABACU, propose a simple system with two degrees of freedom based on anon-linear elastic element and the hypothesis for the coefficients of the elastic force. Forthis system, it isproved in the present paper that the motion is stable, but not asymptotically stable. A comparison between thenon-linear case and the linear case is performed, and for the both cases the eigenpulsations are also determined. All theoretical results are validated by numerical simulation. Finally, they considered the general case[10].

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2. Force relation ship:

Several forces can be defined relative to the orthogonal cutting model. Based on these forces, shear stress, coefficient of friction, and certain other relationship can be defined.

2.1 Force in metal cutting:

Consider the forces acting on the chip during orthogonal cutting, as sketched in Fig. (1).

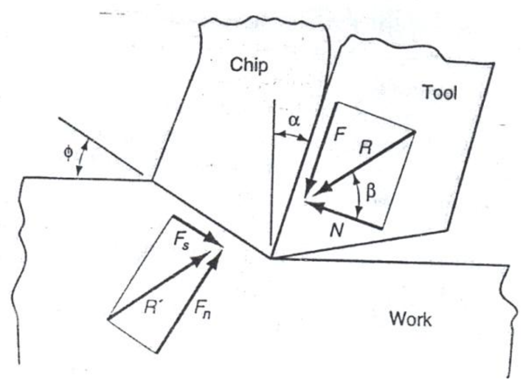


Fig.(1):Cutting force components

The forces applied against the chip by the tool can be separated into two mutually perpendicular components:

1. Friction force (F): This is the friction force between the tool and ship resisting the flow of the ship along the rake face of the tool.
2. Normal force to friction (N).

$$\mu = \frac{F}{N} \dots\dots\dots\dots\dots\dots (1)$$

The friction force and normal force can be added vectorially to form a resultant force (R).

(R) is oriented at an angle (β) is called the friction angle. The friction angle is related to the coefficient of friction as

$\mu = \tan\ \beta$

In addition to the tool forces acting on the chip, the work imposes tow force components on the ship.

3. Shear force (F_S): this is the force that causes shear deformation to occur in the shear plane.
4. Normal force to shear (F_N). This force is normal to the shear force.

Based on the shear force, we can define the shear stress that acts along the shear plane between the work and the chip.

$$\tau = \frac{Fs}{As} \dots\dots\dots\dots\dots\dots\dots\dots (2)$$

Where As = area of the shear plane. This shear plane area can be calculated as :$As = \frac{tow}{sin\emptyset} \dots\dots\dots\dots\dots\dots.. (3)$

The shear stress determined by eq.(2) represents the level of stress required to perform the machining operation. In principle, this stress is equal to the shear strength of the work material under the conditions at which cutting occurs.

Two additional force components can be directly measured. These two components act against the tool:

5. Cutting force (Fc): this force is in the direction of cutting.
6. Thrust force (Ft): This force in the direction of t_o.

It is perpendicular to the cutting force.The cutting force and thrust force are shown in the Fig. (2), together with their resultant force (R").

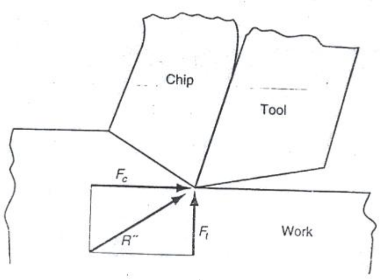


Fig.(2): Thrust and Tangential and their resultant

By using force diagram in Fig. (3) the following trigonometric relationships can be defined:

$$F = Fc \sin \propto\ + FtCOs \propto \cdots \dots\dots\dots. (4)$$

$$\check{N} = Fccos \propto - Ftsin \propto \cdots \dots\dots\dots.. (5)$$

$$Fs = Fccos\emptyset - Ftsin\emptyset \dots\dots\dots\dots\dots\dots (6)$$

$$Fn = Fcsin\emptyset + Ftcos\emptyset \dots\dots\dots\dots\dots.. (7)$$

Note. That in the special case of orthogonal cutting when the rake angle $\propto = o$ Eq. (4) and (5) reduce to F=Ft and N = Fc, respectively. Thus, in this special case, friction force and its normal force could be directly measured by the dynamometer.

Cutting force and thrust force are related to the shear strength of the work material. The relationship can be established in a more direct way.

Recalling from eq.(2) that the shear force Fs = $\mathcal{T}$ As, the force diagram of Fig. (3) can be used to derive following equations:

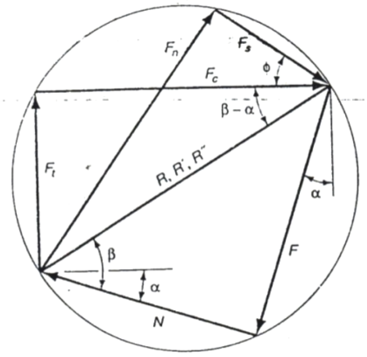


Fig.(3): Cutting force Daigram

$$Fc = \frac{Ttow\cos(B-\propto)}{sin\emptyset\cos(\emptyset + B-\propto)} \dots\dots\dots\dots\dots (8)$$

Or

$$Fc = \frac{Fc\cos(B-\propto)}{\cos(\emptyset + B-\propto)} \dots\dots\dots\dots\dots..(9)$$

And

$$Ft = \frac{Stow\cos(B-\propto)}{sin\emptyset\cos(\emptyset + B-\propto)} \dots\dots\dots\dots\dots\dots..(10)$$

Or

$$Ft = \frac{\text{Fs}\sin(B-\propto)}{\cos(\emptyset + B-\propto)} \dots\dots\dots\dots\dots\dots..(11)$$

These equations allow one to estimate cutting force and thrust force in an orthogonal cutting operation if the shear strength of the work material is known.

2.2 Approximation of turning by orthogonal cutting:

The orthogonal model can be used to approximate turning and certain other single – point machining operations as long as the feed in these operations is small relative to depth of cut. Thus most of cutting will take place in the reactionof the feed, and cutting on the nose of the tool will be negligible Fig.(4) indicates the conversion from one cutting situation to the other part (a) shows turning operation, while part (b) depicts the corresponding orthogonal case.

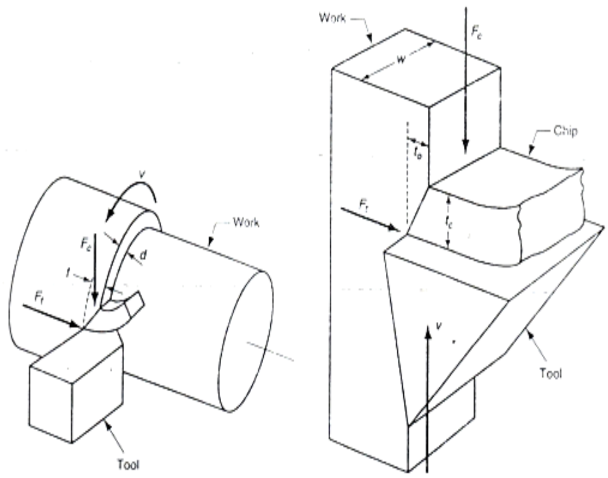


Fig.(4): Orthogonal cutting force components

The interpretation of cutting conditions is different in the two cases. The ship thickness before the cut (to) in orthogonal cutting corresponds to the feed in turning and the width of cut (w) in orthogonal cutting corresponds to the depth of orthogonal model corresponds to the depth of cut in turning. In addition, the thrust force in the orthogonal model corresponds to the feed force (Tf) (that is, the force on the tool in the direction of feed) in turning. Cutting speed and cutting force have the same interpretations in the two cases[12].

3.Modeling of the Machine Foundation:

The structure was modeled using a shell-64(elastic shell) element, in a 3D modeling program and after it was imported into a simulation program as a whole. Simulation of the static behavior of the structure for the multifunctional machine for turning processing it was performed using ANSYS 5.4, a finite element program. The whole assembly was meshing using 5272elements and 8470 nodes. The structure is shown in Fig. (5).
The simulation was made in order to determine maximum displacements (translation or rotation) on the three axes X, Y and Z.

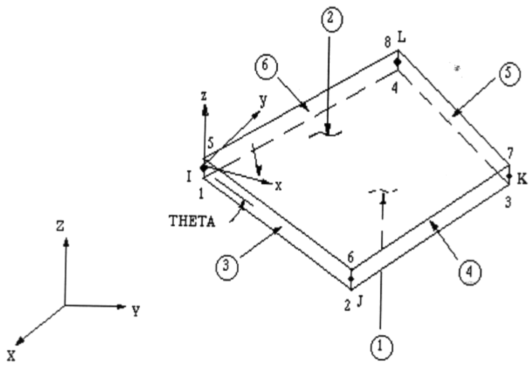


Fig.(5):Shell 63 used in building the foundation models

Table (1): The calculated values of cutting and thrust forces.

Shear(n/m^2)	Rake angle(°)	Cutting force(n)	Thrust force(n)
40	-5	816.718	597.549
	0	854.015	640.490
	10	713.076	381.882
25	-5	510.262	354.588
	0	533.105	399.816
	10	445.528	238.598
130	-5	2653.326	1843.860
	0	2772.150	2079.044
	10	2316.745	1240.713

4. Discussion:

Nine models of structures are built using ANSYS 5.4 with different stiffeners shape as shown in Fig.(6) and two boundary conditions had been taken in present work as follows:

a) All points of the lower face of the structure are built with the ground directlyFig.(7-a).
b) Some of points are built with the ground as shown in Fig. (7-b).

The force and moment are applied on the models with variable cases depending on variation of rake angle of cutting with three values (-5 º, 0 º, 10 º), this change of angles tends to load more in compression and less in shear, thus favoring the high compressive strength of these harder materials [12], so use cemented carbide with previous values of rake angle.

Three types of materials had been chosen (steal SAE 4340 annealed, Monel metal 70Ni-30Cu and Ttanium 99.0 Ti annealed) so that the shear stress would be various.Depending on these values of shear and rake angle by using equations (3, 4) we getnew values of cutting and thrust forces, for each case as follow in **table (1)**,the cutting force plus the weight of chosen machine (2000Kg) had been applied on the models and thrust force translated to bending moment on the foundation of machine, for each case we get on the translation and quotation as follows :

1. From results **table(2,3 and 4),**for three types of materials we noticed that in each model from nine models the rake angle (10 º) get lowest values of translation, that mean the rake angle (10 º) is the best value of rake angle chosen for cutting. And the lowest value of translation in model (1) equal to (2.9674E-5) m.
   Similar try for nine models with each type of material the rake angle (0 º) is getting largest values of translation so the value (0 º) is the worst in present work.
2. From results **table(2,3 and 4)**, we noticed that the rotation and translation values of structures increased with shear stress increase, that is mean the steel SAE 4340 annealed gives highest results from the other materials for two boundary conditions.
3. The deformation results of structure explain the model (1) is the lowest translation with rake angle (10 º) and shear stress (25 N/m^2), this is mean that the design (1) is the best design to current load applied with monel metal (70Ni-30Cu) and the value of translation is (2.9674E-5)m.
   So from the present work we get good results by using Monel metal with rake angle (10 º) for cutting and model has mass (561.6)Kg, so if we noticed **table(5)** we see the good design that chosen has lowmass.
4. For each model we noticed that model (3) had value of translation (0.86354E-3) m,

with (0 º) rake angle for steel SAE 4340 annealed, these value is the worst for two boundary conditions.

5. From **table(2,3 and 4)**, we noticed that the best results from (bc1) that is mean the foundation of machines must be built with ground directly for all points to get the best results.
6. Frequency characteristics of machine foundation must be studied for safety and stability reasons therefore the first three mode shapes for the nine models are obtained as shown in **Fig.(8)** and their natural frequencies are tabulated in **table(6).**

5. Conclutions:

The present work investigates that the best value of rake angle chosen for cutting is angle (10 º) while the worst the rake angle is (0 º).Model (1) is the best design comparatively gets lowest values of translation,(2.9674E-5) m and lowest mass. We also noticed that model (3) is the worst for two boundary conditions and had value of translation (0.86354E-3) m, with (0 º) rake angle for steel SAE 4340 annealed. Building almost the point of the foundation (bc1) results in less deformation in all the models than that resulted from the second boundary condition (bc2), because the models stiffness with (bc1) is larger than for (bc2).

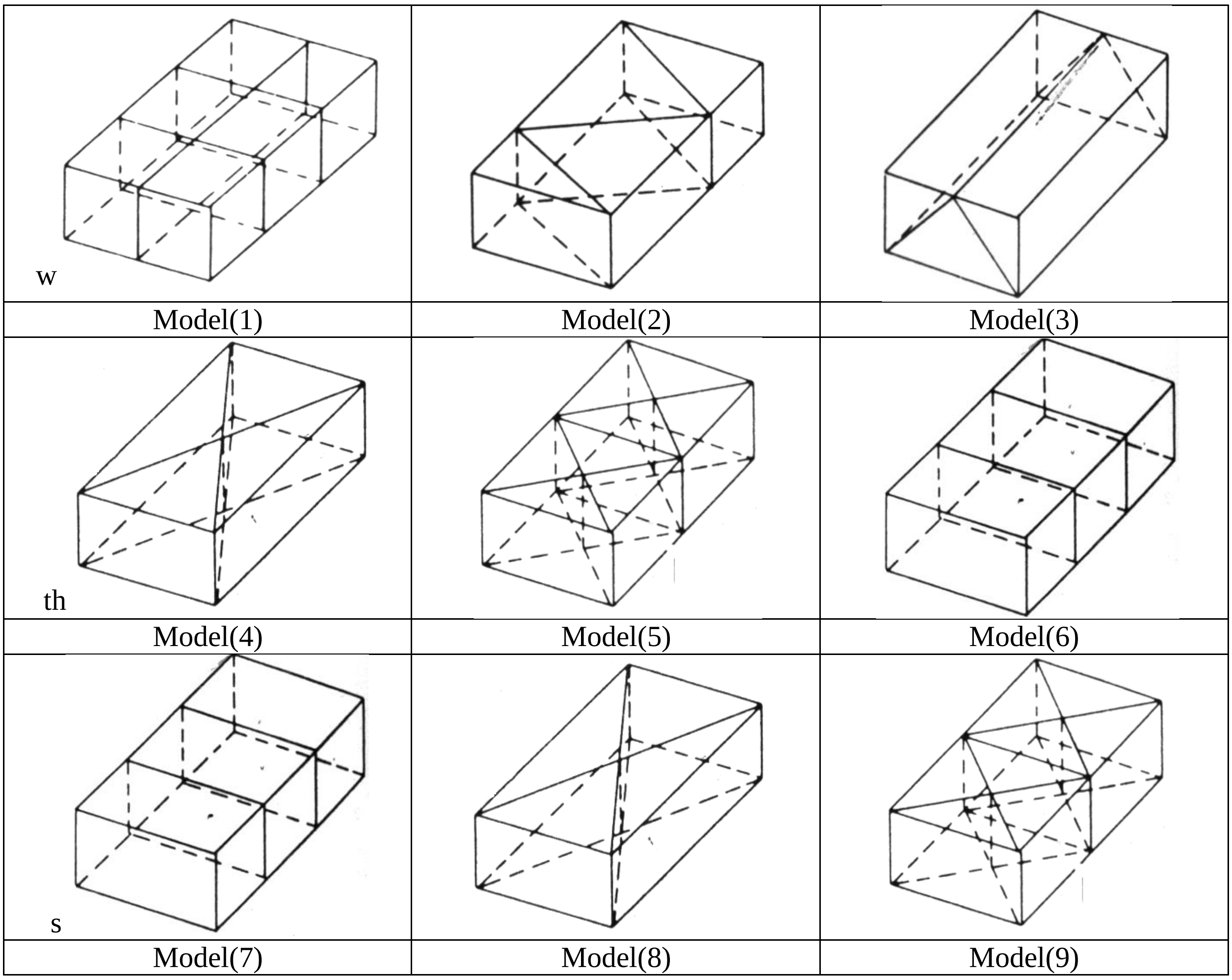
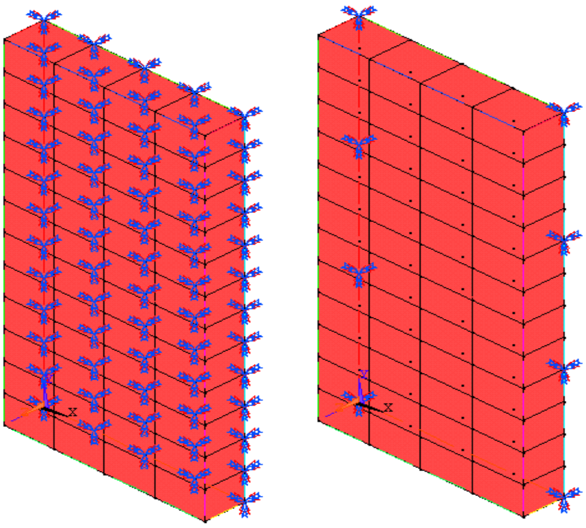


Fig. (6) :Types of Models



(7-a):bc1 (7-b):bc2

Fig.(7): Types of boundary conditions.

Table (2): Maximum displacements components results for models (1,2 and 3).

Shear N/m^2	Rake angle(0)	Trans.(mm) Rot.(rad)	BC1 Model (1)	model (2)	model (3)	BC2 Model (1)	model (2)	model (3)
40	-5	tz	-.30710E-05	-.30867E-03	-.61145E-03	-.14271E-04	-.31047E-03	-.62432E-03
		rx	-.66442E-03	.12570E-02	-.12848E-02	-.67086E-03	.12629E-02	-.12872E-02
	0	tz	-.30814E-05	-.30971E-03	-.61350E-03	-.14319E-04	-.31151E-03	-.62642E-03
		rx	-.71216E-03	.12691E-02	-.13772E-02	-.71907E-03	.12750E-02	-.13797E-02
	10	tz	-.30421E-05	-.30576E-03	-.60567E-03	-.14136E-04	-.30754E-03	-.61843E-03
		rx	-.42461E-03	.12048E-02	-.11561E-02	-.42874E-03	.12107E-02	-.11683E-02
25	-5	tz	-.29855E-05	-.30007E-03	-.59441E-03	-.13874E-04	-.30182E-03	-.60693E-03
		rx	-.39427E-03	.11785E-02	-.11298E-02	-.39809E-03	.11843E-02	-.11417E-02
	0	tz	-.29919E-05	-.30071E-03	-.59569E-03	-.13903E-04	-.30247E-03	-.60823E-03
		rx	-.44456E-03	.11895E-02	-.11428E-02	-.44887E-03	.11954E-02	-.11549E-02
	10	tz	-.29674E-05	-.42910E-02	-.59080E-03	-.13789E-04	-.29999E-03	-.60325E-03
		rx	-.26530E-03	.76745E-02	-.10957E-02	-.26787E-03	.11552E-02	-.11072E-02
130	-5	tz	-.42865E-05	-.36010E-03	-.82463E-03	-.16649E-04	-.36220E-03	-.83971E-03
		rx	-.20502E-02	-.29872E-02	-.39647E-02	-.20701E-02	-.29881E-02	-.39721E-02
	0	tz	-.47823E-05	-.36343E-03	-.86354E-03	-.17182E-04	-.36555E-03	-.87891E-03
		rx	-.23117E-02	.14780E-02	-.44703E-02	-.23341E-02	-.33693E-02	-.44787E-02
	10	tz	-.34891E-05	-.35069E-03	-.72348E-03	-.16214E-04	-.35273E-03	-.73781E-03
		rx	-.13795E-02	-.20101E-02	-.26678E-02	-.13929E-02	-.20107E-02	-.26727E-02

Table (3): Maximum displacements components results for models (4,5 and 6).

Shear N/m^2	Rake angle(0)	Trans.(mm) Rot.(rad)	BC1 Model (4)	model (5)	model (6)	BC2 Model (4)	model (5)	model (6)
40	-5	tz	-.27433E-03	-.59354E-03	-.80552E-04	-.27793E-03	-.59796E-03	-.10907E-03
		rx	.11690E-02	-.12806E-02	-.10887E-02	.11728E-02	-.12816E-02	-.10948E-02
	0	tz	-.27525E-03	-.59553E-03	-.80822E-04	-.27886E-03	-.59996E-03	-.10943E-03
		rx	.11812E-02	-.13726E-02	-.11670E-02	.11848E-02	-.13737E-02	-.11735E-02
	10	tz	-.27174E-03	-.58794E-03	-.79791E-04	-.27531E-03	-.59231E-03	-.10804E-03
		rx	.11155E-02	-.11363E-02	-.69579E-03	.11200E-02	-.11406E-02	-.69968E-03
25	-5	tz	-.26669E-03	-.57700E-03	-.78308E-04	-.27018E-03	-.58130E-03	-.10603E-03
		rx	.10906E-02	-.11105E-02	-.64606E-03	.10952E-02	-.11147E-02	-.64967E-03
	0	tz	-.26726E-03	-.57824E-03	-.78476E-04	-.27076E-03	-.58254E-03	-.10626E-03
		rx	.11020E-02	-.11232E-02	-.72847E-03	.11064E-02	-.11275E-02	-.73253E-03
	10	tz	-.26507E-03	-.40631E-02	-.77832E-04	-.26855E-03	-.57777E-03	-.10539E-03
		rx	.10610E-02	-.74937E-02	-.43473E-03	.10660E-02	-.10812E-02	-.43715E-03
130	-5	tz	-.32004E-03	-.80357E-03	-.10472E-03	-.32424E-03	-.80031E-03	-.13920E-03
		rx	-.29317E-02	-.39516E-02	-.33595E-02	-.29421E-02	-.39545E-02	-.33783E-02
	0	tz	-.32300E-03	-.84206E-03	-.11457E-03	-.32724E-03	-.84739E-03	-.14971E-03
		rx	-.33057E-02	-.44556E-02	-.37880E-02	-.33173E-02	-.44589E-02	-.38092E-02
	10	tz	-.31167E-03	-.70357E-03	-.91517E-04	-.31576E-03	-.70852E-03	-.12392E-03
		rx	-.19727E-02	-.26590E-02	-.22606E-02	-.19797E-02	-.26610E-02	-.22732E-02

Table (4): Maximum displacements components results for models (7,8 and 9).

Shear N/m^2	Rake angle(0)	Trans.(mm) Rot.(rad)	BC1 Model (7)	BC1 model (8)	BC1 model (9)	BC2 Model (7)	BC2 model (8)	BC2 model (9)
40	-5	tz	-.28536E-03	-.37291E-04	-.48032E-03	-.29265E-03	-.57132E-04	-.48847E-03
		rx	.11900E-02	-.60469E-03	.13498E-02	.11897E-02	-.65798E-03	.13576E-02
	0	tz	-.28632E-03	-.39938E-04	-.48194E-03	-.29363E-03	-.60049E-04	-.49011E-03
		rx	.12020E-02	-.64814E-03	.13706E-02	.12012E-02	-.70526E-03	.13784E-02
	10	tz	-.28267E-03	-.23997E-04	-.47579E-03	-.28989E-03	-.42610E-04	-.48386E-03
		rx	.11378E-02	-.38644E-03	.12531E-02	.11398E-02	-.42050E-03	.12613E-02
25	-5	tz	-.27741E-03	-.22307E-04	-.46694E-03	-.28449E-03	-.40473E-04	-.47486E-03
		rx	.11127E-02	-.35882E-03	.12217E-02	.11148E-02	-.39045E-03	.12298E-02
	0	tz	-.27801E-03	-.25095E-04	-.46794E-03	-.28510E-03	-.43520E-04	-.47588E-03
		rx	.11238E-02	-.40459E-03	.12421E-02	.11254E-02	-.44025E-03	.12501E-02
	10	tz	-.27573E-03	-.15157E-04	-.46411E-03	-.28277E-03	-.32649E-04	-.47198E-03
		rx	.10838E-02	-.24145E-03	-.11705E-02	.10871E-02	-.26273E-03	-.11772E-02
130	-5	tz	-.33291E-03	-.11416E-03	-.58502E-03	-.34141E-03	-.14300E-03	-.59544E-03
		rx	-.29778E-02	-.18659E-02	-.35262E-02	-.30044E-02	-.20303E-02	-.35356E-02
	0	tz	-.33599E-03	-.12866E-03	-.61313E-03	-.34457E-03	-.15884E-03	-.62382E-03
		rx	-.33576E-02	-.21039E-02	-.39760E-02	-.33876E-02	-.22893E-02	-.39865E-02
	10	tz	-.32421E-03	-.76985E-04	-.54571E-03	-.33248E-03	-.10231E-03	-.55496E-03
		rx	-.20037E-02	-.12555E-02	-.23728E-02	-.20216E-02	-.13662E-02	-.23791E-02

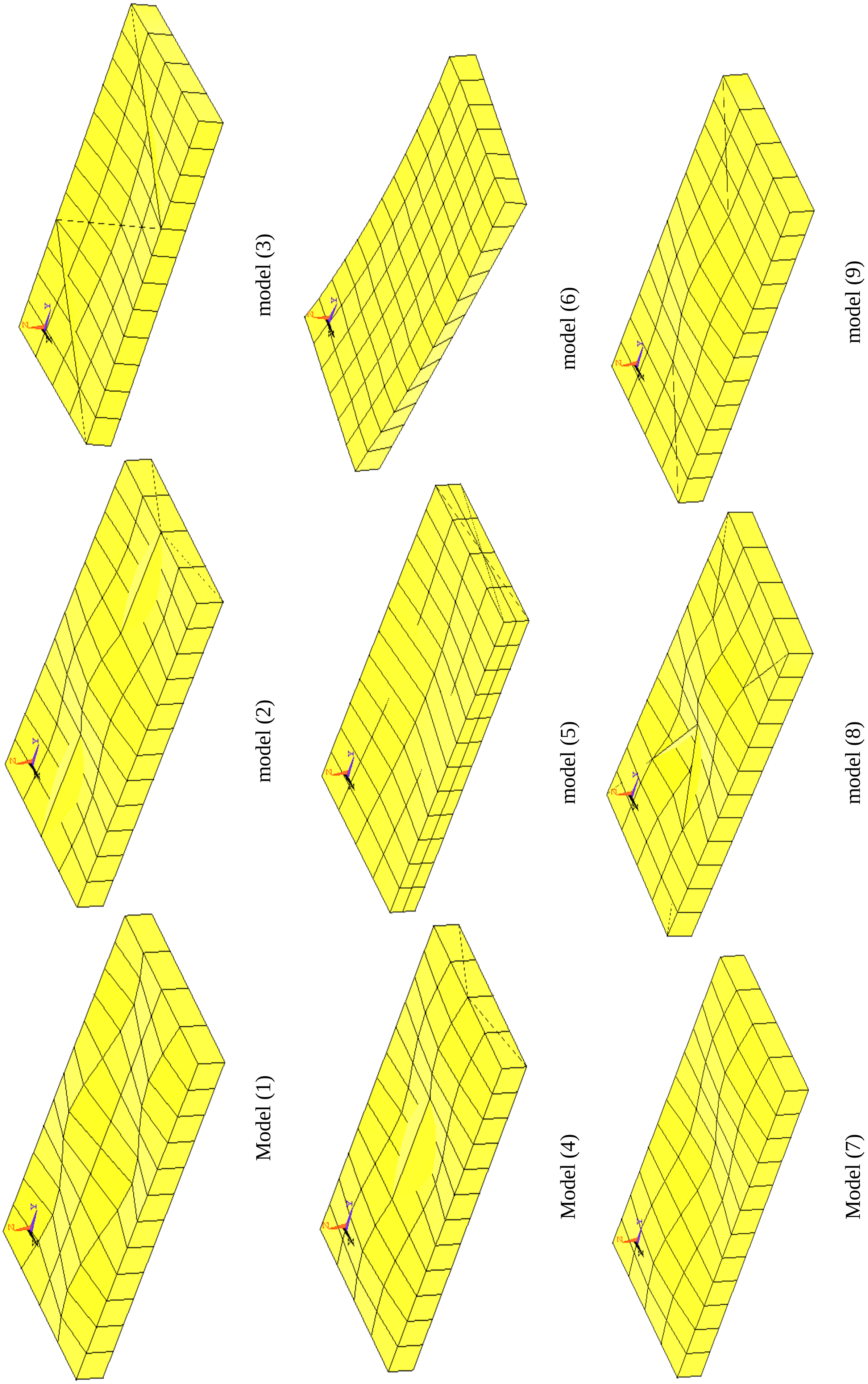


Fig.(8):First three mode shapes for the nine model of machine foundations.

Table (5): Mass for nine models.

Foundation No.	Mass(Kg)
Model (1)	561.6
Model (2)	667.68
Model (3)	524.16
Model (4)	936
Model (5)	1136.9
Model (6)	1447.5
Model (7)	873.6
Model (8)	950.6
Model (9)	987.6

Table (6): First three Natural frequencies for nine models.

Foundation No.	First Natural Mode Freq.(Hz)	Second Natural Mode Freq.(Hz)	Third Natural Mode Freq.(Hz)
Model (1)	418.32	428.71	444.88
Model (2)	192.07	223.14	239.50
Model (3)	107.10	139.06	195.79
Model (4)	219.51	240.99	267.72
Model (5)	109.62	141.46	199.52
Model (6)	603.73	650.71	672.18
Model (7)	226.03	242.49	266.25
Model (8)	277.01	299.36	315.12
Model (9)	152.90	273.31	286.68

References:

[1]A. Myers et al,"Measurement techniques for determining the static stiffness offoundations for machine tools.", Journal of Physics: Conference Series 13 ,(2005),pp.(410–413).

[2] Brogen T H N *MTIRA*"The stiffness of machine tool foundations", Research Report No. 33,1970.

[3] Brogen T H N and Stansfield F M,"The design ofmachine tool foundations", *International MTDR Conference*Proc.11th , 1970,pp. 333-353.

[4]P.Seinivasulu and C.V. Vaidyanathan,"Hand book of machine foundations",TATAMc GRAW-HILL Publishing company LTD. New Delhi,1990.

[5]Bhatia, K.G.“Machine Foundation in Power Plant and Other Industries—Case Studies”, Proceedings of the International Conference on Case Histories in Geotechnical Engineering, St. Louis, U.S.A., Vol. 2, (1984),pp. 775–779.

[6]Z. Huang and S. Hinduja," Shape optimization of a foundation for a large machine tool ", Mechanical Eng., U.M.I.S.T., P.O. Box 88, Manchester M60 1QD, U.K, Available online 1 April 2003.

[7] SHAMSHER PRAKASH VIJAY K. PURI,"FOUNDATIONS FOR VIBRATING MACHINES", *the Journal of Structural Engineering, SERC,April-May 2006*.

[8]LOREDANA THEODORA PAUN,"Theoretical study of static behavior for a multifunctional machine structure made by composite material". ISET Journal of Earthquake Technology, Paper No. 495, Vol. 45, No. 1-2, 2008,pp. 13–29,March-June.

[9]K.G. Bhatia, "FOUNDATIONS FOR INDUSTRIAL MACHINES AND EARTHQUAKE EFFECTS",ISET Journal of Earthquake Technology, Paper No. 495, Vol. 45, No. 1-2, 2008, pp. 13–29,March-June.

[10] NICOLAE–DORU STĂNESCU1, STEFAN TABACU,"Stability Analysis for Neo-Hookean Machine-Tools Foundations",WSEAS TRANSACTIONS on APPLIED and THEORETICAL MECHANICS, Issue 1, Volume 5, January 2010.

[11]NICOLAE–DORU STĂNESCUand STEFAN TABACU,"Stability Analysis for Neo-Hookean Machine-Tools Foundations",WSEAS TRANSACTIONS on APPLIED and THEORETICAL MECHANICS, Issue 1, Volume 5, January 2010.

[12]Mikell P. Groover,”Fundamentals of Modern Manufacturing , Materials, Processes, and Systems”, Prentice Hall,Upper Saddle River.New Jersey,2006.

[13]Manfred Weck and H. Bibring,"Handbook of machine tools Vol.(2)", John Wiley & Sons,1984.