Mechanical Redesign of Egyptian Made Milling Machines for Modification and Updating by using CAD

M. A. Nasser
Professor of Machine Dynamics,
Production Engineering and Mechanical Design
Department, Faculty of Engineering, Menoufia University,
Shebin El Kom, Egypt.

M. A. Asy
Associate Professor,
Production Engineering and Mechanical Design
Department, Faculty of Engineering, Menoufia University,
Shebin El Kom, Egypt.

K. M. Khedr
Associate Professor,
Production Engineering and Mechanical Design
Department, Faculty of Engineering, Menoufia University,
Shebin El Kom, Egypt.

R. M. Hashem
Instructor,
Production Engineering and Mechanical Design
Department, Faculty of Engineering, Menoufia University,
Shebin El Kom, Egypt.

Abstract: **For decades the Egyptian made machine tools have not updated since they were first introduced in 1960. The goal of this study is to introduce a working model for the modification of the present produced milling machine. Modification of the milling machine column is carried out by redistribution of the material were used in the casting of its column. The same mass of the material will be reshaped to be honeycomb-like rather than the present hollow structure. Several scenarios for the design of gearboxes are carried out. For the design and updating of milling machines, an integrated set of Ansys and Solidworks beside tailored written Visual Basic codes are used. Gearlab and VB codes are used for calculating the cutting parameters, gearboxes design, kinematic and calculating forced vibration frequencies. The redistribution of the material of the machine column gives light, stiff and well-damped structure. The redesign proposal gives versatile design and more reliable modifications.**

Keywords: Milling Machine; Modal Analysis; Redesign; Vibration; CAD; Machine Tools; Structure; Gearbox.

I.   INTRODUCTION

The goal of this study is to introduce a working model for the modification of the present Egyptian made machine tools. For decades the Egyptian made machine tools produced by Hulwan Company for Machine Tools have not been updated since they were first introduced. Hulwan Company for Machine Tools is a Factory 999 at Ein Hulwan region, Cairo. This military factory is affiliated with the Ministry of Military Production. The factory produces several civilian products. Civilian products include lathes, drilling machines, grinders, wood sawing machines, shapers, milling machines. Around the world, the great progress in manufacturing traditional machine tools and a competitive market put the stagnant or old fashion Egyptian made machine tools products under pressure. So, the modification and updating of the Egyptian made machine tools are essential. The universal milling machine 6P80 is selected for this study.

Machine tools fundamentals, theory and application are studied in detail [1-7]. Computer-aided design is widely used in machine tool design and dynamic analyses by using Ansys software [8 - 9]. References [10 - 44] concerned with the study and application of vibration, modal analysis, static and kinematic analyses of machine tools structures and gearboxes. Several references, [10 - 27] studied milling machines structure's static and dynamic analyses, as well as kinematic analysis, has received a lot of attention. Other applications of computer-aided design on other machine tools such as shaper [28] & [29], Gaint [30] are done. References from [32] to [44] studied in detail the static and dynamic behavior of machine tools gearboxes. Bearings as one of the most important parts in machine tools are considered, as the bearing frequencies arising from some faults represents excitation frequencies, [45].

A comparison of the experimental obtained from experimental modal analysis and theoretical data obtained from Ansys revealed that the natural frequencies and corresponding mode shapes are in good agreement for the gearbox structure model. The well correlate results between experimental and theoretical prove the validation of Ansys to construct the good design, [32]. The validated FE model by Ansys could be used in the design process with greater confidence that the calculated mechanical properties of the properties' machine tools structures and/or parts. It is reliable. The accuracy of the results highly depends on user experience and knowledge. The designer should know the physics of his problem and use his solid engineering judgment and intuition to understand what is going on in his design.

This study started with the calculation of cutting speed and power required for the milling process. The conventional methods of calculation with the associated equations of metal cutting in the milling process as well as the nomogram, [3]. A step forward was given by building electronic nomograms and special-purpose calculators in the VB6 language. These computerized tools make the design process easy, faster, accurate and more reliable. Soon after this, the structure of the machine was modeled in Solidworks and dynamically analyzed by using Ansys software. In this paper, the milling machine structure is modified by redistribution of the mass of

the machine column by using part of this material as ribs in the form of honeycomb, as the reference [14] & [31] done before. Moreover, the study and modification of the milling machine main and feed gearboxes were done. Three scenarios of optimum design of the main gearboxes were proposed and analyzed for the design parameters of gears and shafts. These results of other models of the main gearbox were obtained by using a special code written by a student in (VB6). Moreover, these gearboxes were modeled in Solidworks software and analyzed by using Ansys software. Generally, the results obtained from an integrated set of Ansys and Solidworks for design modification, besides Gearlab [39] and tailored written Visual Basic codes for calculating cutting process and gearboxes design, kinematic and forced vibration frequencies give versatile design tool and reliable design modifications.

II. CUTTING FORCES, TORQUE & POWER

A. Analysis

Software for calculating the mean power and cutting force in slab and milling processes using VB6 is written, tested and applied. The cutting parameters are cutter diameter, cutter number of teeth, spindle revolution, feed rate, cutting depth and width. The cutting speed is calculated from the following:

$$V = \pi.d.n \qquad (1)$$

where n is spindle revolution number, d is the cutter diameter and V is cutting speed.

The cutter's angle of rotation relies on the cutter diameter and the depth of cut and it can be calculated as follows [5]:

$$\varphi_t = 360°/n_t \qquad (2)$$

$$\cos\varphi_s = 1 - \frac{2a}{d} \qquad (3)$$

where φ_t is pitch angle of the cutter teeth, φ_s is the angle of rotation of the cutter, n_t is the number of teeth of the cutter, d is cutter diameter and a is the depth of cut.

When the cutter tooth has entered the workpiece material by an angle φ the chip thickness as follows can be calculated:

$$c_t = s_t. \sin\varphi \qquad (mm) \qquad (4)$$

where s_t is the feed per tooth (mm).
The feed per minute is equal to:

$$s = s_t. n_t.n \qquad (mm/min) \qquad (5)$$

And the chip thickness:

$$c_t = \frac{s}{n.n_t}. \sin\varphi \qquad (mm) \qquad (6)$$

Then maximum chip thickness can be calculated as follows:

$$x_{max} = \frac{s}{n*n_t}. \sin\varphi_s \qquad (mm) \qquad (7)$$

where s is the feed rate and x_{max} is the maximum chip thickness.
The middle chip thickness can be calculated as follows:

$$h_m = \frac{s}{n*n_t}. \sin\frac{\varphi_s}{2} \qquad (mm) \qquad (8)$$

The mean power required at cutting edge can be calculated as follows:

$$N_m = K_m.a. b. s \qquad (kw) \qquad (9)$$

where K_m is the specific cutting energy (N.mm^2).
The mean cutting force acting at the cutter can then be calculated:

$$P_{tm} = \frac{N_m}{V} \qquad (N) \qquad (10)$$

As well as the mean torque:

$$T_m = \frac{P_{tm}.d}{2} \qquad (N.mm) \qquad (11)$$

B. Software Code Construction

Nomogram Software is a code constructed by using Visual Basic language. The cutting parameters; workpiece material, cutter speed, a width of cut, depth of cut and feed rates. The relevant parameters either to be read from the given hard copy nomogram, [3] or to be calculated from the given computerized nomogram. The computerized nomogram is written in VB6. Visual Basic code is constructed to be used in slab and face milling processes metal cutting calculations. The code uses the horizontal and vertical scroll bars. Scroll Bar Events are used in this code. The designer uses this event to retrieve the cutting parameter after any changes in the scroll bar. As the user changes the scroll bar value, both the cutting force and power, you have to calculate this, in integer format and animated lines on the electronic nomogram displayed on the computer screen as shown in Figs. 1 &2. The formula for cutting force and power is given in the analysis. Scroll event-triggered continuously whenever the scroll box is being moved. Clicking an end arrow increments the scroll box a small amount, clicking the bar area increments the scroll box a large amount, and dragging the scroll box (thumb) provides continuous motion. Using the properties of scroll bars, the designer can completely specify how one works. Another code is written to calculate the vertical and horizontal milling processes as shown in Fig 3. The user will input in text boxes the cutting parameters and the specific cutting energy of the workpiece material.

The outcomes of the code will appear text boxes. Case study of cutting speed V= 30480 (mm/min), tool diameter d= 100 (mm), number of teeth of the cutter n_t = 8, width of cut b=67.35 (mm), depth of cut a= 4.06 mm and feed rate s= 152.46 (mm/min) is used in this article.

C. Results and Discussion

The two Softwares were built and tested for calculating the mean cutting forces and the mean required power for the milling machine. The results obtained from the software were compared with those obtained from manual calculations by using the calculator and nomograms given in textbooks [3]. Many cases of cutting conditions were investigated. Table (1) presents the power & cutting force estimated by manual nomogram (hard copy), nomogram software (VB) and equations software (VB) for steel alloy, carbon steel, light alloy.

Table (1) The Power & Cutting Fore Estimated By Different Methods.

	Power (kw)		
	steel alloy	carbon steel	light alloy
Manual Nomogram (hard copy)	3.8	2.2	1
Nomogram Software (VB)	3.8	2.3	1
Equation Software (VB)	3.8	2.2	1
	Cutting force (N)		
	steel alloy	carbon steel	light alloy
Manual Nomogram (hard copy)	7480	4331	1969
Nomogram Software	7480	4331	1969
Equation Software	7480	4331	1969

The same results are obtained in the three calculation methods of the cutting parameters. These results proved that the proposed given computerized nomogram and calculators written in VB(6) are accurate, faster, reliable and expandable tool for more new parameters to be added.
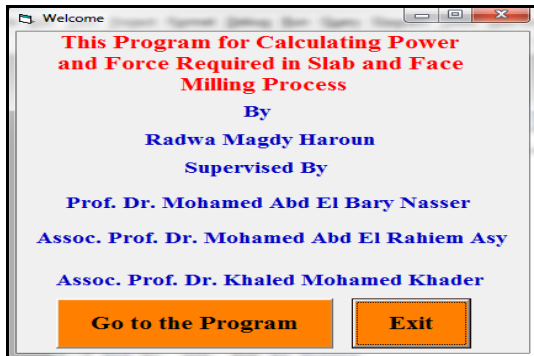

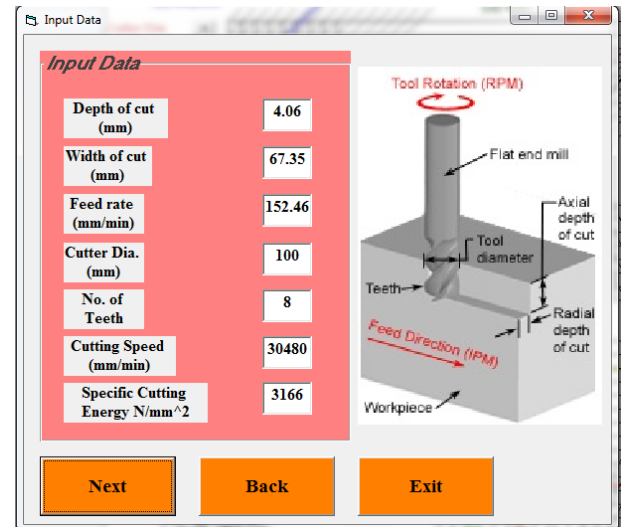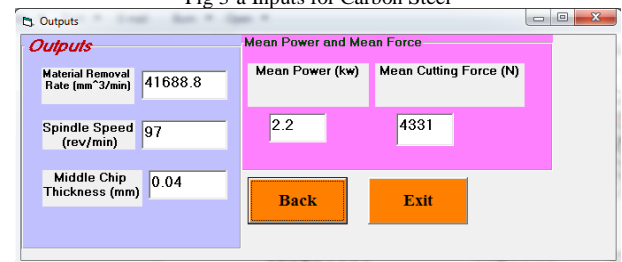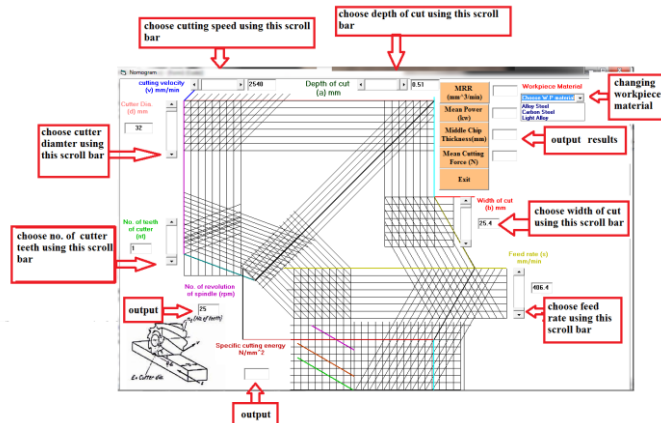Fig. 1. Welcome Screen of the Milling Nomogram.

Fig 2 a Screenshot of the Computerized Nomogram.
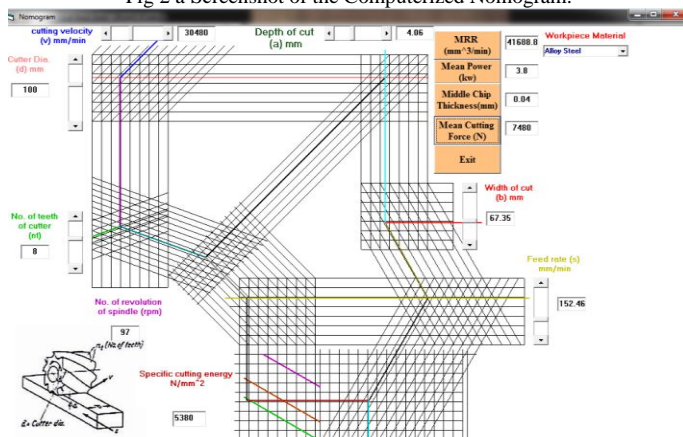
Fig 2-b Case Study Carried out on the Nomogram.

Fig. 2. Slab & Face Milling Nomogram.

Fig 3-a Inputs for Carbon Steel

Fig 3-b- Results of Carbon Steel.

Fig. 3. Milling Process Calculator.

III. STRUCTURE MODIFICATION

This study presents four different suggested structure modifications for the milling machine to enhance its performance. These modifications can increase the mitigation of harmful vibration. Harmful vibration adversely affects the milling process. The mass of the milling machine column is 1175 kg. 165 kg is subtracted from the base of the column and added again in the form of honeycomb form to be as reinforcement ribs added to the core of the machine column. The redistribution and the reshaping of the column are shown in Fig -5. The current height of the machine base is 130 mm. This height reduced to 80 mm through the suggested four modifications. Figure (5) summarizes the four suggested modification scenarios. The first one is adding a 6×6 matrix of ribs with the same thickness equals to (10 mm). Also, the second one is adding a 5×5 matrix of ribs with the same thickness is (12 mm). While the third one is adding a 3×3 matrix of ribs with a thickness equals to (12 mm) in addition to one diagonal cross ribs with the same thickness equals to (12 mm). Finally, the fourth one is adding a 3×3 matrix of ribs with a thickness equals (7 mm) in addition to 16 internal diagonal cross ribs with the same thickness (7 mm). The surface area of the base and column before adding ribs is 5.43 m^2. The surface area are 9.33, 8.65, 8.29 & 11.11 m^2 for modifications #1 to #4. This means that the increase in surface area are1.72, 1.59, 1.53 & 2.05 times higher than the original one. This increase in the surface area could play an important role in structural heat transfer.

A. Static analysis

Static, modal and frequency analyses were done by using Ansys software. The evaluation of the static loads' effect on structure deformation is {x}, this deformation can be evaluated as follows:

$$[Q]\{x\}=\{F\} \qquad (12)$$

where; {F} denotes to the static force applied to the spindle nose (which equals 4000 N as assumed value) and [Q] is a constant. The displacements {x} can be evaluated under assumptions as; small deflection theory is used, linear elastic material behavior is assumed, there are no varying forces and inertial effects (mass and damping) via considering linear elastic material behavior in addition to some nonlinear boundary conditions may be included.

The total deformation (U_{total} in µm) can be calculated as follows;

$$U_{total} = \sqrt[2]{u_x{}^2 + u_y{}^2 + u_z{}^2} \qquad (13)$$

where; (u) are the deformation components in the orthogonal directions (x, y, and z).
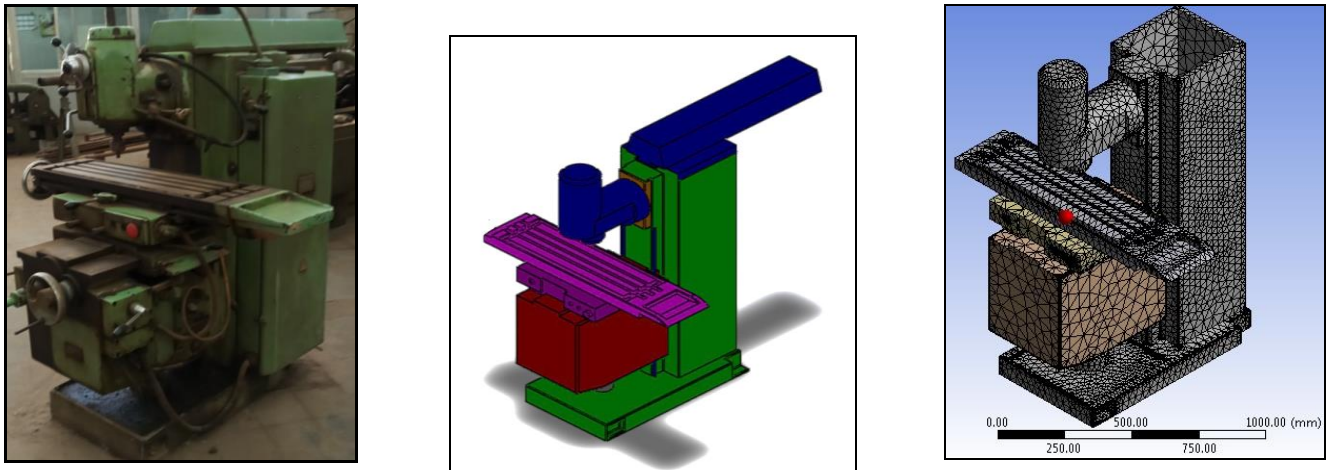
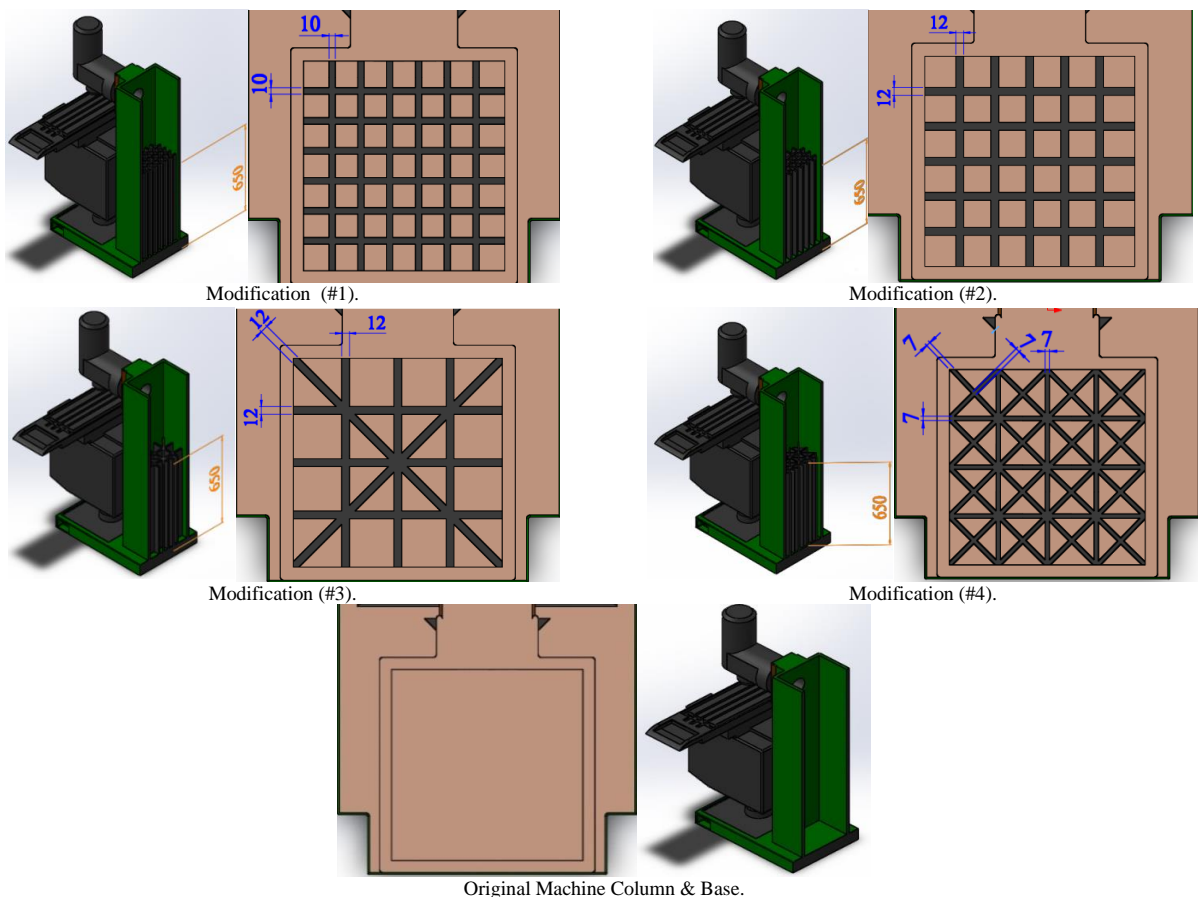**Fig. 4. Physical Universal Milling Machine. Solidworks Model & FEM Ansys Model.**

Modification (#1).

Modification (#2).

Modification (#3).

Modification (#4).

Original Machine Column & Base.

Fig. 5. The Original and Proposed Modifications in the Universal Milling Machine.

B. *Results of Static Analysis*

Higher rigidity, higher natural frequencies, and lightweight machine tool structures are the best from static, dynamic and feasibility points of view. These goals have to be considered while designing the column of the milling machine. Stiffness is the degree to which the structure resists deformation as a response to an applied force. The stiffness is defined as the ratio between the applied force and deflection. The stiffness of a structure is of major significance in machine tool structures, so the young's modulus is the most important property considered when selecting a material. Since the deformation in machine tool structures is undesirable, a high modulus of elasticity is recommended. Higher stiffness mitigates the milling machine's deformation. The results show that the maximum deflection occurs at the spindle head. Hence, the spindle head can be marked as the weakest location from the characteristic of static behavior.

The static deformation obtained from Ansys is given in Table (2). the percentage change is calculated from the following form:

Rate of Improvement (%) = [Modified value - Current value] / Current Value x 100.

From table (2) it is clear that the maximum static deformation is reduced from 145.1 μm for the present machine tool structure to 102.9, 103.7, 104.2 & 99.7μm for the modifications #1 to # 4. The less deformation the higher stiffness. The maximum deformations at spindle head for modifications #1 to #4 decreases by 29%, 28.5%, 28% & 31.2% lower than the original one. This means that modification # 4 reduced the milling machine structure by 31.2%.

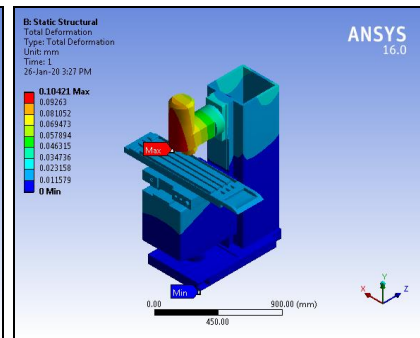**Table (2) The Maximum Deformations at Spindle Head.**

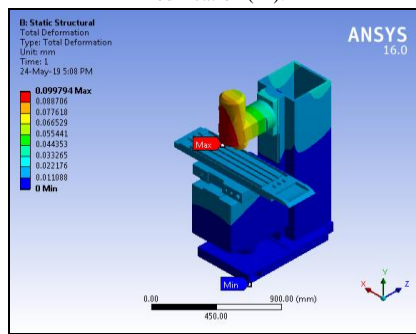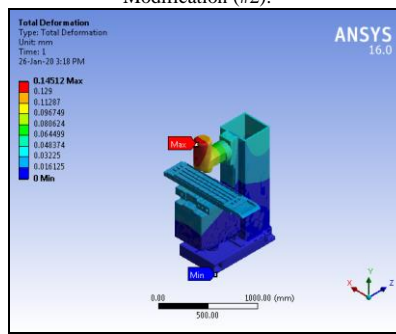| Current	Deformation (μm)	Reduction in Deformation (%)
Current Machine	145.1	-
Modification # 1	102.9	29
Modification # 2	103.7	28.5
Modification # 3	104.2	28
Modification # 4	99.7	31.2
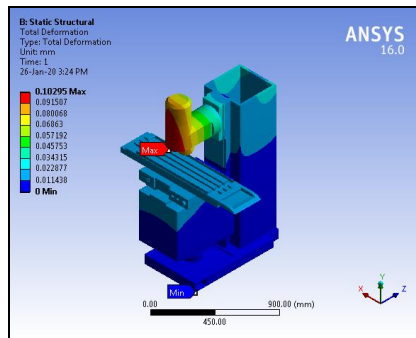
Modification (#1).
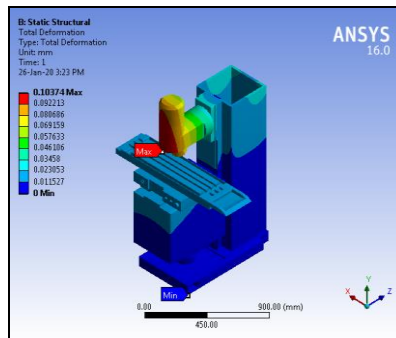
Modification (#2).

Modification (#3).

Modification (#4).

Original Column base.

Fig. 6. The Total Deformation Shape of the Orignal & Modifed Structures.

C. *Modal analysis*

Modal is carried out on the milling machine column. When exposed to loads or displacements, all true physical buildings act dynamically. The equations of motion can be written depending on solving the Eigen-Values problem's equation of multi-degree of freedom systems as follows;

$$[M]\{\ddot{u}\}+[C]\{\dot{u}\} + [K]\{u\} = \{F(t)\} \qquad (14)$$

where; {u} denotes to the Eigenvectors, [M] is the mass matrix, [C] denotes to the damping matrix, [K] is the stiffness matrix and F(t) is the applied force. The previous equation can be rewritten in the following form by assuming free vibrations and ignoring damping;

$$[M]\{\ddot{u}\}+ [K]\{u\} = \{0\} \qquad (15)$$

Also, the following equation can be formulated considering the harmonic motion, i.e., u=U sin (ωt), as follows;

$$([K]- \omega^2[M])\{u\} = \{0\} \qquad (16)$$

where; the angular frequency (ω) is equal to the square roots of the eigenvalues. Also, the structure's circular natural frequencies *f* (in cycles/sec) can be calculated as follows;

$$f= \omega /(2\pi) \qquad (17)$$

The eigenvectors {u}$_i$ represent the mode shape when the system is vibrating at f$_i$, where i=1,2,..,n using Ansys software.

D. Modal Analysis Results

Modal analysis is the study of the dynamic properties, "mode shapes, natural frequency and damping", of systems in the frequency domain. Structural modal analysis of milling machine structure uses the overall mass and stiffness of the structure to find the modal parameters. The animated display of the mode shapes of the milling machine column is very beneficial to study and understand the vibration and dynamic behavior. Machine tool designer tends to learn from modal analysis, how potential influence could happen in product quality produced by using the machine tool. Modal analysis is also important in the machine tool structure where the engineer should attempt to keep the exciting natural frequencies away from the frequencies of the machine tool structure and its components. From Tables (2) and (3), the dynamic deflection values of the milling machine are smaller than the static deflections. From Table (3) and Fig -7, it is clear that all modifications have lower dynamic deformations. Table (3) shows the maximum deflection (μm): frequency, location, and direction. The detailed study of such parameters is essential to understand the milling machine column dynamic behavior. As can be seen in Table (4), the natural frequencies of all ten first modes of all modifications are higher than those of the original structure. The first mode natural frequencies of the original structure and modifications #1 to #4 are 73.8, 111.2, 110.7, 109.3 & 112.6 Hz respectively. The improvement is by increasing the fundamental natural frequency of the proposed scenarios by 50.6, 50, 48 & 52.5% higher than those of the original structure. Modification #4 is the best stiff structure. So, the fourth modification is the versatile design against harmful vibration.

Table (3) The Maximum Deflection (μm): Frequency, Location & direction.

	Location of Max. Deformation	Maximum Deflection (μm)	FRF Direction
Present Machine	Spindle head	22.87 (@ 73.8 Hz)	X-direction
	Table	0.262 (@ 247 Hz)	Y-direction
	Spindle head	6.16 (@ 121Hz)	Z-direction
Modif. # 1	Spindle head	1.91 (@170 Hz)	X-direction
	Column	0.447 (@ 442 Hz)	Y-direction
	Column	0.287 (@ 442Hz)	Z-direction
Modifi. # 2	Spindle head	1.77 (@ 111 Hz)	X-direction
	Spindle head	0.103 (@ 145 Hz)	Y-direction
	Spindle head	0.191 (@145 Hz)	Z-direction
Modifi. # 3	Spindle head	3.61 (@169Hz)	X-direction
	Table	0.338 (@340Hz)	Y-direction
	Spindle head	0.288 (@215Hz)	Z-direction
Modifi. # 4	Spindle head	1.65 (@ 174Hz)	X-direction
	Spindle head	0.469(@ 146 Hz)	Y-direction
	Spindle head	0.867(@ 146 Hz)	Z-direction

The mode shapes of the milling machine and the milling machine with the modifications added to the machine column are thoroughly investigated to see haw significance in structural behavior is obtained. Fig 8, shows the first mode shape of the original milling machine and the milling machine with modifications. The milling machine can safely use wide cutting conditions as the cutting speed without dangerous resonance occurrence. Generally, the fourth modification has the highest natural frequency among all modifications, the

widest range of natural frequencies, and the minimum value of deformation. These good dynamic results are a result of reinforcement achieved by redistribution and reshaping the structure through stiffener elements added, which play as multiple ribs.

Table (4) The Natural Frequencies of the Original & the Modified Machine Columns.

| #	Frequencies and Rate of Improvement of the Current Machine Column and Modifications # 1 : 4.								
	Current	#1	%	#2	%	#3	%	#4	%
1	73.8	111.2	50.6	110.7	50	109.3	48	112.6	52.5
2	97.2	145.2	49.3	144.9	49	144.2	48.3	146	50.2
3	120.8	170	40.7	169.4	40.2	168.9	39.8	173.8	43.8
4	134.4	217.7	61.9	217.5	61.8	214.9	59.8	218.4	62.5
5	156.4	243.6	55.7	242.2	54.8	242.3	54.9	250.6	60.2
6	245.1	340.8	39	340.7	39	339.9	38.6	341.3	39.2
7	259.5	369.2	42.2	368.8	42	367.1	41.4	372.5	43.5
8	281.5	426.9	51.6	426.5	51.5	425.8	51.2	429.1	52.4
9	317.4	442	39.2	441.1	38.9	446.7	40.7	468	47.4
10	336.33	478.6	42.3	480.5	42.8	477.6	42	487.6	44.9
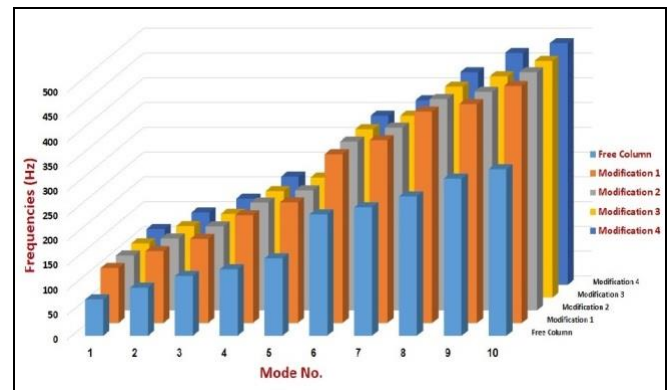
Fig. 7. Natural Frequencies of Current Design and Modified Columns.

E. Harmonic Response Analysis

By selecting the FRF Analysis option, the integrated harmonic analysis tool opens. The frequency response function at X, Y, and Z-direction of the original and the four modifications of the milling machine column are obtained. Many risky effects on the machine tools can be avoided reliant on the harmonic analysis which depends on vibration analysis. Moreover, the Frequency Response Function (FRF) can be considered as an effective design tool to help the designer. Therefore, the designer can depend on the (FRF) to provide him with the necessary details about the frequency of the high response and the time of its occurrence.

The harmonic response equation can be expressed as follows;

$$(-\Omega^2[M] + j\,\Omega[C] + [K])\{x_1 + j\,x_2\} = \{F_1 + j\,F_2\} \qquad (18)$$

$$F = F_o \sin\Omega t \qquad (19)$$

where; (Ω) denotes to the excitation frequency, F_o is the amplitude of the force.

The small deflection theory is applied, the system undergoes a linear elastic material behavior associated with the damping effect via ignoring the nonlinearities. The applied load {F} and its response {x} have a sinusoidal function at a given frequency (ω). All frequency response spectra are obtained and analyzed.

The sample is shown in Fig -9 which shows the frequency response functions of the original & modified structures in the y-direction. It is helpful in dynamic analyses of the milling machine Both excitation and structure natural frequencies must have different values for avoiding the risky response effects.

Sources of forced vibration in machine tools generated mainly from the rotating elements of the machine itself or transmitted from near vibrating machines. Electric motor, belts, pulley, shafts, bearings, gears, spindles are probably sources of forced vibrations.

Mode 1 at f=111.2 Hz, Modification (#1).

Mode 1 at f=110.7 Hz, Modification (#2).

Mode 1 at f=109.3 Hz, Modification (#3).

Mode 1 at f=112.6 Hz, Modification (#4).

Original column base (Free webbed) Mode 1 at f=73.8 Hz.
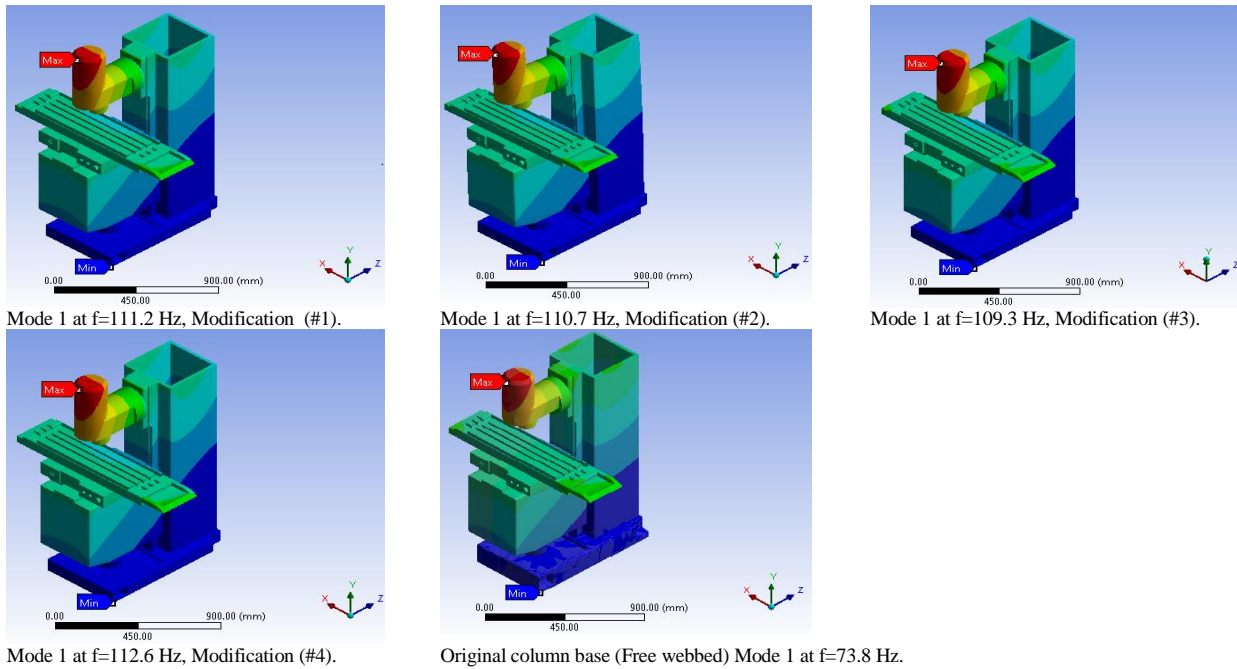
Fig. 8. The Mode Shapes of the Orignal & Modified Structures

Modification (#1).

Modification (#2).

Modification (#3).

Modification (#4).

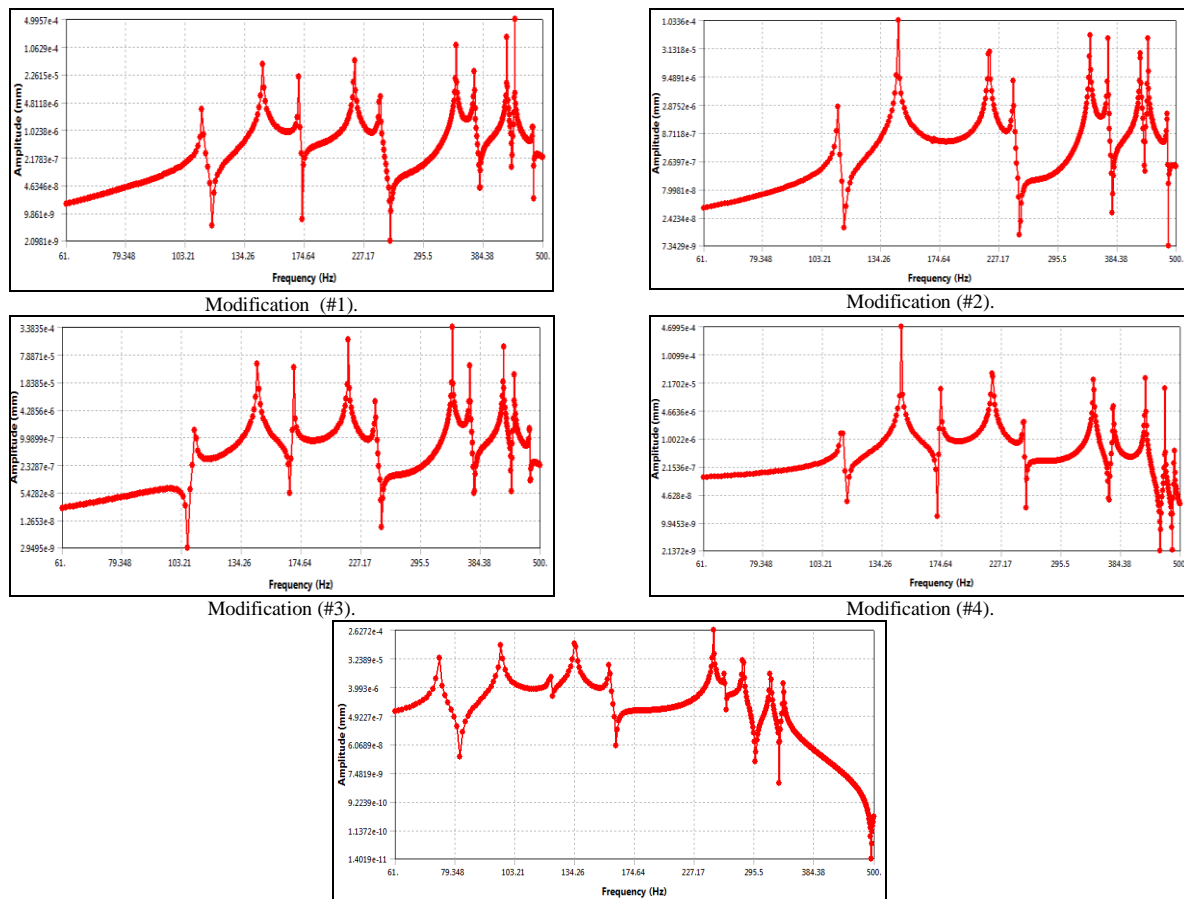Original column base (Free webbed).

Fig. 9. The Frequency Response Functions of the Orignal & Modified Structures in Y-Direction.

IV.   GEARBOX DESIGN

A.   Gearboxes Design Calculations:

The multi-Stage main cutting gearbox is designed in several scenarios. Gearboxes are used to obtain a wide variety of speeds. The multi-stage gearbox has a large size and a high number of elements. The following steps will illustrate the calculations necessary for the gearbox design process. In machine tools, speed regulation is achieved by more than one method using different progression such as geometric, harmonic, logarithmic and differential progression. The best method to reach the proximity of speed is the differential progression. However, as an international standard, geometric progression is widely used in machine tools. Speeds are valued as the following in geometric progression:

First speed:

$$n_1 = n \qquad (20)$$

Second speed:

$$n_2 = \frac{n}{r} \qquad (21)$$

Third speed:

$$n_3 = \frac{n}{r^2} \qquad (22)$$

The general formula for calculation speed is:

$$n_z = \frac{n}{r^{z-1}} \qquad (23)$$

where z is the speed number, n is the motor speed and r is the geometric progression base.

Theoretical rotational speeds are determined according to standard geometric progression base (r). the values of progression base are 1.06, 1.12, 1.26, 1.41, 1.58, 1.78, and 2.

B.   Speed Chart

The speed chart is described as a flow chart describes the distribution of gearbox transmission ratios which determines the probability equation to design the gearbox. It has the following form:

$$z = p_1(x_1)p_2(x_2)p_3(x_3) \dots \dots p_u(x_u) \qquad (24)$$

$$x_1 = 1 \qquad (25)$$
$$x_2 = p_1 \qquad (26)$$
$$x_3 = p_1 p_2 \qquad (27)$$
$$x_u = p_1 p_2 p_3 \cdots p_{u-1} \qquad (28)$$

where p is the number of speed steps and x is the characteristic of the transmission group.

C.   Kinematic Diagram

It is a diagram shows the layout of gears and bearings in the gearbox according to the chosen arrangement equation.

D.   Gears Specifications and Reduction Ratios

The reduction ratios are calculated using the following equations:

$$i_x = \frac{Z_n}{Z_{n+1}} \qquad (29)$$

where; n=2x – 1 and Z is the number of the gear teeth.
The meshed gear group module is calculated as follows:

$$m = \sqrt[3]{\frac{0.64 . M_t . K_d}{y . \psi . \sigma_b . Z_{min}}} \qquad (30)$$

where; M_t is the torsion moment, ψ is the gear width factor, σ_b is the bending stress, y is the gear teeth factor and K_d is a constant.

$$D_{ad} = m(Z + 2) \qquad (31)$$
$$D_d = m(Z - 2.5) \qquad (32)$$
$$D_{pi} = m . Z \qquad (33)$$
$$b = m . \psi \qquad (34)$$

where D_{ad} is the addendum diameter, D_d is the dedendum diameter, D_{pi} is the pitch diameter and b is the gear width.

E.   Actual, Theoretical Speeds and Speed Errors

Actual speeds are produced in the spindle of machines due to the motor speed and the mechanical movement of the meshed gears. Actual speeds are calculated by the motor speed, belt-pulley reduction ratio and the designed gears using the speed chart. Theoretical speed can be obtained from standard tables and it is an approximate value of the actual speed. The error between speeds (theoretical and actual) produce is a result of some approximation in design processes that must be less than permissible errors, all can be calculated as follows:

n_{th}: is achieved from standard tables.

n_{act}: is evaluated from the speed chart.

$$n_{error} = \frac{n_{th} - n_{act}}{n_{th}} \qquad (35)$$

$$\text{Allowed error} = \pm 10 \, [r-1] \, \% \qquad (36)$$

F.   Numerical Case Study of Main Gearbox Design

A multi-stage gearbox with 12 speed and probability 6x2 has the following data: n_{max} = 2240 rpm, n_{min} = 50 rpm, Nm = 3 kw, D_p = 189 mm, d_p = 90 mm, n_m = 1720 rpm, m = 2 mm, $\eta_{g,bp,b}$ = 95%, where D_P is the diameter of driven pulley and d_p is the diameter of the driver pulley.

The progression value (r) is estimated from the following equation:

$$r = \sqrt[z-1]{\frac{n_{max}}{n_{min}}} = \sqrt[11]{\frac{2240}{50}} = 1.41 \qquad (37)$$

$$i_{max} \leq 2 \qquad\qquad i_{min} \geq \frac{1}{4}$$
$$r^x \leq 2$$
$$1.41^x \leq 2 r^x \leq 4 \qquad\qquad \frac{1}{r^x} \geq \frac{1}{4}$$
$$x_{max} \leq 2 \qquad\qquad x_{min} \leq 4 \qquad (38)$$
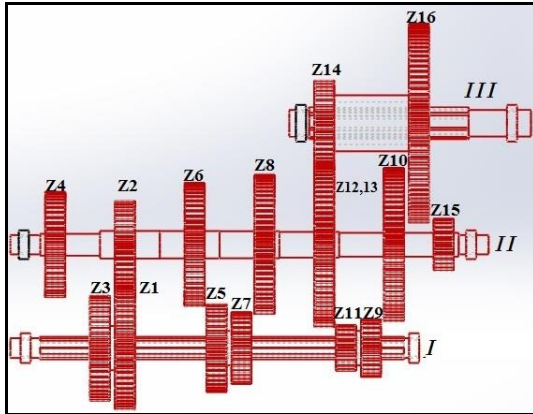
Fig. 10. The Gearbox Kinematic Diagram.
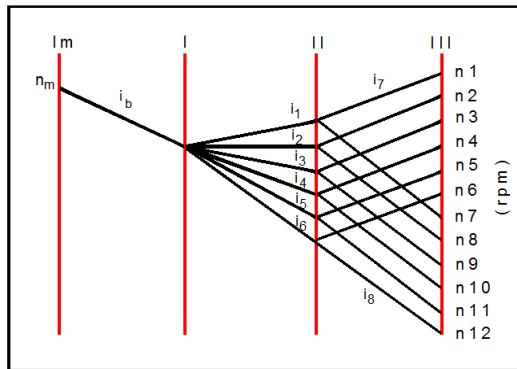
Fig. 11. Speed Chart of Main Gearbox.

G. The Software

The software is constructed to design a multi-stage gearbox of probability (6x2) to give (12) speed. The design includes speed chart, gears design, actual, theoretical speeds, speed errors and gear meshing frequencies. The Input data is for motor and gearbox speeds, the motor speed 1720 rpm, maximum speed = 2240 rpm, minimum speed = 50 rpm, gear module = 3 mm, driver pulley diameter = 90 mm.

The software is written in Visual Basic 6 language, it works as a graphic user interface (GUI) to be used easily. The gearbox kinematic diagram, (Fig -10), speed chart of the main gearbox, (Fig -11), the welcome screen of the gearbox design program, (Fig -12), the input data, (Fig-13), the gearbox speed chart, (Fig -14), the gears data obtained from the software, (Fig -15), theoretical, actual and speeds errors of the gearbox, (Fig -16), the gear meshing frequencies, (Fig -17) are given.
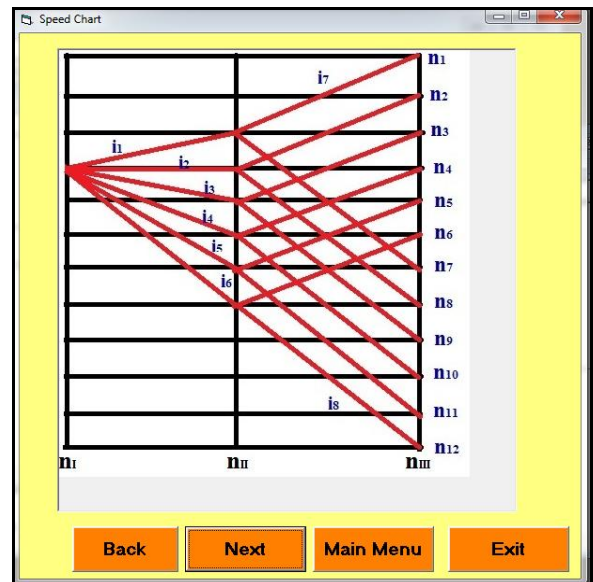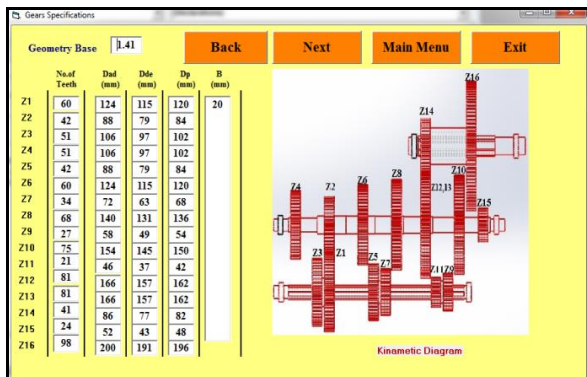

Fig. 12. Welcome Screen of Gearbox Design Program.

Fig. 13. The Input Data.

Fig. 14. The Gearbox Speed Chart.

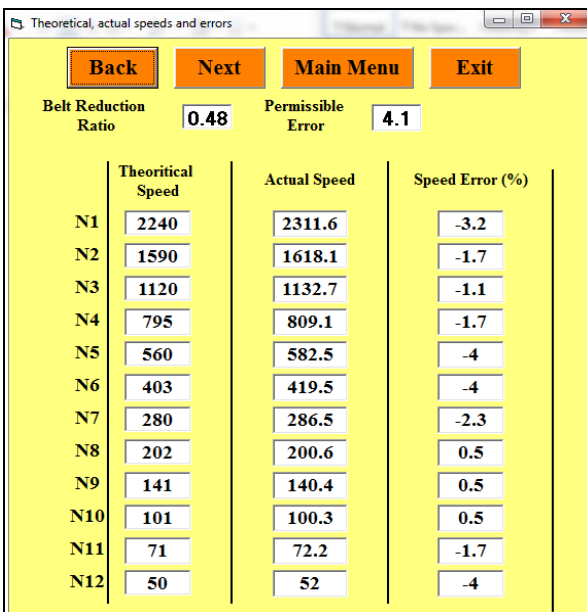**Fig. 15. The Gears Data obtained from the Software.**

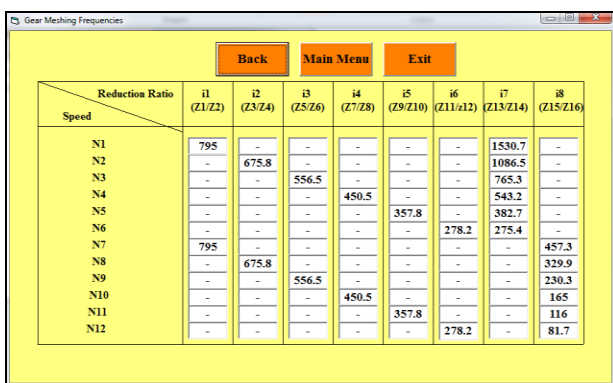**Fig. 16. Theoretical, Actual and Speeds Errors of the gearbox.**

Fig. 17. The Gear Meshing Frequencies.

The 1st probability (4x3) of 12 speeds main gearbox is presented as a case study. The solution of the gearbox consists of three steps. In the first step, choosing the number of speeds and motor speed, maximum and minimum speed, motor power and progression ratio. In the second step, entering the minimum number of teeth, pressure angle, efficiency, width factor and strength of gears, and shaft strength. The third step is used to enter width and efficiency of bearing, inclined angles and reduction ratio of the belt, pulley diameter and width and distance between pulleys axis. The input data for gearbox design, [39], (Fig -18), theoretical, actual speeds and percentage errors between them (Fig -19), gears specifications (Fig -20), speed and flow chart. (Fig -21), gearbox shafts design (Fig -22) are given to show how a friendly user software is.

H. Modal Analysis of the main gearbox

The kinematic diagrams of the minimum and maximum speeds gear meshing of the milling machine gearbox of probability 6x2 is given in Fig -23, while The First Mode Shape of them are shown (Fig -24). The gearbox assembly geometry modeled in Solidworks was imported to Ansys software for studying the dynamic performance. The bearing supports were assumed as a boundary condition for the gear components, which allows rotational motion along the shaft axis but restricts axial motion and radial motion.

I. Harmonic Response Analysis of the Gearbox

The gearbox exposed to defects in the gears or shafts due to varying magnitude loading. According to the previous calculations, a moment was applied as a boundary condition at the input shaft of each model of the main gearbox. Table (5) presents the natural frequencies (Hz) of the gearbox with the probabilities (6x2), (4x3) & (3x2x2). The purpose of the harmonic response analysis is to measure the maximum deflection and its corresponding frequency also determining the resonance frequencies. The maximum deflection of 2.33 μm of the 6x2 probability occurs at x-direction at a frequency of 276 Hz at the maximum speed. The maximum deflection of 2.3 μm of the minimum speed is found in y-direction at frequency 592 Hz. The maximum deflection and the corresponding natural frequencies of the gearboxes with the the Probabilities (6x2), (4x3) &(3x2x2) are given in Table (5). The results show that the maximum deflection 0.702 μm at y-direction at 188 Hz and 3.16 μm at z-direction at 230 Hz of the maximum and minimum speed respectively. The maximum deflection 13.51 μm at y-direction at 475 Hz and 5.59 μm at z-direction at 210 Hz at the maximum and minimum speed respectively for the probability 3x2x2.

Direct Speed GearBox Input Data

GearBox Speeds Data

No. of GearBox Speeds = 12

Speeds — Motor Speed = 1720 Rpm — Motor Power = 4 HP — F = 1.41

Progression — Max. Speed = 2240 Rpm — Min. Speed = 50 Rpm — Read Data

Optimum Speed Probability — ☑ 1st Probability — ☐ 2nd Probability — 4 X 3

Gear shaft and Casing Data

Minimum No. of Teeth = 21 — Pressure Angle = 20 — Gears Efficiency = 0.95 — Gear Bending Stress = 1147 kg/cm2

Difference Between Each Gears Groups tooth = 0 — Teeth Gear Width Factor = 10 — Casing Wall Clearance = 1 cm

Shaft Yield Strength = 4591 kg/cm2 — Shaft Permissible Bending Stress = 1148 kg/cm2 — Shaft Safety Factor = 1

Gearbox Casing Material CI Casting — Gears Hardness Type Non-Case Hardened Gears — Read Data

Bearing, Belt and Pulley Data

Initial Bearing Width = 2 cm — Bearing Efficiency = 0.95 — Iclind Degree of Belt to First Shaft = 250

Belt Reduction Ratio = 0.48 — Pulley Efficiency = 0.95 — Driver Pulley Diameter = 9 cm

Distance Between Pulley Axis = 30 cm — Driven Pulley Width = 10 cm

Type of Load = Steady — Type of Gearbox Service = 3 Hours/Day — Read Data

Calculate — Main Menu — New — Exit

Fig -18 The Input Data for Gearbox Design, [39].

Theoritical & Actual Speeds & Speeds Errors

<<<<Back — Next >>>> — Main Menu — New — Exit

Maximum Spped Error = ± 4.0999 (%)

	Theoritical Speed	Actual Speed	Speed Error (%)		Theoritical Speed	Actual Speed	Speed Error (%)
N1	2240	2215.51	1.09	N2	1590.4	1526.57	4.01
N3	1120	1102.07	1.6	N4	795.2	795.2	0
N5	560	560.55	-0.1	N6	403.2	386.24	4.21
N7	280	278.84	0.41	N8	201.6	201.2	0.2
N9	141.12	141.83	-0.5	N10	100.8	97.72	3.06
N11	70.56	70.55	0.01	N12	50.176	50.9	-1.44

Fig -19 Theoretical, Actual Speeds and Percentage Errors between Them.

Designed Gears Data

<<<<Back — Next >>>> — Main Menu — New — Exit

Whole depth — Working depth — Chordal addendum — Circular pitch — Circular thickness — Clearance — Dedendum — Addendum — Chordal thickness — Pitch circle (d₂) — Root diameter (d₁) — Outside diameter (d₀)

Spur gear terms.

(-- Gears Calculations --)

	Teeth	Module(cm)	Do(cm)	Dp(cm)	Di(cm)	B (cm)	Reduction Ratio		Teeth	Module(cm)	Do(cm)	Dp(cm)	Di(cm)	B (cm)
Z1	43	0.3	13.5	12.9	12.15	3	i1 0.7	Z2	61	0.3	18.9	18.3	17.55	3
Z3	34	0.3	10.8	10.2	9.45	3	i2 0.5	Z4	70	0.3	21.6	21	20.25	3
Z5	27	0.3	8.7	8.1	7.35	3	i3 0.35	Z6	77	0.3	23.7	23.1	22.35	3
Z7	21	0.3	6.9	6.3	5.55	3	i4 0.25	Z8	83	0.3	25.5	24.9	24.15	3
Z9	83	0.5	42.5	41.5	40.25	5	i5 3.95	Z10	21	0.5	11.5	10.5	9.25	5
Z11	52	0.5	27	26	24.75	5	i6 1	Z12	52	0.5	27	26	24.75	5
Z13	21	0.5	11.5	10.5	9.25	5	i7 0.25	Z14	83	0.5	42.5	41.5	40.25	5

Fig -20 Gears Specifications.

Speed & Flow Chart

<<<<Back — Next >>>> — Main Menu — New — Exit

Speed Chart — I II

Flow Chart — I II

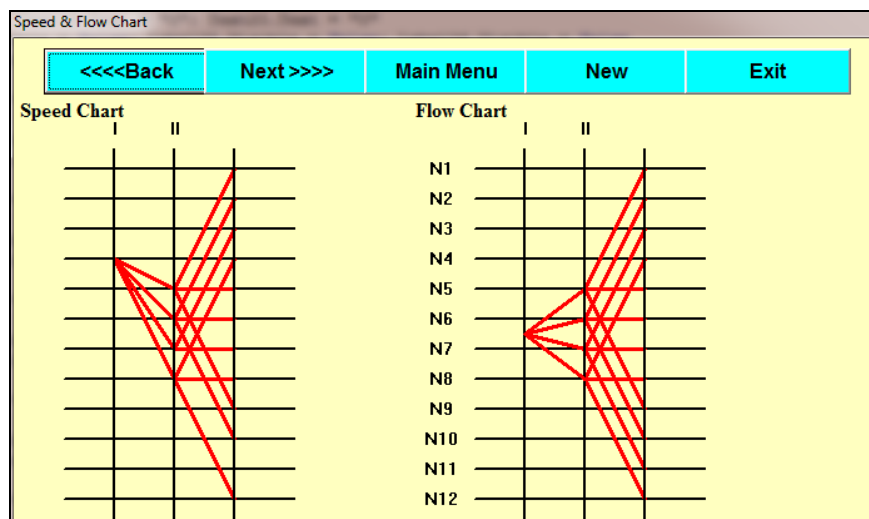N1 N2 N3 N4 N5 N6 N7 N8 N9 N10 N11 N12
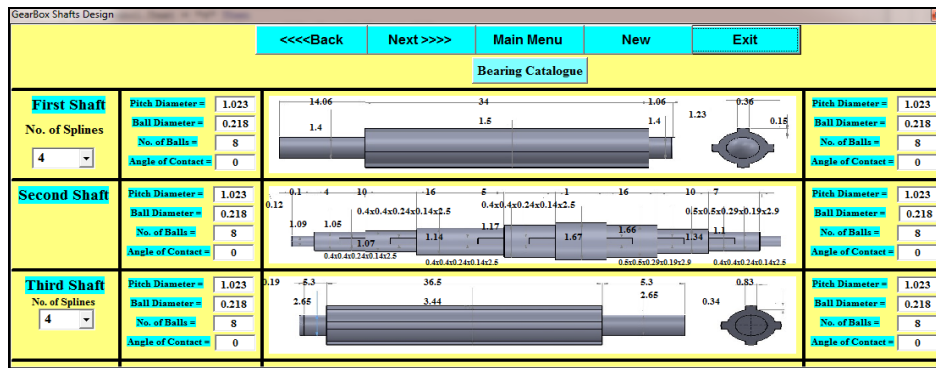
Fig -21 Speed and Flow Chart.

Fig -22 Gearbox Shafts Design.

Table (5) The Natural Frequencies (Hz) of the Highest Deflection at Gearbox with the Probabilities (6x2), (4x3) &(3x2x2).

Probability	Speed	FRF direction	Maximum Deflection (µm)	Natural Frequency (Hz)	Position
6x2	Maximum	X-direction	2.33	276	Gear 5
		Y-direction	0.532	400	Gear 7
		Z-direction	0.042	636	Gear 7
	Minimum	X-direction	1.756	688	Gear 5
		Y-direction	2.3	592	Gear 1
		Z-direction	0.088	592	Gear 1
4x3	Maximum	X-direction	8.4861e-004	188	Gear 11
		Y-direction	0.702	188	Gear 11
		Z-direction	0.229	140	Gear 11
	Minimum	X-direction	2.3024e-002	230	Gear 1
		Y-direction	1.8912	230	Gear 1
		Z-direction	3.1666	230	Gear 1
3x2x2	Maximum	X-direction	0.136	475	Gear 10
		Y-direction	13.518	475	Gear 10
		Z-direction	5.833	402	Gear 7
	Minimum	X-direction	4.3479e-003	282	Gear 11
		Y-direction	0.199	240	Gear 7
		Z-direction	5.593	210	Gear 4

J. Gear Meshing Frequencies

The frequencies of the gear meshing are defined in Hertz (Hz) or [cycles / sec.]. Gear meshing frequency is a harmonic phenomenon and it can be calculated by the following equations:

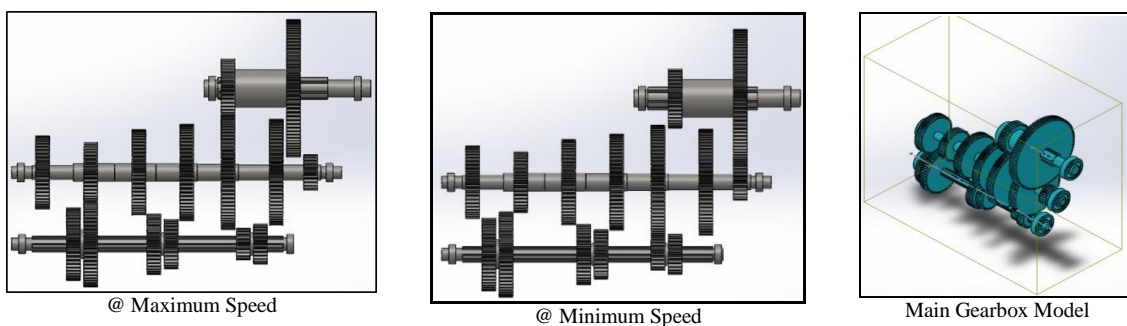$$f = z.n/60 \tag{39}$$

where; n is rotational speed (rpm).

@ Maximum Speed   @ Minimum Speed   Main Gearbox Model

Fig -23 The Kinematic Diagrams of the Minimum and Maximum Speeds Gear Meshing of the Milling Machine Gearbox of Probability 6x2.

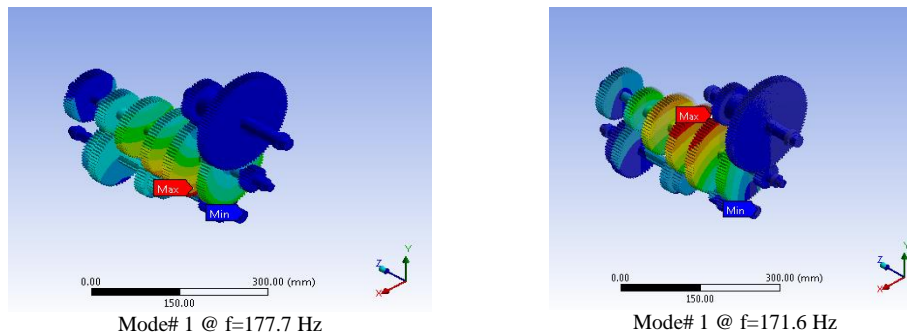Mode# 1 @ f=177.7 Hz   Mode# 1 @ f=171.6 Hz

Fig -24 The First Mode Shape of the Minimum and Maximum Gear Meshing of the Milling Machine Gearbox.

V. SUMMARY

The virtual design carried out for Mechanical Redesign of Egyptian Made Milling Machines for Modification and Updating by Using CAD. This is done by using a set of commercial packages like Ansys and Solidworks beside Gearlab and tailored codes written in VB 6 for calculating the cutting forces and required power, as well as codes to design main and feed gearboxes. This technique has several advantages as indicated below. Virtual design helps designers to prepare and conduct their design, modification, and updating without being restricted to factory locations or official work times. It helps the designers to repeat the same and/or modified one more than once, helping the designer in covering all aspects of the design. The virtual design helps managers to solve the problem of lack of instrumentation and experiments fund. It protects designers from the dangers they face during conducting some laboratory experiments on the first or modified prototype. The designer is allowed to control the inputs of the design, change the different design parameters and observe the changes in the results without being exposed to any hazards. It provides cooperation and interaction between designers and each other, and between designers and production engineers. It saves a lot of money for manufacturing institutions. Moreover, it saves time and effort of the designers, as they avoid moving between the places in their factory.

VI. CONCLUSIONS

With the increasing demand for higher-productivity machines and devices, shorter design times, and lower machine costs, machine designers need to reconsider their design approaches. An integrated set of software is successfully used and applied in machine tool design. This machine design guide examines design techniques that successful machine builders are using to improve the productivity and lower the cost and risk of their machines. The following conclusion can be achieved:

1. The virtual tools enable the design, test, optimize and control machine parts in a computer simulation environment. A virtual machine is proven to be a good tool when it comes to predict and optimize a complete machine tool's properties. An efficient optimization strategy is key when facing problems with a large number of design variables.

2. The need to design faster and more efficient machine tools has changed the way design tools are perceived. Mechanical engineers use finite element technology-based tools to study strain and deformation while the machine is operating. With this new approach, engineers can integrate different design tools to achieve a faster view of how design changes affect overall performance. They can then build prototypes not to check the performance but to modify, update and validate the overall design.

3. The reshaping of the milling machine column increases surface area for modifications #1 to #4. The increase is in surface area 1.72, 1.59, 1.53 & 2.05 times higher than the original one. This increase in the surface area could play an important role in structural heat transfer.

4. The less deformation the higher stiffness. The maximum static deformations at spindle head for modifications #1 to #4 decrease the deformation 29%, 28.5%, 28% & 31.2% lower than the original one, (145.1 μm). This means that modification # 4 improved the milling machine structure by 31.2%.

5. The reshaping of the milling machine column increases all-natural frequencies for all modified structures. The natural frequencies of all ten first modes of all modifications are higher than those of the original structure. The natural frequencies of the original structure and modifications #1 to #4 are 73.8, 111.2, 110.7, 109.3 & 112.6 Hz respectively. The improvement is by raising the natural by 50.6, 50, 48 & 52.5% up those of the original structure.

6. Generally, the fourth modification has the highest natural frequency among all modifications, the widest range of natural frequencies, and the minimum value of deformation.

7. Modification #4 is the best stiff structure. So, the fourth modification is the versatile design against harmful vibration. Thus, the milling machine can safely use wide cutting conditions as the cutting speed without dangerous resonance occurrence.

8. These results proved that the given computerized nomograms and calculators written in VB(6) are accurate, faster, reliable and expandable design tools.

REFERENCES

[1] P. H. Joshi, "Machine Tools Hand Book: Design and Operation", McGraw-Hill Education, 2013.

[2] V. Bhandari, "Design of Machine Elements", Tata McGraw-Hill Education, 2010.

[3] F. Koenigsberger, "Design Principles of Metal-Cutting Machine Tools", Elsevier, 2013.

[4] F. Koenigsberger & J. Tlusty, "Machine Tool Structures", Pergamon Press Ltd., Headington Hill Hall, Oxford, vol. 1, 1970.

[5] N. K. Mehta, "Machine Tool Design and Numerical Control 3rd ed", Tata McGraw-Hill, 2012.

[6] H. A. Youssef & H. El-Hofy, "Machining Technology: Machine Tools and Operations", CRC Press, 2008.

[7] A. A. Nasser, "Machine Tool Design-Power Transmission", Faculty of Engineering, Menoufia University, Egypt, 1979.

[8] E. Madenci & I. Guven, "The Finite Element Method and Applications in Engineering Using ANSYS", Springer, Boston, MA, 2015.

[9] X. Chen & Y. Liu, "Finite Element Modeling and Simulation with ANSYS Workbench", USA, CRC Press, Florida, pp 225-260, 2014.

[10] J. Tlusty, "Dynamics of High-Speed Milling", Journal of Engineering for Industry, vol. 1, pp 59-67, 1986.

[11] J. Muñoa, M. Zatarain & I. Bediaga, "Determination of the Cutting Performance and Dynamic Behaviour of Universal Milling Machines by Means of A Stability Model", Proceedings of ISMA, pp 3789-3804, 2006.

[12] Chen Shiguang, Zhao Honglin, Wu Zhiheng, Zhang Guangpeng, Wang Bo & Wang Qingfu, "Static and Dynamic Characteristic Analysis of XDK650X Engraving and Milling Machine", Applied Mechanics and Materials Vols. 34-35, pp 582-586, 2010.

[13] Dr. Vvrls Gangadhar, S Shashank, Gyadari Ramesh & V.P. Raju, "Random Virational Analysis on Milling Machine Structure", International Journal of Engineering Science & Advanced Technology, [IJESAT], Volume-7, Issue-1, pp 080-092, Jan-Feb 2017.

[14] Tsung-Chia Chen, Yong-Jun Chen, Ming-Hung Hung & Jui-Pin Hung, "Design Analysis of Machine Tool Structure with Artificial Granite Material", Advances in Mechanical Engineering, Vol. 8(7), pp 1-14, 2016.

[15] A. A. S. Shaaban, "Simulation Approach to Evaluate the Performance of A Milling Machine Based on the Accuracy of the Milling Products", Ph.D. thesis, Design and Production Engineering Department, Faculty of Engineering, Ain Shams University, 2015.

[16] Q. Wei, "Design and Analysis of a Small-Scale Cost-Effective CNC Milling Machine", M.Sc. thesis, Mechanical Engineering, Graduate College, University of Illinois: Urbana Champaign, 2013.

[17] S. Hassan, E. A. Rahim, Z. Mohid, & N. Warap, "Dynamic Analysis of Micro-Milling Machine", Applied Mechanics And Materials: Trans Tech Publications Ltd., vol. 465, pp. 699-703, 2014.

[18] B. M. Swami, K. S. R. Kumar & C. Ramakrishna, "Design and Structural Analysis of CNC Vertical Milling Machine Bed", International Journal of Advanced Engineering Technology, vol. 3, pp 97-100, 2012.

[19] S. Pedrammehr, H. Farrokhi, A. Khani Sheykh Rajab, S. Pakzad, M. Mahboubkhah, M. M. Ettefagh & M. H. Sadeghi, "Modal Analysis Of The Milling Machine Structure through FEM and Experimental Test", Advanced Materials Research: Trans Tech Publications Ltd, vol. 383, pp 6717-6721, 2012.

[20] Y. Altintas, C. Brecher, M. Weck & S. Witt, "Virtual Machine Tool", CIRP Annals, vol. 54, pp 115-138, 2005.

[21] A. U. Patwari, W. F. Faris, A. Nurul Amin & S. Loh, "Dynamic Modal Analysis of Vertical Machining Centre Components", Advances in Acoustics and Vibration, vol. , pp 1-10, 2009.

[22] Francesco Aggogeri 1, Alberto Borboni, Angelo Merlo, Nicola Pellegrini & Raffaele Ricatto, "Vibration Damping Analysis of Lightweight Structures in Machine Tools", 10, 297, pp 1-15, Materials 2017

[23] M. Assefa, "Modal Analysis of Machine Tool Column using Finite Element Method", International Journal of Mechanical, Industrial Science and Engineering, vol. 7, pp 51-60, 2013.

[24] Xiangman Zhou1, Qihua Tian, Yixian Du & Ruiling Zhong, "The Finite Element Analysis of Machine Tool Column of NC Gear Shaper Based on HyperWorks", Applied Mechanics and Materials Vols. 229-231, pp 491-494, 2012.

[25] Mahmood Gaznavi, Anil Kumar G., Satheesh .J & Madhusudhan T., "A Review On Dynamic Analysis of Machine Structure", International Research Journal of Engineering and Technology (IRJET), Volume: 04 Issue: 06, pp 1120-1123, June-2017.

[26] B. Niesterowicz & D. Jastrzębski, "Analytical Studies of Static Stiffness of Carrying Subsystem Saddle-Table of Machine Tool", Advances in Manufacturing Science and Technology, vol. 41, pp 17-25, 2017.

[27] Y. X. Wang, D. Li, Y. Wang, Y. L. Guo, & X. Chen, "The Dynamic Design Of Mechanical Structures of CNC Milling Machine based on Virtual Prototyping", Applied Mechanics and Materials, Trans Tech Publications Ltd, vol. 220, pp 410-413 pp 410-413, 2012.

[28] M. A. Nasser & M. A. Asy, "Dynamic, Kinematic and Static Analysis of a Shaper Machine", International Journal of Advanced Engineering and Global Technology Vol-03, Issue-06, pp 715-725, June 2015.

[29] Adesh Bokde, Abhishek Pandekar, Nehal Bante, Shubham Shukla, Prashant Kuralkar & Prof. Amit Meshram., "Modification of Shaper Machine with the Replacement of Tool Holder", International Journal for Research in Applied Science & Engineering Technology (IJRASET), Volume 6 Issue IV, pp 1538-1543, 2018.

[30] Ruchita S Gawande & S N Khan, "Modal Analysis of Gait Frequency Structure on ANSYS 15.0 based on FEA", International Journal of Engineering Research & Technology (IJERT), Vol. 5, Issue 12, pp 376-379, December 2016.

[31] M. A. Nasser, "Efficient Procedure for Modifying Structural Dynamics to Solve Structural Vibration and Acoustic Noise Problem", Proceedings of the 22nd International Modal Analysis Conference (IMAC), Dearborn, Michigan, USA, January 26-29, Paper# 389, 2004.

[32] M. A. Nasser, F. Gomaa, M. Asy, & A. Deabs, "Structural modifications of 1k62 Engine Lathe Gearbox Casing", International Journal of Advanced Engineering and Global Technology (IJAEGT), vol. 3, pp 325-334, 2015.

[33] Bajirao H. NangarePatil & Prof. P. R. Sawant, "Design and Development of Gearbox for Multi-Spindle Drilling Machine (SPM)", International Journal of Engineering Research & Technology (IJERT), Vol. 2 Issue 5, pp 1414-1423, May 2013.

[34] L. Anto & C. Kannan, "Design and Static Analysis of Gearbox for a CNC Stallion-200 Lathe", International Journal of Advanced Engineering and Global Technology (IJAEGT), vol. 4, pp 1753-1761, 2016.

[35] M. A. Asy, "Design of Machine Tool Gearboxes Driven by Double Speed Electric Motors", International Journal of Advanced Engineering and Global Technology Vol-03, Issue-03, pp 404-413, March 2015.

[36] M. A. Asy, "Adaptive Design of Machine Tool Gearboxes", International Journal of Advanced Engineering and Global Technology Vol-03, Issue-06, pp 690-698, June 2015.

[37] F. Choy, Y. Tu, M. Savage & D. Townsend, "Vibration Signature and Modal Analysis of Multi-Stage Gear Transmission", Journal of the Franklin Institute, vol. 328, pp 281-298,1991.

[38] Mr. Thite Sagar S. & Mr. Prof. Mhaske M.S, "Modal and Stress Analysis of Gear Train in Portal Axle System", International Journal of Engineering Research & Technology (IJERT), Vol. 6 Issue 05, pp 257-263, May 2017.

[39] M. A. Nasser, F. Gomaa, M. Asy, & A. Deabs, "Computer-Aided Design of Multi-Stage Gearboxes", International Journal of Advanced Engineering and Global Technology (IJAEGT), vol. 2 pp 148-157, 2014.

[40] M. Santhanakrishnan & N. Maniselvam, "Design and Fabrication of Six-Speed Constant Mesh Gear Box", International Journal of Engineering Research & Technology (IJERT), Vol. 3 Issue 9, pp *662-666*, September 2014.

[41] Rahi Jain & Pratik Goyal, "Design and Analysis of Gear-Box using Spur Gear and Eliminating the Differential Unit", International Journal Of Mechanical Engineering And Technology (Ijmet), Volume 7, Issue 6, pp 510-517, November–December 2016.

[42] JongBoon Ooi1, Xin Wang, ChingSeong Tan, Jee-Hou Ho & Ying Pio Lim, "Modal and Stress Analysis of Gear Train Design in Portal Axle Using Finite Element Modeling and Simulation", Journal of Mechanical Science and Technology 26 (2), pp 575-589, 2012.

[43] Yangyu Wang, Pengcheng Ni, Donghui Wen, Qiong Lin, DejieWang & Changqing Ma, "Dynamic Performance Optimization of Circular Sawing Machine Gearbox", Applied Sciences, 9, 4458, pp 1-20, 2019, 2019.

[44] D Mohammed Rafi, Mr. B. Raja Kumar & G.S.R.Nagamallesw A Rao, "Design and Development of Gearbox for Multi-Purpose Milling Machine (Spm) ", Anveshana's International Journal of Research in Engineering and Applied Sciences, Aijreas Volume 1, Issue 12, pp 90-100, Dec 2016.

[45] Bearing Frequencies, [Online], Available:
http://www.ntnamericas.com/en/website/documents/brochures-and-literature/tech-sheets-and-supplements/frequencies.pd