Acta Technica 62 (2017), No. 5A, 53–62

An integrative methodology to efficiently design the optimal structure of column-type machine tools

Hong-Yi Chen[2], Hui-Cun Shen[3,5], Kun-Chieh Wang[3], Chi-Ching Yang[4]

Abstract. For tool machinery, the rigidity of structures plays the most important role for their final machining precision. In the marketplace, there exist lots of structure types which change very quickly to meet the ever-increasing needs of customers. Facing the challenge, all machine makers should have a good methodology to optimally design and verify their machine products. However, designing a good machine structure has never been a simple and unilateral thing. Therefore, this study, selecting a double-column machining center as the target because of its highly unstable structure, uses a hybrid design procedure which combines the experience as initial design bases and together with an integrative numerical examination of both static and dynamic rigidity to theoretically obtain a high-rigid structure with low cost. With this proposed methodology, machine designers may efficiently and quickly determine the optimal structure of a column-type machine tool.

Key words. Finite element method, structure design, machine tools design, high-rigidity design.

1. Introduction

Nowadays, machine makers want to shorten the developing time to cope with the constant change of customers' needs. But, in the viewpoints of a machine producer, building a reliable machine in the way of time-saving and cost-saving seems a conflict event to each other. To solve this conflict, some following basic concepts

[1]Acknowledgement - This work was financially supported by the Project for cooperation between production and education of Fujian province of China (No. 2017H6018), Project of Ningde Normal University (No.2015Z03 and 2015T07), and Projct of Ningde Municipal Bureau of Ocean and Fishery.

[2]Workshop 1 - Overseas Chinese University, Taiwan

[3]Workshop 2 - Nindge Normal University, Fujian, China

[4]Workshop 3 - Hsiuping University of Science and Technology, Taiwan

[5]Corresponding author: Hui-Cun Shen; e-mail: `shenhuicun@126.com`

should be aware first. To design a good structure of tool machinery, there are lots of factors which influence their precision behavior during machining. Among these factors, the original type of structure is especially crucial. Examining the development procedures of tool machinery, it is found that most of the machine structures were designed by experiences. In the early age, it was O.K. for using the experience-based potential rules to design simple machine structure with low precision. However, in the age of high-competition environment and high-precision requirement today, this experience-based design can no longer provide enough knowledge for developing a good-quality machine which frequently requires features like low-weight, multi-function, high rigidity and at the same time possesses complex structure. Therefore, for saving development time and money, as well as introducing more solid mechanics knowledge to cope with the ever-growing difficulties for designing a good machine, a rapid and efficient way in conjunction with the accumulated know-how or past knowledge are needed.

So far, there is a popular efficient way to develop or analyze the machine structure, called "Finite Element Method (FEM)," which is a numerical method. Using FEM as a developing tool started in 1990s. In the past, lots of researches were successfully made in many different areas [1-5]. Meanwhile, many studies were made about the subject of analyzing the stiffness of machine tools using experimental, analytical or numerical methods [6-10]. However, reports which include detailed technical know-how in analyzing the rigidity of machine tools by the numerical method of FEM were never seen. Therefore, this report attempts to use the integrative knowledge-based FEM technique to better design the structure of a machine tool.

For designing the strong structure of a machine tool, the most important factors are the static and dynamic rigidity, and the modal shapes of natural vibration. The static rigidity concerns the magnitude of deformation when applying a static loading on the machine. The dynamic rigidity concerns the magnitude of deformation under the periodic external stimuli. And the natural frequencies and their corresponding modal shapes give the knowledge about avoiding machine resonance. In the machine structure design, these three parameters interact with each other. Sometimes a structure with good static rigidity would still easily appear damage due to resonance or failure during normal cutting operations. Thus, an integrative examination of these parameters should be made to provide an overall well design guide of the machine structure.

As such, this report aims to use the integrative know-how based FEM technique to design an optimal machine structure by investigating the static rigidity, dynamic rigidity and natural frequencies as well as their corresponding modal shape, which may provide an overall detailed knowledge of machine structure.

2. Manupulation Procedure

The manipulation procedure of our proposed integrative know-how-based FEM technique includes eight steps, as shown in Fig. 1. First, the target of a column-type machine tool is chosen. All possible related restricted conditions are indicated, which includes various boundary conditions, material, physical properties of machine structure, and so on. Second, introducing the past know-how about the structure of the similar kind as the basic structure reference (this structure must be tested, confirmed its excellent quality by the market or users). Third, building a structure prototype based on the referenced structure. Fourth, performing the mathematic modelling in finite element form. Fifth, constructing the mesh of the prototype. Sixth, performing the FEM calculation using SOLIDWORKS program. Seventh, analyzing and evaluating the static rigidity, dynamic rigidity, and vibration mode of every case. Eighth, giving a suggestion to find out an optimal structure.

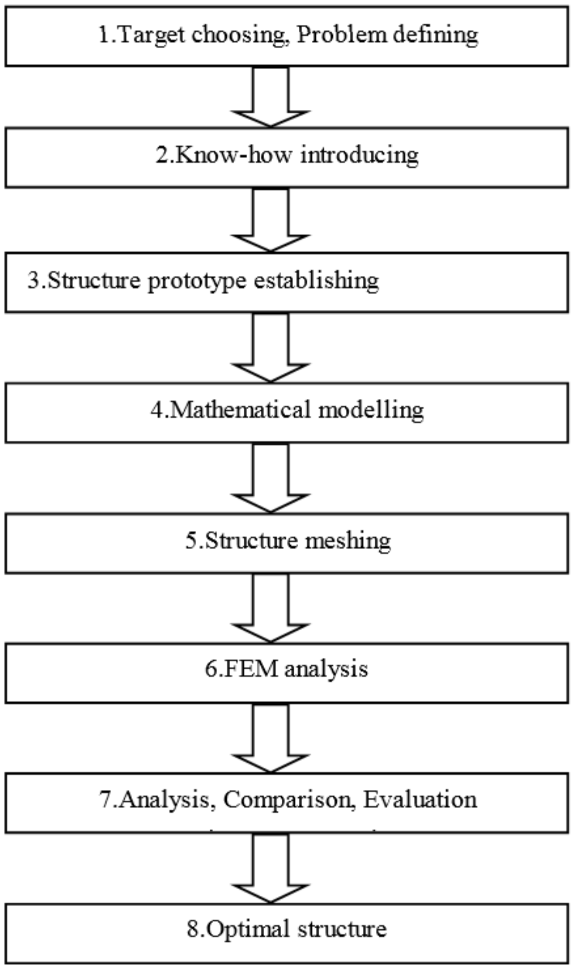

Fig. 1. Manipulation procedure

3. Theories

3.1. Static Rigidity and FEM

For tool machinery, the governing equation of the structure displacement can be expressed as

$$\tau = \tau_0 \left(1/2 - \xi\right) , \tag{1}$$

or

$$E = E_0 \left(1 - \gamma\tau\right) , \tag{2}$$

where kg_f is the system stiffness matrix, kg_f ; kg_f is the displacement vector; kg_f is the element number; kg_f is the element stiffness matrix; kg_f is the total external force vector; kg_f is the reaction load vector. In Eqn. (2), if sufficient boundary conditions are provided, then the displacement at every node (kg_f) may be obtained. If so, Eqn. (2) may be rewritten as

$$E = E_0 \left(1 - \alpha \left(1/2 - \xi\right)\right) , \tag{3}$$

Where, the subscript "i" means the degree of freedom without displacement restriction. The kg_f is known but not necessarily equals to kg_f . Since the reaction force must be zero when there is no displacement restriction, Eqn. (3) may be written as

$$h(\xi) = h_0 \left[1 - \left(1 - \beta_1\right)\left(\xi + 1/2\right)\right] \cdot \left[1 - \left(1 - \beta_2\right)\left(\eta + 1/2\right)\right] , \tag{4}$$

Solving the upper part of Eqn. (4), we may obtain

$$\rho = \rho_0 \left[1 - \left(1 - \beta\right)\left(\xi + 1/2\right)^2\right] , \tag{5}$$

The reaction forces kg_f may be obtained via the upper equation. Solving the lower part of Eqn. (4), we may get

$$T = \frac{ab}{2}\omega^2 \int_A h(\xi)\rho w^2 \,\mathrm{d}A \tag{6}$$

The strain energy kg_f is further obtained as

$$V = \frac{ab}{2} \int_A D(\xi)[G - 2(1-\nu)H] \,\mathrm{d}A , \tag{7}$$

The relationship between nodal displacement vector kg_f and displacement field is described as

$$D(\xi) = D_0 \left[\left[1 - \left(1 - \beta_1\right)\left(\xi + 1/2\right)\right] \cdot \left[1 - \left(1 - \beta_2\right)\left(\eta + 1/2\right)\right]\right]^3 , \tag{8}$$

where kg_f is the shape function matrix. The strain- displacement relationship is

$$D_0 = \frac{Eh_0^3}{12\,(1-\nu^2)} \tag{9}$$

where kg_f means the linear differential operator. For a linear structure, the stress vs. strain relationship is

$$D(\xi) = \frac{E_0 h_0^3}{12\,(1-\nu^2)} (p_1 p_2)^3 p_3\,. \tag{10}$$

where kg_f is the elasticity coefficient matrix.

3.2. Modal analysis and FEM

Modal analysis of a machine structure usually presents its results in the form of natural frequencies and mode shapes. Three basic assumptions must be made in the modal analysis: (1) The structure is linear, (2) No any damping effect, (3) All physical properties under consideration is independent of time, such as force, displacement, or temperature. In other words, the structure is under free vibration. Based on these assumptions, the governing equation of a structure under free vibration may be expressed as

$$T = \frac{ab}{2}\rho_0 h_0 \omega^2 \int_A p_1 p_2 \left[1-(1-\beta)\left(\xi+\frac{1}{2}\right)^2\right] w^2 \,\mathrm{d}A\,, \tag{11}$$

For a linear structure system, the motion of free vibration is harmonic, i.e.,

$$V = \frac{ab}{2}\frac{E_0 h_0^3}{12\,(1-\nu^2)} \int_A (p_1 p_2)^3 p_3 (G - 2(1-\nu)H)\,\mathrm{d}A \tag{12}$$

here kg_f means the amplitude or mode shape for the ith frequency kg_f . Substituting Eqn. (12) into Eqn. (11), we have

$$\delta\,(V-T) = 0\,. \tag{13}$$

The above equation is an "eigenvalue problem" in which non-trivial solutions occur under the following condition

$$w = A_1 q_1^2 q_2 q_3 + A_2 q_1^3 q_2^2 q_3^2\,, \tag{14}$$

From Eqn. (14), we may obtain n numbers of eigenvalues kg_f , i=1,2,.., n and their corresponding eigenvectors kg_f , i=1,2,.., n. The variable n is the number of degree of freedom of the structure system. For further application, the obtained eigenvectors are usually normalized based on the mass matrix as

$$\delta\left(V_1 - \lambda^2 T_1\right) = 0\,, \tag{15}$$

3.3. Forced Vibration and FEM

The forced vibration response of machine structure follows the equations from the linear dynamic structural analysis with considering external forces (may include inertial force, damping force, and impact force) as:

$$\eta = \frac{c}{4b} - \frac{\xi}{2} + \frac{1}{4} + \frac{c\xi}{2b}, \; \eta = -\frac{c}{4b} + \frac{\xi}{2} - \frac{1}{4} - \frac{c\xi}{2b}, \; \xi = -\frac{1}{2}, \; \xi = \frac{1}{2}. \qquad (16)$$

where M is the material mass matrix, C is the damping matrix, K is the material stiffness matrix, τ is the acceleration vector, kg_f is the velocity vector, and u is the displacement vector, and $f(t)$ is the time-dependent external load vector.

4. Results

Based on the good experience-based old column-type machine structure, we modify its related size and configuration. The obtained reference-based new column-type structure is shown in Fig. 2. The magnitude of external applying forces is setting as 100 kg_f in the x, y, and z directions. Applying the proper boundary and initial conditions and choosing cast iron as material, we may perform the meshing work (results shown in Fig. 2b) and FEM calculation. Throughthe element-independent tests, the proper total nodal points of 421507 and total element numbers of 225137 are obtained. Further, three different cases are under consideration: the spindle-head positions are located at the top, middle (550 mm from the top) and bottom (and 1100 mm from the top) of the vertical ram.

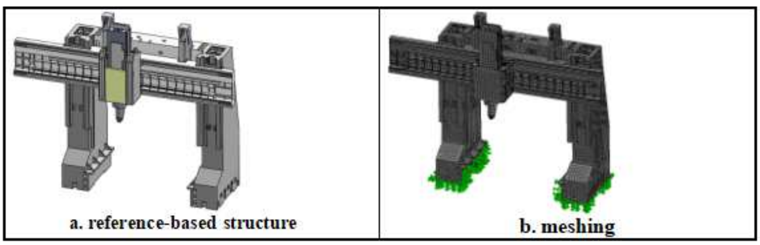

Fig. 2. Referencec-based structure and its meshing

4.1. Static rigidity

For case A, the calculated results of displacement distributions under the action of external forces in x, y, and z directions are shown in Fig. 3. The obtained maximum displacements are 0.0104 mm, 0.0133 mm, and 0.0075 mm in the x, y, and z direction, respectively.

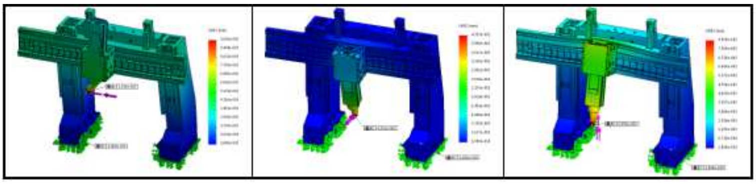

Fig. 3. Displacement distributions under the action of forces in the x, y, and, z directions for case A

4.2. Natural frequencies and mode shape

The calculated natural frequencies of case A (with in the rage of 0˜500 Hz) are shown in Table 1 and the corresponding first eight modal shapes are shown in Fig. 4. The lowest natural frequency occurs at 21.2 Hz. Under the normal cutting conditions which the rotational speeds of spindle are frequently set within the range of 0-5000 rpm, the first 10 natural frequencies and their corresponding modal shape should be noticed. The structure resonance always happens around these critical frequencies.

Table 1. Natural frequencies of case A

β_1	$\beta = 0.4,\ \beta_2 = 0.0$				$\beta = 0.4,\ \beta_2 = 0.6$			
	$\alpha = 0.0$		$\alpha = 0.4$		$\alpha = 0.0$		$\alpha = 0.4$	
	first mode	second mode	first mode	second mode	first mode	second mode	first mode	second mode
1.0	24.6196	166.561	23.0109	152.343	33.1729	189.644	30.9359	173.336

4.3. Dynamic rigidity

The harmonic wave analysis was made to calculate the responses under the stimuli of periodic applying forces. The loading force with a magnitude of 100 kg_f is set acting on the lowest end of spindle nose, which has a frequency range of 0-440 Hz (normal operation range). Table 2 and Fig. 5 show the dynamic rigidity distribution results of the stimuli of periodic force acting in the x-direction. The dynamic weak points are found occurring at 21.2 Hz, 21.4 Hz, 268.7 Hz, and 297.3 Hz with the magnitudes of 0.291mm, 0.349 mm, and 0.095 mm, respectively. Larger vibration would occur around the above frequencies, the user should be aware.

Table 2. Results of dynamic rigidity

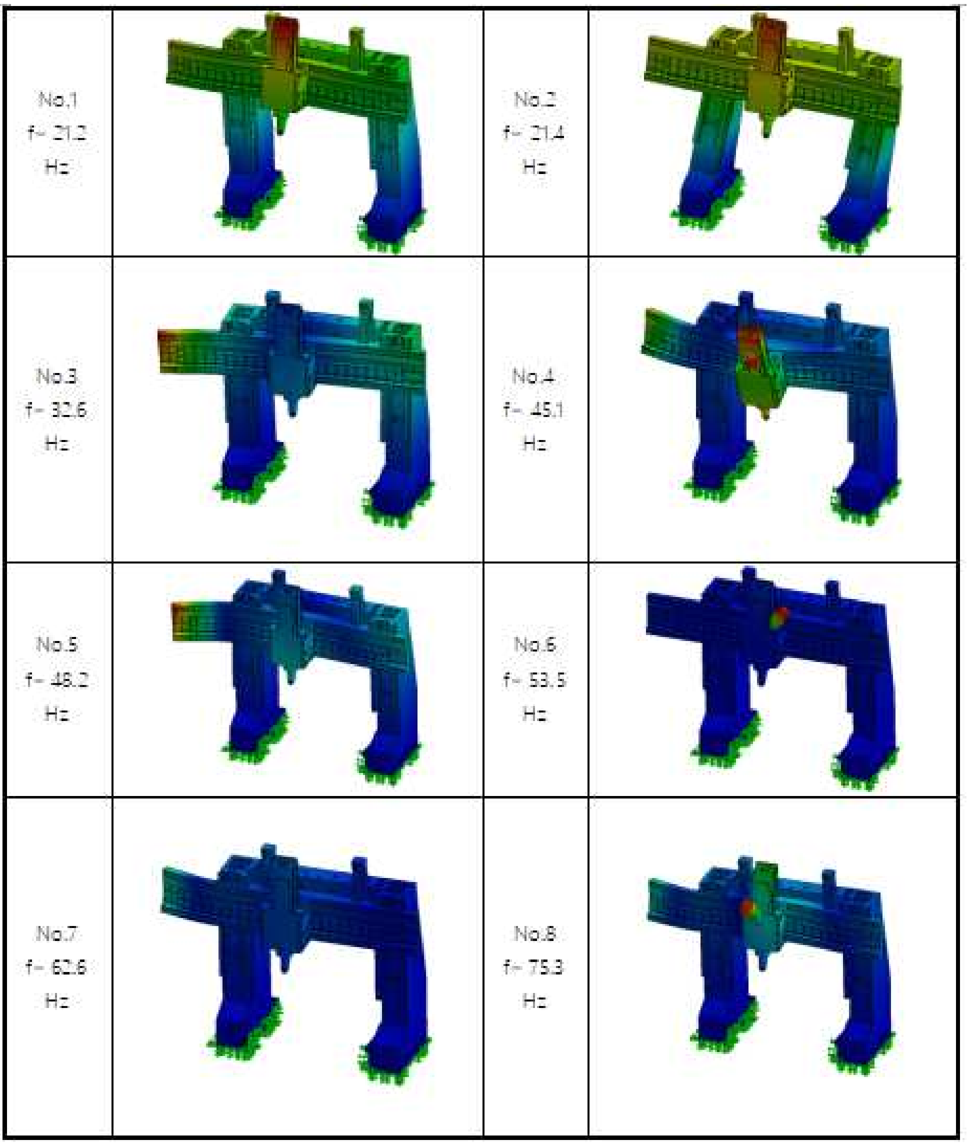

Fig. 4. Modal shape (first eight frequencies)

β_1	$\beta = 0.4,\quad \beta_2 = 0.0$				$\beta = 0.4,\quad \beta_2 = 0.6$			
	$\alpha = 0.0$		$\alpha = 0.4$		$\alpha = 0.0$		$\alpha = 0.4$	
	first mode	second mode	first mode	second mode	first mode	second mode	first mode	second mode
0.0	20.5249	114.338	19.7942	107.769	27.5140	131.284	26.5642	124.196
0.2	20.8680	121.896	20.0278	114.019	27.9417	139.219	26.8341	130.581
0.4	21.4753	131.325	20.4902	121.995	28.7586	149.560	27.4395	139.181
0.6	22.3218	142.132	21.1574	131.242	29.9311	161.693	28.3482	149.431
0.8	23.3796	153.962	22.0055	141.436	31.4175	175.165	29.5258	160.917
1.0	24.6196	166.561	23.0109	152.343	33.1729	189.644	30.9359	173.336

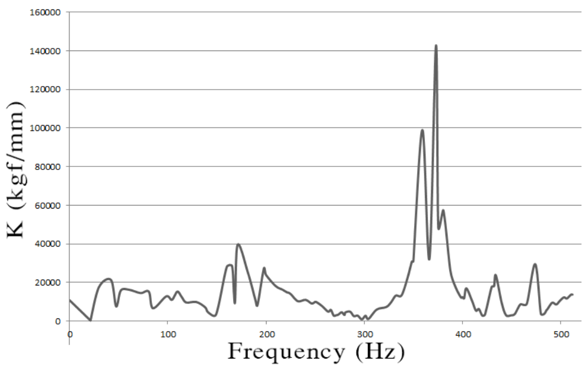

Fig. 5. Dynamic rigidity distribution

4.4. Comparison and optimal structure

Changing the positions of spindle nose from the top (case A), the middle (case B), to the bottom (case C) results in different structure-rigidity behaviors. The obtained correspondent maximum stresses appeared in the spindle nose are 713916 N/m^2 , 705594 N/m^2 , and 880096 N/m^2 , respectively. It is apparently seen that case C would induce a larger stress inside the machine structure. Consequently, it is suggested that cases of spindle-nose positions located at the lowest, the most right, and the most front positions should be considered. On the other hand, the obtained maximum displacements of the spindle nose for case A, B, and C are 0.013 mm, 0.024 mm and 0.044 mm, respectively. Further, the static rigidity of the machine structure for case A, B, and C can be calculated as 7.512 kg_f/mm , 4.128 kg_f/mm , and 2.297 kg_f/mm . Through the above finding, it is suggested that an optimal column-type machine structure should have strong inner structure (e.g. rib type, thickness, support), avoid operating at the first ten critical frequencies, and notice the user not operating the external stimuli at the specific frequencies where weak points occur.

5. Conclusion

This study used the know-how based FEM technique to explore the structure of a double-column machine tool. Three critical parameters of the machine structure: static rigidity, dynamic rigidity and natural frequencies as well as its modal shapes were investigated simultaneously to obtain an optimal structure in which it ensures a high precision behavior during machining. Three cases with the spindle locating at the top, middle, and bottom positions are examined. Results show that the know-how based prototype can give an efficient design guide to a new machine prototype. (2) The FEM technique, in the least price, may be used to get a detailed insight of machine structure and find out what on earth a good structure should be. (4) Combining these two techniques result in a quick efficient way to explore a good machine structure with high rigidity.

References

[1] T. R. Thomas, R. S. Sayles: *Stiffness of machine tool joints: A random-process approach.* ASME Journal of Engineering for Industry *88* (1977) 250–256.

[2] S. Hinduja, B. C. Nakra: *Study of the torsional stiffness of machine tool columns with different forms of ribbing.* Thin Solid Films *1* (1977) 10–22.

[3] T. Inamura, T. Sata: *Stiffness and damping identification of the elements of a machine tool structure.* CIRP Annuals *28* (1979), No. 1, 235–239.

[4] S. Shihab, H. Riahi, L. Thevenard: *A Lemaître Systematic study of the spin stiffness dependence on phosphorus alloying in the ferromagnetic semiconductor.* As Applied Physics Letters *106* (2015), No. 14, 187.

[5] D. T. Y. Huang, J. J. Lee: *On obtaining machine tool stiffness by CAE techniques.* International Journal of Machine Tools & Manufacture *41* (2001) 1149–1163.

[6] D. Kano, T. Lorenzer, S. Weikert, K. Wegener: *Evaluation of modeling approaches for machine tool design.* Precision Engineering *34* (2010) 399–407.

[7] J. J. Wu: *Finite element analysis and vibration testing of a hree-dimensional crane structure.* Measurement *39* (2006) 740–749.

[8] Y. Kang, Y. P. Chang, J. W. Tsai: *Integrated CAE strategies for the design of machine tool spindle-bearing systems.* Finite Elements in Analysis and Design *37* (2001) 485–511.

[9] J. J. Wu: *Finite element modeling and experimental modal testing of a three-dimensional framework.* International Journal of Mechanical Sciences *46* (2004) 1245–1266.

[10] R. Mahdavinejad: *Finite element analysis of machine and workpiece instability in turning.* International Journal of Machine Tools & Manufacture *45*, (2005) 753–760.

[11] Y. Kumar: *Free vibration analysis of isotropic rectangular plates on Winkler foundation using differential transform method.* IJ Applied Mechanics and Engineering *18* (2013), No. 2, 589–597.

Received November 16, 2017