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Design Methods that Work

DRAFT DESIGN SUPPORTED BY EVALUATION OF MACHINE TOOL DYNAMIC STIFFNESS

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Abstract: *Modern machine tools are usually built from modules connected by standard ball and/or roller slides. The proper kinematics structure of the machine tool must be chosen for given technological tasks. During draft design of a machine tool, it is important to be able to evaluate dynamic properties of different structural variants before deciding about final configuration. The paper proposes a procedure for such evaluation. The procedure is based on a linear discrete model transformed from 3D CAD design. The comparative researches have been made on a small table evaluated by above method and then tested experimentaly.*

1. INTRODUCTION

In the recent years the market of machine tools became the global one. The rapid changes on that market along with the growing up competition force continuous changes of requirements defined for manufactures of machine tools. The process of adapting machine tools to alternating technical requirements as well as the concurrent demands for production's low costs, flexibility, high precision and effectiveness inspired researches for optimal structural solutions. The foregoing tendencies have a huge impact on design process.

The notions designing and constructing, often associate with the drawing board, drafting and doing simple calculations. The well know, prof. Janusz Dietrych, an indisputable authority in the systematizing notions and definitions in constructing science domain, defines designing as selecting the way of subsystem working, and creating, in special circumstances, the whole system as a formal basis for working any subsystem [1]. It can be also stated, that the design process starts in when the particular need appears and the decision for satisfying that particular need is made. Having this in mind, the design process is done, when there is available detailed and reliable information on how and which techniques and tools must be used, to satisfy the particular need [2].

In the literature, one can find many design process definitions. Those definitions are diverse, as their authors concentrate on different aspect of design process. In machine tool industry the design and constructing process is understood as a chain of tasks that results in more and more detailed design of machine tool. Each of the tasks lead to a set of potential solutions. And for the next designing stage the best one must be selected [3].

The decision on choice of the solution from the set of the potential ones is always made by designers. And, the decision is made subjectively and arbitrary. The choice is a result of gathering and processing information, and its goal is to select one solution that meets all requirements and is the best one with regard to the given quality requirements.

The optimal solution choice is done either by heuristic method or analytical method or both of them. The heuristic method is an individual method, that requires a reliable knowledge and wide experience. The outcome of the heuristic method depends heavily on the designer talents and his/her present psychical and physical condition. The analytical method is based on mathematical model of the entire system.

The model must include all constraints for the given system and the quality requirements that must be met. In practice, the heuristic method is applied at the stage of the draft design (Fig. 1). At that stage the general solution for further development is chosen. Therefore, the proper choice of the solution at the drafty design stage is crucial. Analytical method is usually applied when the full technical documentation of the future prototype is available.

New machine tools, are frequently designed basing on already existing solutions. However, when the machine tool is to be build as a completely new one then the design process has the from presented on Fig. 1.

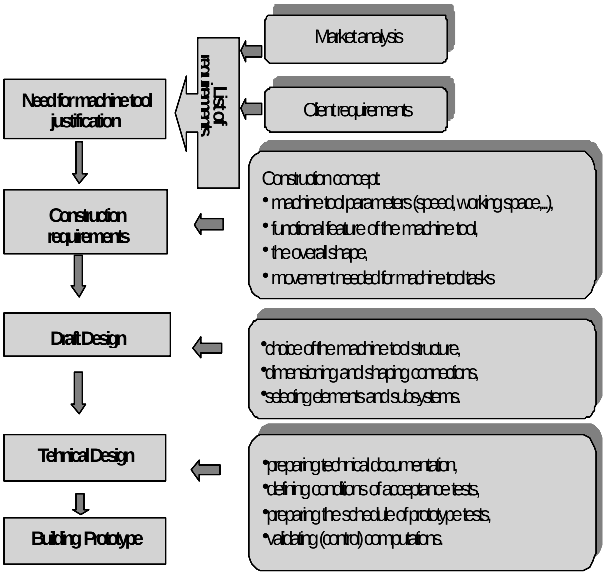


Fig.1 *Scheme of the design process*

2. DRAFT DESIGN STAGE

The draft design, as it was already stated, is a crucial stage in the design process, as it determines the geometric and movement parameters of the machine tool. At this stage - after a thorough analysis – the main features and optimal solution of the whole machine and most of its subsystems are determined. Moreover, the draft design, in its final shape, brings the information on all selected elements and subsystems, the overall dimensions and types of connections between the parts of the designed machine tool. So, after the draft design stage the structure of the machine tool as well as its dimensions, ball guideways connections are defined. If the choice of structure version was made wrong, the results of it will appear on the next stage – technical design – or even later, when the prototype is built and tested.

In the recent years rapid evolution of computer science took place. As result of it design engineers may use computer tools that significantly supports the design process. The most popular are: CAD systems, finite elements method systems (FEM) and simulation systems for examining kinematics and dynamics of multi-mass systems. The foregoing computer tools can be used for testing the construction on the stage of technical design (Fig. 1) – even before building the prototype. Now, it is possible to perform precise static, dynamic and thermal analysis of the entire construction. However, besides many advantages, the computer tools, have some drawbacks. Most of the professional computer tools are complicated and expansive systems. Therefore, the implementation of them into a small and medium machine tool enterprises is an expensive and time consuming process. Hence, those large system, such as UNIGRAPHICS offered by Unigraphics Solutions, are mainly used by the special design divisions in the automation, airspace or armaments industry.

There is a need for computer tools that can be successfully applied at the draft design stage, when the decision on the configuration of the future machine tool is made.

3. EVALUATION ON THE DRAFT DESIGN STAGE

Manufactures of machine tools use a lot of elements and certain sub-modules (such as ball guideways). Ther producer offer off-the shelf modules that can be connected with each other and machine tool body in order to get the whole machine. Also, the development of open control systems (NC, PLC, CNC) [4] gives manufacturers the large flexibility in design new machine tools. Hence, the design and construction process can be substantially fastened [5].

As everyone knows, the dynamic stiffness of the machine tool is one of the most important parameters of the machine tool. The dynamic stiffness may be computed by the finite elements method. In the design and construction process it take place after the stage of technical design, when the full technical documentation is prepared – so, it is the next stage after the configuration of the machine tool has been chosen (Fig. 2).

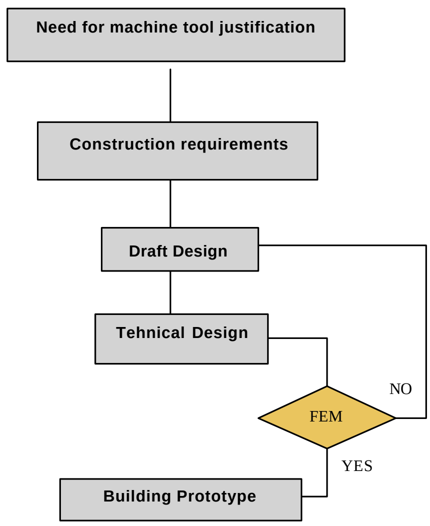


Fig.2 *The FEM computations in design process*

In order to minimize the impact of incorrect choice of the geometric and kinematic version of the entire machine tool it would be helpful to evaluate the stiffness of the construction at the stage of draft design.

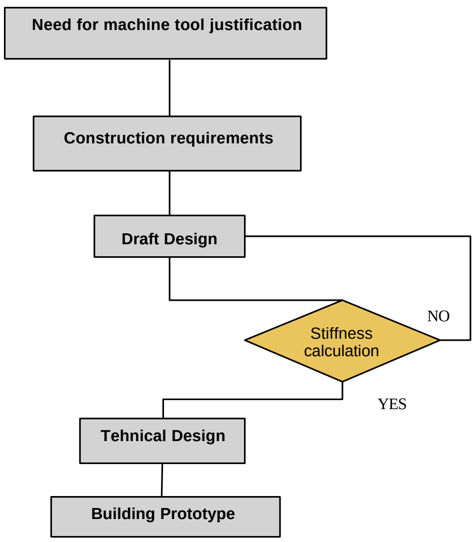


Fig.3 *The draft computations of stiffness in the design process.*

Now, it is even simpler, because in modern machine tools, ball guideways are applied. The, ball guideways can be modeled as elastic element with known coefficient of elasticity, given by their manufacturers.

The idea of evaluating machine tools version (variants) on the draft design stage consists in the running calculations of static and dynamic stiffness of the entire structure. Those calculations can be done basing on parameters of load that come from technical requirements, geometrical structure of the machine tool and overall dimensions given in CAD-solid model. The evaluation of the geometric and kinematic features of the construction can be done by treating the machine tool as a set of stiff solids connected by ball guideways. The ball guideways are off-the shelf elements, which operating parameters are given by their manufactures. It is also assumed, that the entire version of the machine tool is treated as a linear object.

In order to check the foregoing assumptions for the modeling process of machine tool experimental researches and measurements of a linear module, as a example of basic machine tool subsystem, were performed. Then, the mathematical model of the tested module was prepared.

4. MEASUREMENTS AND MODELLING THE MACHINE TOOL SUBSYSTEM TKK 15-155

In our experiments we have used a typical ball rail tables from Rexroth. It consists of aluminum table, two ball guideways, rolling screw and drive. The table is supported by four roller bearings that enables movements of the table on rails (Fig. 4).

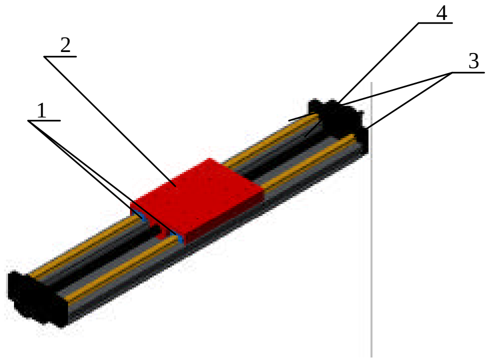
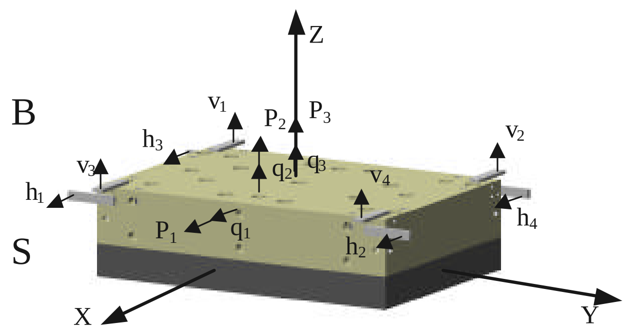


Fig.4 *The measured object: 1-roller bearings, 2-aluminum table, 3-ball guideways, 4-rolling screw*

In the experiments, an additional aluminum block with holes for screwing in sensors was fasten to the table. The goal of applying the above block was to decrease the spectrum of free vibration frequency. There were assumed three points for input function (P1, P2, P3), one force frequency 700Hz (non-resonance one), two sets of displacement and acceleration sensors (h1, h2, v1, v2) and (h3, h4, v3, v5) – see Fig. 5.

a)



b)

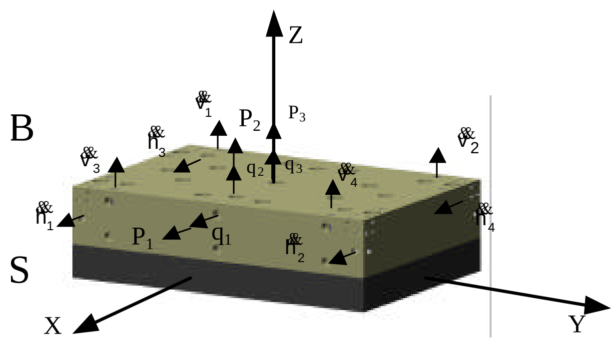


Fig. 5 *Sensors lay-out a) movement sensors b) acceleration sensors*

Movements and acceleration sensors were mounted (assembled) in four corners of the aluminum block, creating a configuration of movements and acceleration two horizontal levels h1, h2 and two vertical v1, v2. After changing the assembly of sensors, we have obtain a configuration of movements and acceleration two horizontal levels h3, h4 and two vertical v3, v4. Making a transformation the movements in three points and directions of input function can be determined. The computation algorithm has been changed in a such a way that the stiffness and dumping of roller bearings can determined using the measurement with one only inducing frequency. Moreover, the measured pair of movements verify each other as well as the bisymmetry assumption.

Before preparing the mathematical and physical model of the examined system we have made following assumptions:

- the corps of the table is nondeformable,
- trolleys are deformable in horizontal transverse direction and in vertical direction,
- each trolley is modeled by two damping-elastic elements: the horizontal one and vertical one,
- there is a bisymmetry of the system along two planes.

The symmetry of the system and load along horizontal transverse plane allows adapting physical model of the entire system in the form of flat disk, that represent table and the additional block. The disk is supported by two pair of damping-elastic constraints. Each pair of constraints models two trolleys, where horizontal elements represent transverse stiffness and transverse adhesive dumping in vertical direction; while the vertical ones represent horizontal stiffness and horizontal adhesive dumping of trolleys in horizontal direction. Hence, the system is modeled as a discrete system with three degrees of freedom in movements base u1, u2, u3 in a central point of supporting plane. It is the indirect coordinate system in our problem (Fig. 6).

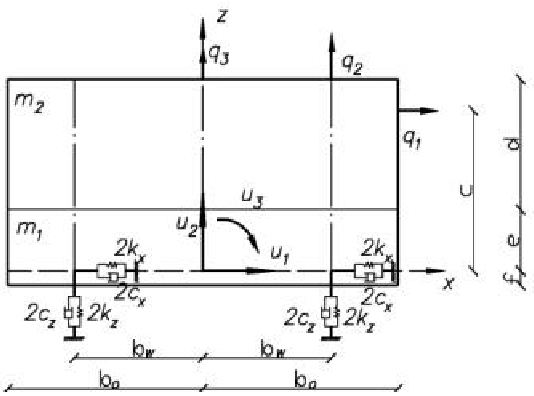


Fig.6 *The stiff disk with dumping-elastic support*

The equation of motion was formulated using the classical matrix algorithm, where the balance of kinetic energy (1), elastic energy (2), dumping power (3) and the work of active load was made as well as the Lagrange's equation type II is used [7].

$$E_k = \frac{1}{2}\dot{\underline{u}}^T \underline{B}_u \dot{\underline{u}}, \quad \dot{\underline{u}} = \underline{A}\dot{\underline{q}}, \quad E_k = \frac{1}{2}\dot{\underline{q}}^T \underline{B}\dot{\underline{q}} \qquad (1)$$

$$\underline{B} = \underline{A}^T \underline{B}_u \underline{A}$$

where

$$\begin{bmatrix} u_1 \\ u_2 \\ u_3 \end{bmatrix} = \begin{bmatrix} 1 & \frac{c}{b_w} & -\frac{c}{b_w} \\ 0 & 0 & 1 \\ 0 & -\frac{1}{b_w} & \frac{1}{b_w} \end{bmatrix} \cdot \begin{bmatrix} q_1 \\ q_2 \\ q_3 \end{bmatrix}$$

and (results comming from the design)

$$\underline{B}_u = \begin{bmatrix} b_{11}^u & 0 & b_{13}^u \\ & b_{22}^u & 0 \\ \text{sym.} & & b_{33}^u \end{bmatrix},$$

$$b_{11}^u = b_{22}^u = \sum m_i = 7{,}60\,\text{kg}$$

$$b_{13}^u = \sum m_i h_i = 0{,}2509\text{kg}$$

$$b_{33}^u = \sum \left[\frac{1}{12} m_1 \left(a_i^2 + b_i^2\right) + m_i h_i^2 \right] = 0{,}02713\text{kgm}^2$$

$$E_p = \frac{1}{2}\underline{u}^T \underline{K}_u \underline{u}, \quad \underline{u} = \underline{A}\underline{q}, \quad E_p = \frac{1}{2}\underline{q}^T \underline{K}\underline{q}$$

$$\underline{K} = \underline{A}^T \underline{K}_u \underline{A}, \qquad \underline{K}_u = \begin{bmatrix} 4k_x & 0 & 0 \\ & 4k_z & 0 \\ \text{sym.} & & 4k_z b_w^2 \end{bmatrix} \qquad (2)$$

$$\underline{K} = \begin{bmatrix} 4k_x & 4k_x c/b_w & -4k_x c/b_w \\ & 4k_x c^2/b_w^2 + 4k_z & -\left(4k_x c^2/b_w^2 + 4k_z\right) \\ \text{sym.} & & 4k_x c^2/b_w^2 + 8k_z \end{bmatrix}$$

$$\Phi = \frac{1}{2}\dot{\underline{u}}^T \underline{C}_u \dot{\underline{u}}, \quad \dot{\underline{u}} = \underline{A}\dot{\underline{q}}, \quad \Phi = \frac{1}{2}\dot{\underline{q}}^T \underline{C}\dot{\underline{q}}$$

$$\underline{C} = \underline{A}^T \underline{C}_u \underline{A}, \qquad \underline{C}_u = \begin{bmatrix} 4c_x & 0 & 0 \\ & 4c_z & 0 \\ \text{sym.} & & 4c_z b_w^2 \end{bmatrix} \qquad (3)$$

$$\underline{C} = \begin{bmatrix} 4c_x & 4c_x c/b_w & -4c_x c/b_w \\ & 4c_x c^2/b_w^2 + 4c_z & -\left(4c_x c^2/b_w^2 + 4c_z\right) \\ \text{sym.} & & 4c_x c^2/b_w^2 + 8c_z \end{bmatrix}$$

As we can see above, the structure of the damping matrix (3) is identical to the structure of the stiffness matrix – this is caused by the same configuration of damping and elastic constraints.

The matrix equation of the identified system has the well know form (4)

$$B\ddot{q} + C\dot{q} + Kq = F_0 \qquad (4)$$

where the values of elements of the matrix of inertia are know. However, only the structure of the stiffness and damping matrices is known. The elements of those matrices are expressed in terms of stiffness coefficients k_x, k_z and damping coefficients c_x, c_z. We have used results from the measurement of vibrations in directions corresponding to directions of harmonic force inducing vibration, to find the numerical values of elements of stiffness and damping matrices.

The three variants of induce can be described by one matrix equation with sinus Q_s and cosine Q_c amplitudes know from measurements (5)

$$\underline{B}\ddot{\underline{Q}} + \underline{C}\dot{\underline{Q}} + \underline{K}\,\underline{Q} = \underline{F} \qquad (5)$$

where

$$\underline{Q} = \underline{Q}_s \sin wt + \underline{Q}_c \cos wt$$

After the algebraization of the problem, the matrix system of equations with unknows stiffness matrix K and dumping matrix C is obrained.

Having solved the foregoing system of equation, we get the final formulas for K and C matrices coming from the vibration measurement results

$$\underline{C}^e = \begin{bmatrix} c_{11} & c_{12} & c_{13} \\ c_{21} & c_{22} & c_{23} \\ c_{31} & c_{32} & c_{33} \end{bmatrix} \quad \underline{K}^e = \begin{bmatrix} k_{11} & k_{12} & k_{13} \\ k_{21} & k_{22} & k_{23} \\ k_{31} & k_{32} & k_{33} \end{bmatrix} \qquad (6)$$

If we compare the theoretical matrices and experimental ones we get a system of independent conditions that allows to determined k_x, k_z and c_x, c_z. The remaining elements form control and identification conditions (7). From the stiffness matrices we get

$$\begin{aligned} \underline{K} = \underline{K}^e \quad &\Rightarrow \quad 4k_x = k_{11} \quad \Rightarrow \quad k_x = \frac{1}{4}k_{11} \\ &4k_x \frac{c^2}{b_w^2} + 4k_z = k_{22} \quad \Rightarrow \quad k_z = \frac{1}{4}\left(k_{22} - k_{11}\frac{c^2}{b_w^2}\right) \end{aligned} \qquad (7)$$

with control condition

$$4k_x \frac{c^2}{b_w^2} + 8k_z = k_{33} \quad \Rightarrow \quad k_z = \frac{1}{8}\left(k_{33} - k_{11}\frac{c^2}{b_w^2}\right)$$

For the dumping matrices we get

$$\begin{aligned} \underline{C} = \underline{C}^e \quad &\Rightarrow \quad 4c_x = c_{11} \quad \Rightarrow \quad c_x = \frac{1}{4}c_{11} \\ &4c_x \frac{c^2}{b_w^2} + 4c_z = c_{22} \quad \Rightarrow \quad c_z = \frac{1}{4}\left(c_{22} - c_{11}\frac{c^2}{b_w^2}\right) \end{aligned}$$

with control condition

$$4c_x \frac{c^2}{b_w^2} + 8c_z = c_{33} \quad \Rightarrow \quad c_z = \frac{1}{8}\left(c_{33} - c_{11}\frac{c^2}{b_w^2}\right)$$

5. TROLLEYS STIFFNESS AND DAMPING COMPUTATION RESULTS

The presented above stiffness and damping identification algorithm was implemented in Pascal programming language. The table 1 contains results of the identification of the linear, damping-elastic system at the inducing frequency 700Hz.

Table 1 *Stiffness and dumping computations results*

	Manufactures' parameters	f = 700 Hz
k_z [MN/m]	200/303	153
c_x[kNs/m		9.7
c_z [kNs/m]		28.1

6. CONCLUSIONS

The responses of the examined module to the input signal shows that the entire system can be approximately modeled as a linear damping-elastic system for the frequency 700Hz. The nonlinearity of elastic features of trolleys caused some distortions of the system's harmonic response. However, it is possible to construct the trolleys stiffness computation algorithm using one frequency – in our case it was 700Hz. the computations, for this frequency, give a positive results. The values of the horizontal and vertical stiffness are close to the values from the catalogues – the value of the horizontal stiffness is 10 times lower from the catalogue one, while the value of vertical stiffness $k_z = 150 MN/m$ is 20% smaller than catalogue one for tension. The small value of the horizontal stiffness can be justified by the fact that the construction of the test bed does not fully guarantee that the foundation is not susceptible in horizontal direction.

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