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SOME ASPECTS FOR DETERMINING THE VALUES OF THE DYNAMIC STABILITY THRESHOLD AND FOR BUILDING THE STABILITY DIAGRAMS AT THE CENTRES OF WOOD PROCESSING

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Abstract: The equations of the stability threshold are determined in an easy programmed form, by building matrices specific to the splinting process, necessary to the vector of the splinting process. The equations of the stability threshold have the splint's width as the unknown, and the frequency of the self-vibrations at splinting as the independent variable. The dynamic stability diagram is obtained in $b_{\lim} - t$ coordinates by solving the equations.

For calculating the dynamic stability of the centres of wood processing, average values experimentally determined for $\psi \in [0,2\pi]$ and for *t* frequencies can be considered for the coefficients of the dynamic splinting force, in the usual frequency field of specific vibration ways of the elastic structures at the processing centres $(t \in [5,400]Hz)$. [1], [2], [3]

The transfer function of the elastic structure of the processing centres is defined on the basis of the following hypotheses:

1). the elastic system of the machine tool can be represented on a structural model with concentrated parameters;

2). the elastic structure of the machine tool is a linear system with *n* own vibration ways, with different directions according to the cinematic reference system of M.U., OUVW (according to ISO standards). [1], [2], [3]

Thus noting with $G_{uu}(\omega), G_{vv}(\omega), G_{ww}(\omega)$ the direct receptivity of the elastic system of the machine tool corresponding to the directions of the reference system axes OUVW and taking into account the superposition principle applied for the linear systems we'll have:

$$G_{uu}(\omega) = \sum_{i=1}^{n} G_{ww,i}(\omega) = \sum_{i=1}^{n} R_{e_{ii,i}}(\omega) + j\sum_{i=1}^{n} I_{m_{nn,i}}(\omega) \qquad (1)$$

And similar for $G_{vv,i}(\omega)$ si $G_{ww,i}(\omega)$ [1], [4]

The reaching of the dynamic stability threshold of the close linear system of M.U. results from the interaction of the machine's elastic structure with the splinting process.

Thus, the moving on the normal direction to the splinting instantaneous surface results from: [1], [4]

$$Y = P_{d,n} G_{nn}(\omega) + P_{d,v} G_{vv}(\omega) + P_{d,w} G_{ww}(\omega) \qquad (2)$$

Where: $P_{d,n}, P_{dv}$ si $P_{d,w}$ are the components of the dynamic splinting force P_d (relation (3)), after the directions of the reference system axes OUVW. [1], [4]

$$P_d = -b\left\{\left[r + j\left(i + \omega \overset{\approx}{i}\right)\right]Y - (r_0 + ji_0)Y_0\right\} \qquad (3)$$

By introducing the components $P_{d,u}; P_{dv}; P_{d,w}$ in relation (2) results: [1], [4]

$$Y\left(b^{-1}+\sum_{S=u,v,w} C_S \cdot G_{SS}(\omega)\right)=Y_0\left(\sum_{S=u,v,w} D_S \cdot G_{SS}(\omega)\right) \quad (4)$$

Imposing the limit condition of stability in the splinting process, we obtain: [1], [4]

$$\left|Y / Y_0\right|=1 \quad (5)$$

and it results:

$$\left|b^{-1}+\sum_S C_S \cdot G_{SS}\right|=\left|\sum_S D_S \cdot G_{SS}\right| \quad (6)$$

The threshold relation (6) can be brought to an easy programmable form solved with IT methods. For this we have the matrix – line of the elastic structure of the machine tool with the form: [1], [4]

$$B=\left\|R_{e_1},\ldots,R_{e_n},\ldots,I_{m_1},\ldots,I_{mn}\right\| \quad (7)$$

Then the modal directional factors are introduced, corresponding to the components of the splinting dynamic force $R_i, \tilde{\tilde{R}}_i, I_i, \tilde{\tilde{I}}_i$.

With these modal directional factors the matrices specific to the splinting process are built, that include the influence of the processing material (in this case, wood), of the specific splinting regime on the performances of M.U. dynamic stability. These specific matrices are: [1], [3], [4]

$$M=\left\|\begin{matrix} M_1 & -M_2^T \\ -M_2 & M_2 \end{matrix}\right\|; \qquad \overline{M}=\left\|\begin{matrix} M_3 & M_2 \\ M_2^T & M_1 \end{matrix}\right\|$$

$$N=\left\|\begin{matrix} N_1 & -N_2^T \\ -N_2 & N_3 \end{matrix}\right\|; \qquad \overline{N}=\left\|\begin{matrix} N_3 & N_2 \\ N_2^T & N_1 \end{matrix}\right\| \quad (8)$$

Where the superior index T is significant for the transposing of the respective matrices and the generating of the sub-matrices $M_k, N_k (k=1.2,3)$ is simplified taking into account the symmetric character of these matrices.

Then, the vector characteristic to the splinting process is built: [1], [3], [4]

$$C=\left\|R_1,\ldots,R_n\left(I_1+J_1\right),\ldots,\left(I_n+J_n\right)\right\| \quad (9)$$

Finally, we define the global matrix *P* characteristic to the splinting process, under the form of: [1], [3], [4]

$$P=M+\overline{M}-N-\overline{N} \quad (10)$$

With the above mentioned notes, after elementary transformations, the equation of the stability threshold (G_T) takes the form:

$$b^{-2}+2b^{-1}BC^T+BMB^T+B\overline{M}B^T=BNB^T+B\overline{N}B^T \quad (11)$$

or

$$b^{-2}+2b^{-1}BC^T+BP=0. \quad (12)$$

[1], [3], [4]

Where the unknown is the splint's width at stability limits, and the independent variable is the frequency of the self-vibrations at splinting.

By the variation of t frequency in the interest field $(t = 5...500Hz)$ and by solving the threshold equation (12), for a dynamic system of the machine tool, experimentally identified, diagrams of dynamic stability can be obtained in $b_{\lim} - t$ coordinates.

The threshold equation (12) simplified, becomes: [1], [3], [4]

$$b^{-2} + 2b^{-1}BC^{T} = 0 \qquad (13)$$

that admits the simple solution $b^{-1} = 0$ and the solution: [1], [3], [4]

$$b^{-1} = -2BC^{T} = -2\sum_{i=1}^{N} R_{e_1} R_i \qquad (14)$$

Relation that defines the width of the non-detached splint at the stability threshold of the machine tools in the splinting process, within some usual analyses of the dynamic performances of the machine.

The determining of the values for the dynamic stability threshold and the building of some stability diagrams in coordinates $(b_{\lim}, t)$ according to equation (12) presupposes the performing of the following stages:

1). The direct total real receptivity is experimentally determined: $R_{e_{nn}}(t), R_{e_{vv}}(t), R_{e_{ww}}(t)$, corresponding to the directions of the reference system axes OUVW of the machine tools;

2). When applying the method of successive decomposing for processing the curves of the real receptivity $R_{e_{SS}}(t)$, the parameters of the elastic structure and M.V. t_i sau ω_i are deduced. Knowing the proper directions *(X_i)* of the linear elastic structure of M.U. and defining them for a given splinting scheme, the direction of the maximum excitation *(Y)* of the dynamic system of M.V. and the splinting direction *(P)* of the same system, it results the directional factors $U_{u_i}, U_{v_i}, U_{w_i}$;

3). For a processing material given by a known splinting scheme and a chosen splinting regime, the coefficients $r, i, \approx{i}, r_0, i_0$ of the dynamic splinting force are experimentally determined, as average values of various measures where the de-phasing of vectors *Y* and *Y₀* takes successively in field $[0, 2\pi]$;

4). Knowing the structural parameters of the machine tool and the coefficients of the dynamic splinting force, the scalars $R_i, \approx{R}_i, I_i, \approx{I}_i, J$ are successively calculated and the matrices $M, N, \overline{M}, \overline{N}, C$ si P specific to the splinting process are built as structural matrix *B*, finally resulting the form (12) of the dynamic stability threshold equation;

5). The coefficients and the free terms of the threshold equation (12) being functions of self-vibration pulsing $\omega = 2\pi t$, values for pulsation are given successively in the practical field for the investigated machine tool. For every pulsation $\omega_0 = 2\pi t_0$ by solving the threshold equation (12) the limit width $b_0 \lim$ is obtained, at the stability threshold M.U. The results are graphically transposed in stability diagrams of $b_{\lim}(t)$ type (fig.1). [1], [2], [3]

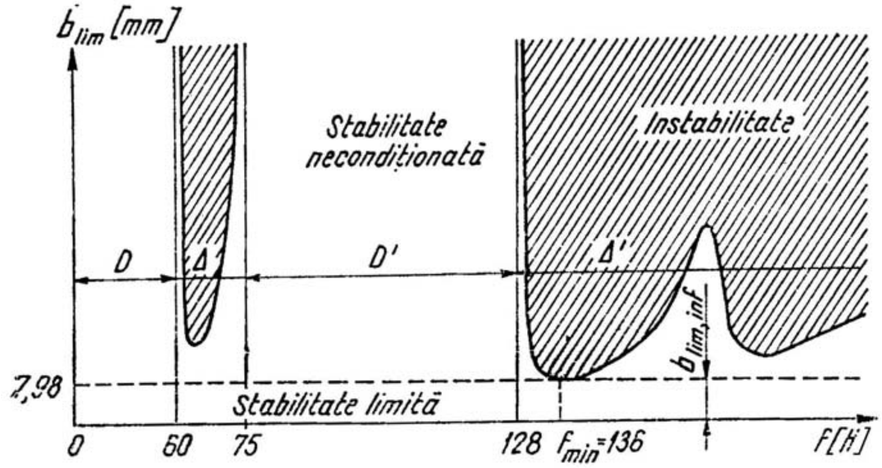

FIG. 1 The stability diagram $b_{\lim}(t)$ [1], [2], [3].

By the stability diagrams $b_{\lim}(t)$ (fig. 1) there can be defined the stability cases and the cases of conditioned or unconditioned stability. [1], [2], [3]

Analysing the stability diagram (fig. 1) we notice that the minimum points of curve $b_{\lim}(t)$ constitute the critical cases, and are produced at a local minimum width of the splint.

The co-domain of width values determined by the relation $b < b_{\lim,\inf}$, define the area of limit stability of the machine tool in the splinting process. In this area, the dynamic system of the machine tool is stable for any value of the self-vibration frequency. [1], [2], [3]

For some frequency fields, the dynamic system of the machine tool does not lose its stability no matter what the value of the splint's size is. These fields form the unconditioned stability areas of the machine tool, and between the limits of this area, the intensity of the splinting regime is limited only by the power reserve of the machine tool. [1], [2, [3]

In other frequency fields, the dynamic stability of the machine tool is ensured only if $b < b_{\lim}$. These fields define the conditioned stability areas in the sense that the dynamic system of the machine tool is stable if the system $b < b_{\lim}$ and it is unstable if $b > b_{\lim}$. For the diagram in fig. 1, the fields of conditioned stability, noted by $\Delta \text{ si } \Delta'$ are given by $\Delta\{t \in [61,74] Hz; b < b_{\lim}\} \text{si } \Delta'\{t \geq 129 Hz; b < b_{\lim}\}$. [1], [2], [3]

In the same frequency fields $\Delta \text{ si } \Delta$, if $b > b_{\lim}$, the dynamic system of the machine tool is instable in the splinting process. The fields of dynamic instability are hachured in fig. 1. [1], [2], [3]

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