COMAT

The 9th International Conference on Advanced Composite Materials Engineering

Transilvania University of Brasov
FACULTY OF MECHANICAL ENGINEERING

17-18October 2022

MILLING MACHINING MODEL AS MECHANICAL SYSTEM WITH 1DOF

Mupona Munyaradzi Innocent[1], Claudiu POZNA[2] Ioan Călin ROȘCA[*3]

1.Harare Institute of Technology, Harare, Zimbabwe, mimmuponal@gmail.com
2.Transilvania University of Brașov, Brașov, Romania, cp@unitbv..ro
3.Transilvania University of Brașov, Brașov, Romania, icrosca@unitbv.ro
[*]Corresponding author: icrosca@unitbv.ro

Abstract: One of the main problems in machining is the tool chatter that can be modelled as a time-delayed mechanical system. As a physical phenomenon, chatter represents the self-excited vibration that has as cause the interaction that exists between the machine structure and the cutting process dynamics.

Keywords: *dynamic model, modal parameters, vibrations*

INTRODUCTION

One of the main problems in machining is the tool chatter that can be modelled as a time-delayed mechanical system. As a physical phenomenon, chatter represents the self-excited vibration that has as cause the interaction that exists between the machine structure and the cutting process dynamics.
Chatter is strong connected with the stability of the mechanical system composed of the machine tool-work piece-cutting tool. Generally, chatter can be highlight through signal measurements during the cutting processes. The recorded data during machining offer a limited view of the phenomenon.
Chatter is a dynamic phenomenon that is connection with stability loss during the stationary machining process and which is followed by large amplitudes of self-excited vibration generated between the cutting tool and the work piece.
The first studies concerning the machining stability were done in the 1950s and 1960s, by Tobias [1] and Tlusty [2]. They defined the so called regenerative effect that was accepted as explanation for machine tool chatter.

The stability of the machining process can be evaluate using the lobe diagrams expressed in depth of cut and spindle speed parameters. Based on stability lobes one can decide on optimal technological parameters to choose a machining regime without chatter.
The stability study is based on some mathematical models and methods connected to the motion equations that are time periodic delay-differential equations.
The used methods are different, mainly numerical methods, such as: analysing of frequency response functions [3, 4, and 5], discretization methods [6, 7], the iteration methods [8, 9], etc.

1.MILLING DYNAMIC MODEL 1DOF

The study of stability in machining involves defining a dynamic system and its motion equations. Taking into consideration that inside equations are dynamic parameters of mass, damping and stiffness it is needed to be done an identification of them.
The used dynamic models are with 1DOF [10, 11, 12] or 2DOF [13, 14, 15] considering the parameters identified by experimental modal analysis method.
The model with 1DOF is governed by the motion equation:

$$m \cdot \ddot{x}(t) + c \cdot \dot{x}(t) + k \cdot x(t) = F(t) \tag{1}$$

which can be rewritten in state space representation as:

$$\dot{\mathbf{X}}(t) = \mathbf{A}\,\mathbf{X} + \mathbf{BU} = \begin{bmatrix} 0 & 1 \\ -\dfrac{k}{m} & -\dfrac{c}{m} \end{bmatrix} \begin{Bmatrix} x_1 \\ x_2 \end{Bmatrix} + \begin{Bmatrix} 0 \\ \dfrac{1}{m} \end{Bmatrix} F(t) \tag{2}$$

where m is the modal mass, c is the modal damping, and k is the modal stiffness.
Modal parameters are to be estimated using the measured FRF of the machine-tool-workpiece system.
The model described by the equation (1) is a linear one. In [16] it is introduced the processed damping force model, the damping force being expressed as:

$$F_d(t) = -C_d \frac{w}{v_c} \dot{x}(t) \tag{3}$$

where C_d is the process damping coefficient, w is the depth of cutting, v_c is the cutting speed.
The cutting force can be expressed as:

$$F(t) = -Cw\left[\, x(t) - x(t-\tau)\right] \tag{4}$$

with the delay $\tau = 60/\Omega$, C is a constant of cutting (cutting coefficient [16]), and Ω angular velocity of the cutting tool expressed in rotation/minute. Introducing (3) and (4) in (1) and taking into consideration the state space equation (2) it is obtained the new form, presented in [16] as:

$$\dot{\mathbf{X}}(t) = \begin{Bmatrix} \dot{x}(t) \\ \ddot{x}(t) \end{Bmatrix} = \begin{bmatrix} 0 & 1 \\ -\omega_n^2\left(1 - \dfrac{Cw}{k}\right) & -\left(2\zeta\omega_n + \dfrac{60}{\pi} C_d \dfrac{w}{\Omega D m}\right) \end{bmatrix} \begin{Bmatrix} x(t) \\ \dot{x}(t) \end{Bmatrix} + \begin{Bmatrix} 0 & 0 \\ -\omega_n^2 \dfrac{Cw}{k} & 0 \end{Bmatrix} \begin{Bmatrix} x(t-\tau) \\ \dot{x}(t-\tau) \end{Bmatrix} \tag{5}$$

The state space representation given by the equation (5) needs to be identified the modal parameters from (5).

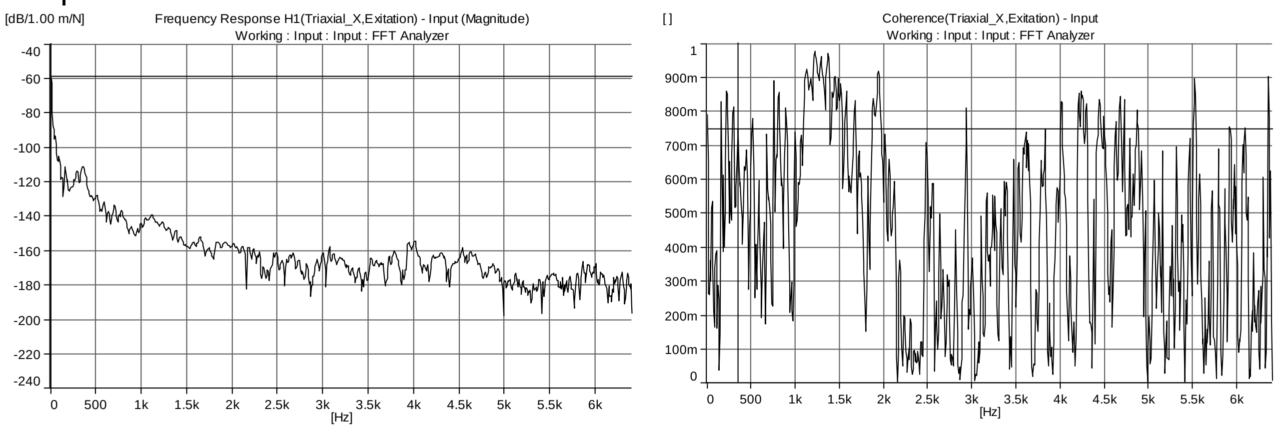
2.MODAL IDENTIFICATION

Modal parameters of the cutting tool were determinate considering a set up made of the milling tool, one accelerometer type 4507Bx (Brűel&Kjær), an impact hammer type 8206-003 (Brűel&Kjær), and the Platform PULSE 12 with modal analysis soft. The hit was done in the forward milling direction and the accelerometer was mounted in the same direction (Figure 1).

a) b)

Figure 1: Experimental set up: a) accelerometer position; b) hit direction

The obtained FRF curves are presented in Figure 2a, and the obtained values are presented in Table 1.



a) b)

Figure 2: Experimental results: a) frequency respone function; b) Coherence values

Table1. Data measured in experimental modal test

No.	Frequency $[Hz]$	$\zeta\ [\%]$	Real $[m/N]$	Imaginary $[m/N]$	Magnitude $[m/N]$	Delta 3dB	Choerence
1.	152.00	5.50	$2.01 \cdot 10^{-6}$	$1.52 \cdot 10^{-6}$	$2.52 \cdot 10^{-6}$	16.71	0.825
2.	288.00	5.37	$1.94 \cdot 10^{-6}$	$-8.85 \cdot 10^{-10}$	$1.94 \cdot 10^{-6}$	30.92	0.792
3.	352.00	6.01	$-2.37 \cdot 10^{-7}$	$-2.58 \cdot 10^{-6}$	$2.59 \cdot 10^{-6}$	42.35	0.745
4.	504.00	5.05	$-3.75 \cdot 10^{-7}$	$1.31 \cdot 10^{-8}$	$3.76 \cdot 10^{-7}$	50.90	0.733
5.	888.00	2.50	$-4.57 \cdot 10^{-8}$	$6.33 \cdot 10^{-9}$	$4.62 \cdot 10^{-8}$	44.33	0.808
6.	1112.00	2.68	$-5.30 \cdot 10^{-8}$	$-8.67 \cdot 10^{-8}$	$1.02 \cdot 10^{-7}$	59.65	0.906

The values of the modal stiffness and modal mass, for a mode „r" can be calculate using the following relations [17]:

$$k_r = \frac{-1}{2\zeta_r \operatorname{Im}\left[G_{ij}(\omega_r)\right]} \tag{6}$$

$$m_r = \frac{k_r}{\omega_{nr}^2} \tag{7}$$

3.SIMULATION

Based on the obtained data, presented in Table 1, it was considered the 6th cases where the coherence value is the best. For this case the data used in relation (5) are presented in Table 2.

No	$C\left[N/m^2\right]$	$\omega_n\ [rad/s]$	$k\ [N/m]$	$\zeta\ [\%]$	$C_d\ [N/m]$	$\Omega\ [rpm]$	$D\ [mm]$	$m\ [kg]$
1.	$1.3755 \cdot 10^{9}$	6986.9	$2.152 \cdot 10^{6}$	2.68	61,000	1,200	40	0.044

The vibration simulation of the milling tool during machining is presented in Figure 3.

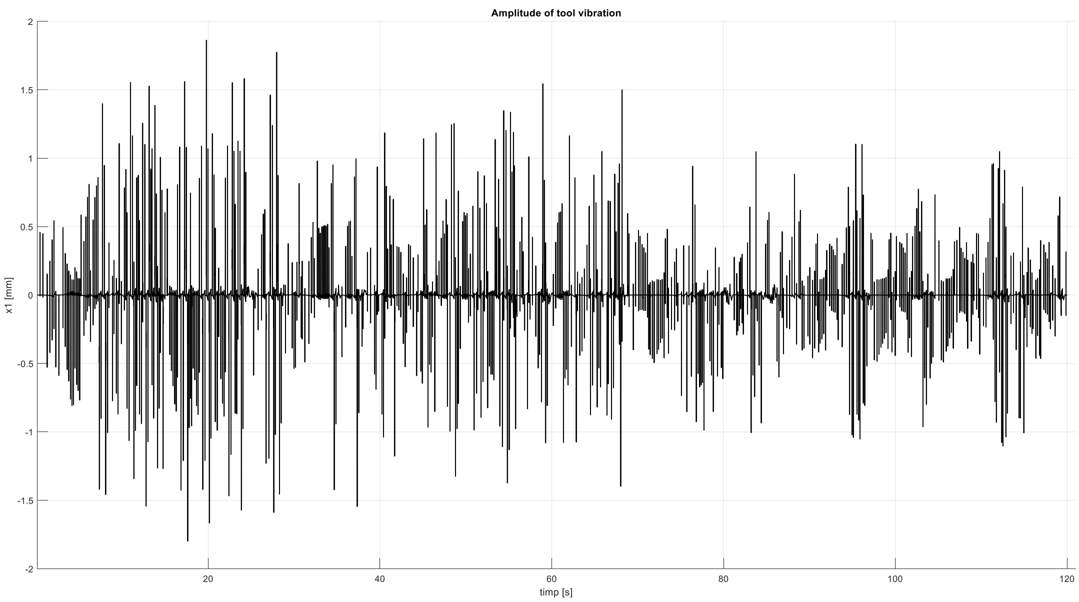


Figure 3: Vibration simulation of the milling tool

The time simulation was chose equal of 120 seconds. The simulation consists in script wrote in MATLA where was defined equatin (5) and its components, and a SIMULINK file where was solved equation (5). The the level of simulated amplitude is no larger than 2 mm.

4.CONCLUSIONS

Based on the state space equation presented in [16] it was done a simulation of the vibration level for a milling tool. The modal parameters were identified using a proper setup. The level of simulated vibration does not exceed the value of 2 mm. This value cannot be considered accepted considering that the machining has an imposed precision of the edges.

One reason of this value can be the 1DOF model considered that take into consideration only the motion on forward direction and do not consider the effect on normal direction influence on forward direction.

REFERENCES

[1] S. Tobias, Theory of Regenerative Machine, Tool Chatter, THE ENGINEER, February, 1958

[2] J. Tlusty, L. Spacek, Self-excited vibrations on machine tools, Prague, Czech: Nakl. CSAV (1954)

[3] Y. Altintas, E. Budak, Analytical prediction of stability lobes in milling, CIRP Ann.-Manuf. Techn. 44 (1995) 357–362.

[4] E. Budak, Y. Altintas, Analytical prediction of chatter stability in milling, part i: general formulation, J. Dyn. Syst. ASME 120 (1998) 22–30.

[5] D. Bachrathy, G. Stepan, Improved prediction of stability lobes with extended multi frequency solution, CIRP Ann.-Manuf. Techn. 62 (2013) 411–414

[6] T. Insperger, G. Stepan, Semi-discretization for Time-delay Systems, 178, Springer, New York, 2011

[7] Y. Ding, L.M. Zhu, X.J. Zhang, H. Ding, A full-discretization method for prediction of milling stability, Int. J. Mach. Tool Manuf. 50 (2010) 502–509.

[8] X.J. Zhang, C.H. Xiong, Y. Ding, M.J. Feng, Y.L. Xiong, Milling stability analysis with simultaneously considering the structural mode coupling effect and regenerative effect, Int. J. Mach. Tool Manuf. 53 (2012) 127–140

[9] M. Zatarain, Z. Dombovari, Stability analysis of milling with irregular pitch tools by the implicit subspace iteration method, Int. J. Dynam. Control 2 (2014) 26–34

[10] D. Hajdu, T. Insperger, G. Stepan, Robust stability analysis of machining operations, Int. J. Adv. Manuf. Technol. 88 (1) (2017) 45–54.

[11] Adam K. Kiss, David Hajdu, Daniel Bachrathy, Gabor Stepan, Operational stability prediction in milling based on impact tests, Mechanical Systems and Signal Processing 103 (2018) 327–339

[12] Mingzhen Li, Guojun Zhang, Yu Huang, Complete discretization scheme for milling stability prediction, Int J Adv Manuf Technol (2017) 88:45–54

[13] Y. Altintas, G. Stepan, D. Merdol, Z. Dombovari, Chatter stability of milling in frequency and discrete time domain, CIRP Journal of Manufacturing Science and Technology 1 (2008) 35–44

[14] Zoltan Dombovaria, Jokin Munoab, Gabor Stepana, General Milling Stability Model for Cylindrical Tools, 3rd CIRP Conference on Process Machine Interactions (3rd PMI), Procedia CIRP 4 (2012) 90 – 97

[15] Hamed Moradi, Gholamreza Vossoughi, Mohammad R. Movahhedy, Mohammad T. Ahmadian, Forced vibration analysis of the milling process with structural nonlinearity, internal resonance, tool wear and process damping effects, International Journal of Non-Linear Mechanics 54 (2013) 22–34

[16] J. L. Chukwuneke, C. O. Izuka, S. N. Omenyi, S-domain stability analysis of a turning tool with process damping, Heliyon 5 (2019) e01906, https://doi.org/10.1016/j.heliyon.2019.e01906

[17] Tony L. Schmitz, K. Scott Smith, Machining Dynamics. Frequency Response to Improved Productivity, 2019, Springer International Publishing, ISBN 978-3-319-93707-6, https://doi.org/10.1007/978-3-319-93707-6