DOI: 10.20535/2521-1943.2021.5.3.242214

UDC 621.9.048.6; 621.9.06-529-8

Modeling of the frequency response function and its evaluation during boring

M. Shykhalieiev[1] • V. Medvedev[1]

Received: 11 October 2021 / Accepted: 24 November 2021

Abstract. *Finite element method of simulating frequency response function (FRF) for boring tool in LS-Dyna solver is investigated in this work.*
Nowadays, computer numerical simulation allows to obtain FRF using different materials model with high precision compared to real experiments with sensors like impact hammer testing.
This function is used in construction of stability lobe diagrams that allows operator of machining center to avoid chatter self-excited vibrations.
Such vibration is led to decreasing of productivity and quality in cutting of metals and other materials.
Amplitude and phase angle for the model is obtained from LS-Dyna result interpreter, that reads binary files, created during simulation by the program.
Amplitude and phase angle of frequency response function are depending on dynamic stiffness of machining system. Real and imaginary part of frequency response function have been obtained during simulation.
With lack of dynamic stiffness amplitudes of response increases.

Keywords: *frequency response function; boring; simulation; vibration; vibroacoustic behavior of machining centers.*

Introduction

Machining of metals is accompanied by vibrations of components of machining centers. Bearings, tools, workpiece etc. These vibrations can be of different nature. For instance, they can occur as self-excited or forced vibrations [1]. And they led to reducing of productivity and quality of machined parts [1]. One way self-excited vibrations appear in boring is when dynamic stiffness of machining system is not enough.

In machining Frequency response function (FRF) is used to characterize natural frequencies of cutting tool, tool holder, workpiece, machine tool and its combination. This function is then used in stability lobe diagram construction (SLD) [2] as the main application. FRF contains information about damping of vibrations, resonance frequencies and mode shapes. This function can be obtained by numerical simulation using computer's software. And it has direct influence of stability of machining.

✉ *M.M. Shykhalieiev*
2texter18@i.ua

[1] *Igor Sikorsky Kyiv Polytechnic Institute, Kyiv, Ukraine*

Subject overview

One of the common method to suppress self-excited vibration in machining of metals, called chatter, is construction of stability lobe diagrams [2]. Basic procedure for this tusk is obtaining FRF from spindle add other machining structures. When transfer function had been identified, evaluating of cutting coefficients is necessary [2]. Different methods of experimental obtaining of frequency response function for specific machining tasks are introduced nowadays by researchers [4, 13]. Thus, it is necessary to obtain dynamic characteristics of the systems. Simulating FRF is simple method to obtain dynamic characteristic of structure without conducting direct experiment. Spectrum of structural response (displacement, velocity, force and acceleration) is computed due to applied unit harmonic excitations, they are given in modeling as nodal force with varying frequencies. Also, FRF can be computed using the method of mode superposition using frequency domain.

ISSN 2521-1943 Mechanics and Advanced Technologies

Dynamic of workpiece can be obtained directly with impact hammer taking into account that accelerometer can be easily mounted. But since the tool tip and cutting edge of instrument usually has small diameter, accelerometer cannot be properly installed to obtain data for frequency responses. In addition, accelerometer can create its influence on frequency response function of measured structure. It makes measures unreliable, so additional approaches need to be applied in experiments.

It is known that installed sensors has impact of FRF of the system structure, so experimental approach need to get rid with effect of sensor quality [8, 9]. And static stiffness of structures is differing from dynamic ones [14]. And this approach also can be assisted by numerical simulation. Variable mass of the boring system is researched in [6]. Dynamics of boring bar with reservoir, containing fluid with changeable volume, simulated using Ansys software. Cutting forces and vibration of titanium alloys machining is described in [11]. It is shown, that FRF of tool tip can be obtained by modeling instead of specific experimental approaches.

Simulation of FRF for boring tool model is proposed in this study, using 6-degree of freedom formulation and dynamic stiffness in three standard directional vectors X, Y, Z of cutting in LS-Dyna solver.

The aim of the study

Object of the study is simulation of responses of boring cutting tool and its edge to FRF analysis. Aim is to obtain frequency response function as amplitude and phase representation and real and imaginary part representation. And its evaluation.

Main material

Common method to obtain FRF is to use impact hammer with accelerometer, as shown of fig. 1. This is the mechanical method to obtain FRF. Another types of input excitations and response outputs are acoustic (output sound preassure (Pascal)/ input volume acceleration (m^3/s^2)), combined Acoustic-Mechanical (uses force as input and sound preassure plus acceleration as output), rotational (rotational displacement / torque). Responses are measured by accelerometers, lasers etc.

Although, FRF can be simulated using computer methods of modeling. Spectrum of structural response (displacement, velocity and acceleration) for applied unit harmonic excitations can be simulated in LS-Dyna solver [3]. Eigenvalues of machining system must be included in simulation of FRF, because the eigenvalues of the closed-loop dynamic system equation allow to construct stability lobe diagram (SLD) that show the maximum stable axial depth of cut as a function of spindle speed. Also, it is necessary to use implicit dynamic simulation to extract and process eigenvalues. In Ls-Dyna they are written to binary file by the solver. FRF is complex function, that has real and imaginary part. Correlation between amplitude and phase can be described by equation

$$\begin{cases} Amplitude = \sqrt{\mathrm{Re}^2 + \mathrm{Im}^2} \\ Phase = \tan^{-1}\left(\dfrac{\mathrm{Im}}{\mathrm{Re}}\right) \end{cases} \tag{1}$$

where Re – real part of FRF, Im – imaginary part of FRF.

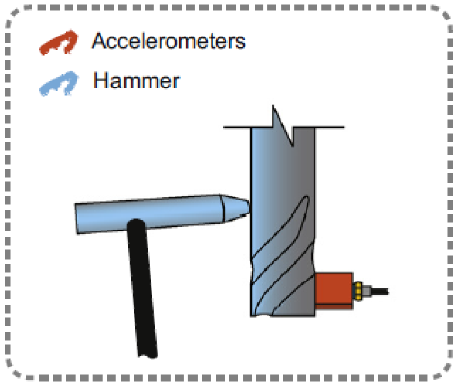


Fig. 1. Procedure for obtaining FRF using impact hammer test [2]

In digital processing block diagram of FRF can be illustrated as shown in fig. 2

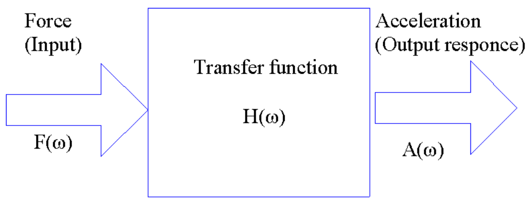


Fig. 2. Block diagram of FRF

$$H(\omega) = \frac{A(\omega)}{F(\omega)} \tag{2}$$

where $H(\omega)$ – transfer function for FRF, $A(\omega)$ – acceleration response, $F(\omega)$ – input force.

Fig. 3 show simplified computer model to simulate FRF. Dynamic stiffness coefficient k (10000–30000 N/mm) and damper coefficient in viscous form are also included in model to simulate dynamic stiffness of machine tool. Material of the tool is steel. Impulse force is applied on the cutting edge. A single-mass system is used. There are 6 degrees of freedom for each of directional axis. Computing FRF for model is provided with LS-Dyna keyword FREQUENCY_DOMAIN_FRF. Also, keyword CONTROL_-FREQUENCY_RESPONCE_FUNCTION is performed in the model and provides information about location of response and range of frequencies for the harmonic nodal force excitation. Keyword CONTROL_IMPLICIT_EU

GENVALUE is included to run modal analysis at the first step. Spindle is not taken into account in this study. Model of material of cutting insert is having elasticity properties and define in solver as material type number 001 MAT_ELASTIC. Which has parameters of mass density 7870 kg/m^3, Young's modulus 207 GPa and Poisson's ratio equal 0.3 [7].

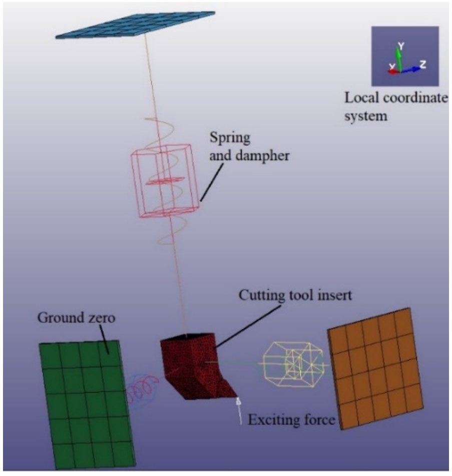
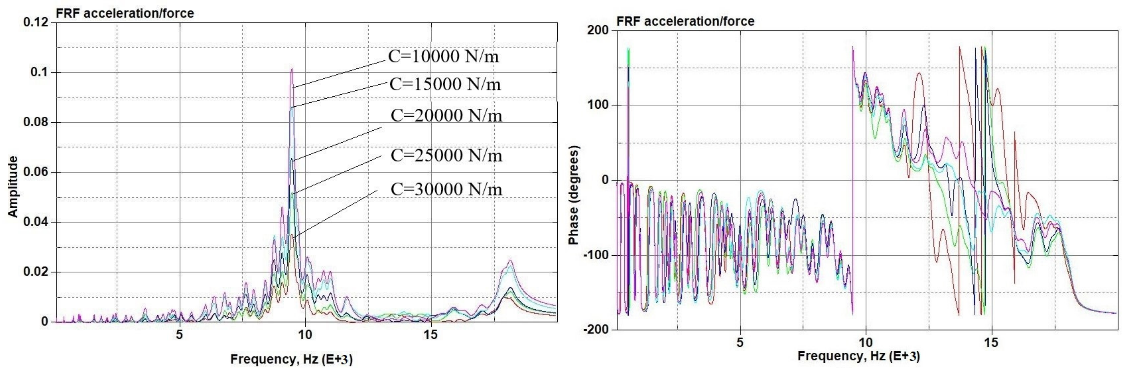


Fig. 3. Frequency response function model

Graphic of FRF is shown on fig. 4, 5. Fig. 4 show amplitude dependence. And fig. 5 show phase angle. The highest peak of signal amplification is under 3000 Hz. Its amplitude is about 80 dB. And this frequency has the highest phase angle oscillation. Frequencies-amplitude relationship of FRF are functions of dynamic stiffness of the machine center [5, 6]. It means that ratio of input/output signal is such that the greater the dynamic stiffness of the system, the smaller the amplitudes of the output signal. And its dynamic characteristic depends on material of the instrument [4]. Comparison of system with different dynamic stiffness (k = 20000 N/mm and k = 15000 N/mm) is shown on fig. 7. Peak amplitude is more than 150 dB for less dynamic stiffness.

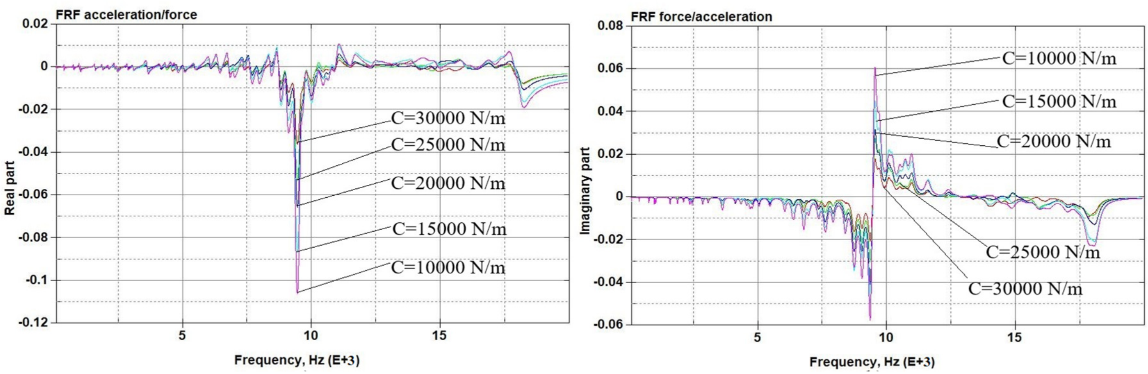
Results of simulation

Graphic representation of simulation is shown in fig. 4–10. Dynamic parameters to change in simulation is stiffness with the same damping constant. Acceleration response to exited force is given in fig. 4–5. Amplitude and phase angle of acceleration output for set of simulation are shown on fig. 4. Real and imaginary part of acceleration output of FRF is shown in fig. 5.



a *b*

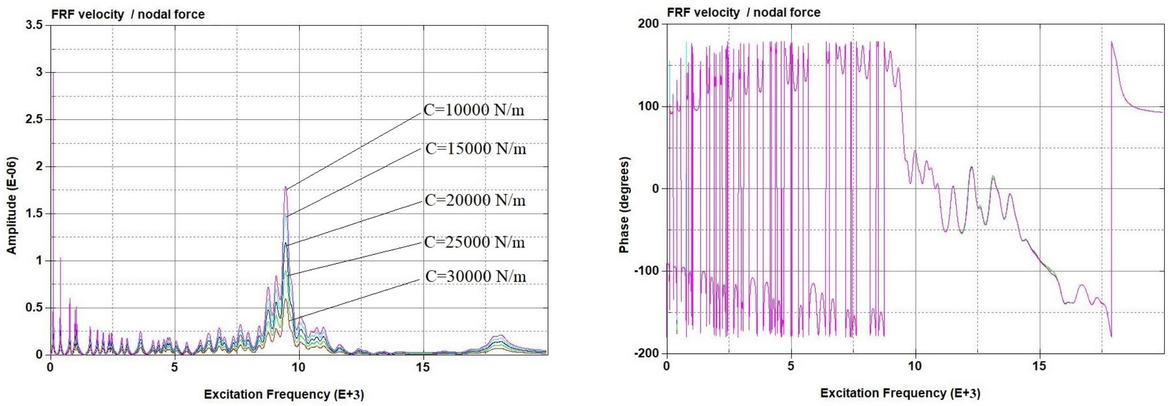
Fig. 4. *a*) Amplitude and *b*) phase angle for different dynamic stiffness of the system (damping is constant) for acceleration output. Each color corresponds to same parameter between *a*) and *b*)



a *b*

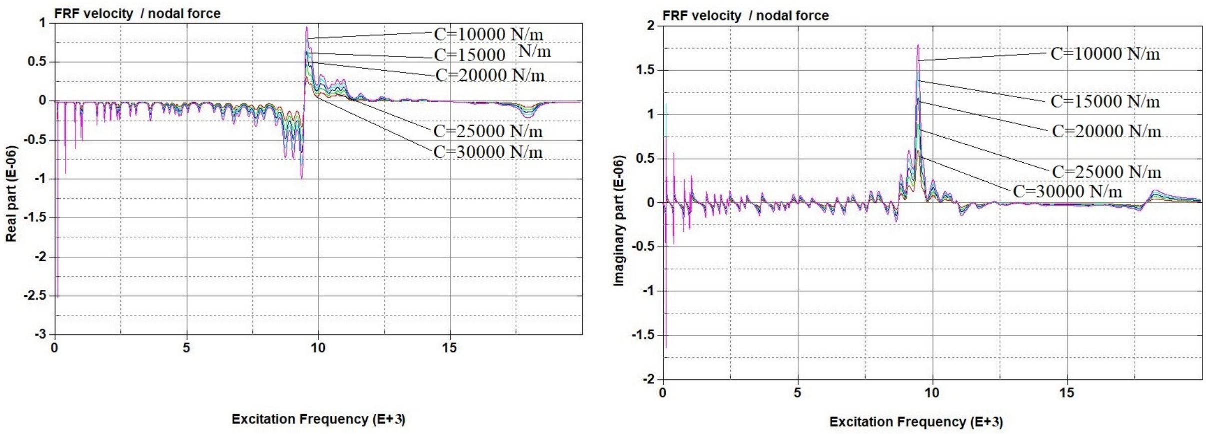
Fig. 5. *a*) Real and *b*) imaginary part of FRF for acceleration output

As shown on fig. 4 *a*, maximum amplitude of system response is achieved at about 1000 Hz. And the highest amplitude is 100 dB for dynamic stiffness 10000 N/mm. The lowest amplitude is about 40 dB for dynamic stiffness 30000 N/mm. Other amplitude growth is about 1800 Hz and it is lowest than previous. It is about 22 dB for dynamic stiffness 10000 N/mm and 10 dB for 30000 N/mm. Phase angle is shown of fig. 4 *b*, and have complicated form.



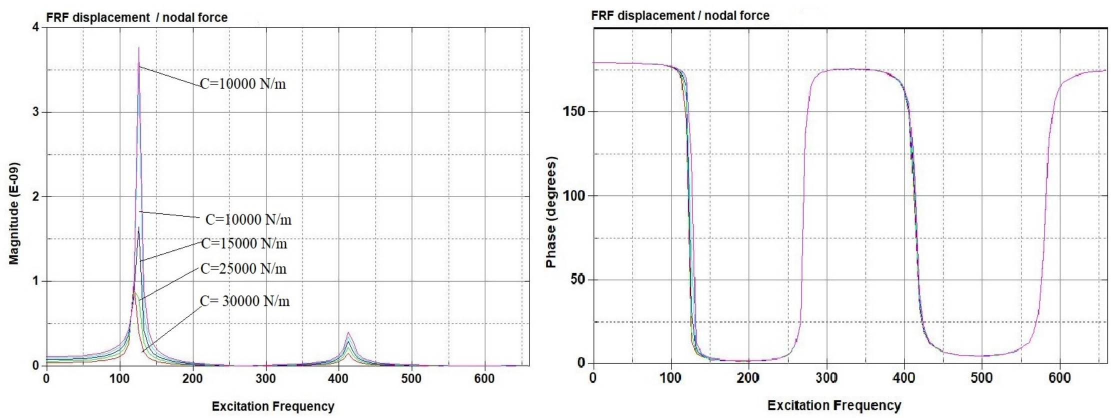
a *b*

Fig. 6. *a*) Amplitude and *b*) phase angle for different dynamic stiffness of the system for velocity output



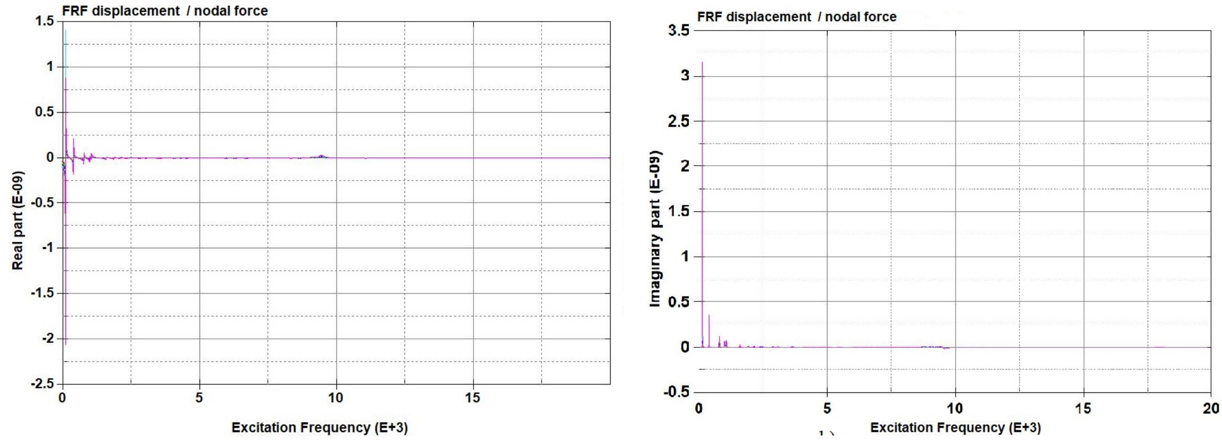
a *b*

Fig. 7. *a*) Real and *b*) imaginary part of FRF for velocity output



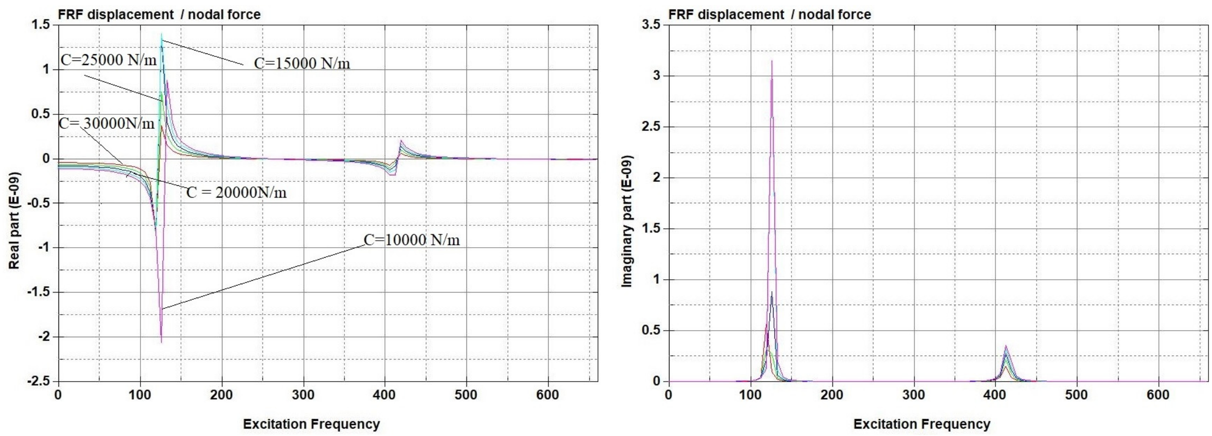
a *b*

Fig. 8. *a*) Amplitude and *b*) phase angle for different dynamic stiffness of the system for displacement output



a *b*

Fig.9. *a*) Real and *b*) Imaginary part for different dynamic stiffness of the system for displacement output given on frequency range up to 2000 Hz



a *b*

Fig. 10. *a*) Real and *b*) Imaginary part for different dynamic stiffness of the system for displacement output given on frequency range up to 600 Hz

The most changeable volume of phase is about 1000 Hz. Each color of the curve is corresponding to certain stiffness parameter of the model. Parameters to change in the model is generalized dynamical stiffness of the system.

So, under condition of same dynamical stiffness proportion in three axes amplitude of simulated system is increased with decrease of its dynamic stiffness value. Simulated amplitude response is inverse proportion of stiffness.

Graphs of output responses by velocity are given on the fig. 6–7.

Graphs of output responses by displacement are given on the fig. 8–10.

As shown on the fig. 8–10, for displacement structural reaction of the system, the highest peak of frequency response is between 100 and 200 Hz. Another sufficient peak is about 400 Hz.

The diameter of the boring tool is 20 mm and the length of the console is 200 mm.

Conclusions and discussion

FRF for boring can be obtained from computer numerical simulation in LS-Dyna nonlinear solver. For this purpose, there are many models of materials. Structural dynamic responses of simplified one-mass boring system with full degrees of freedom have been simulated. Amplitude, phase angle, real and imaginary part of FRF output in terms of acceleration, velocity and displacement with different dynamic stiffness of system have been determined. Material of boring cutting edge is not changed in simulation and have the same parameters for each simulation. With luck of dynamic stiffness of the system, amplitude of responses is increased as shown on the simulated graphics figures. As shown on the figures, the highest peak for acceleration and velocity output is about 1000 Hz.

References

[1] Z. Dombovari, M. Sanz-Calle and M. Zatarain, "The Basics of Time-Domain-Based Milling Stability Prediction Using Frequency Response Function", *Journal of Manufacturing and Materials Processing*, vol. 4, no. 3, p. 72, 2020. doi: 10.3390/jmmp4030072

[2] G. Quintana and J. Ciurana, "Chatter in machining processes: A review", *International Journal of Machine Tools and Manufacture*, vol. 51, no. 5, pp. 363-376, 2011. doi: 10.1016/j.ijmachtools.2011.01.001

[3] D. Kryvoruchko and V. Zaloha, Simulation of cutting processes by the finite element method: methodological foundations: monograph. Sumy: University book, 2012, p. 434.

[4] A. Iglesias, J. Munoa, C. Ramírez, J. Ciurana and Z. Dombovari, "FRF Estimation through Sweep Milling Force Excitation (SMFE)", *Procedia CIRP*, vol. 46, pp. 504-507, 2016. doi: 10.1016/j.procir.2016.04.019

[5] Y. Lin, K. Wu, W. Shih, P. Hsu and J. Hung, "Prediction of Surface Roughness Based on Cutting Parameters and Machining Vibration in End Milling Using Regression Method and Artificial Neural Network", *Applied Sciences*, vol. 10, no. 11, p. 3941, 2020. doi: 10.3390/app10113941

[6] Y. Alammari, M. Sanati, T. Freiheit and S. Park, "Investigation of Boring Bar Dynamics for Chatter Suppression", *Procedia Manufacturing*, vol. 1, pp. 768-778, 2015. doi: 10.1016/j.promfg.2015.09.059

[7] "LS-DYNA® Keyword User's Manual Volume II Material Models", *Dynasupport.com*, 2020. [Online]. Available: https://www.dynasupport.com/manuals/ls-dyna-manuals/ls-dyna_manual_volume_ii_r12.pdf

[8] "LS-DYNA Manual R12.0-Vol I", *Welcome to the LS-DYNA support site*, 2020. [Online]. Available: https://www.dynasupport.com/manuals/ls-dyna-manuals/ls-dyna_manual_volume_i_r12.pdf/view

[9] D. Ewins, *Modal testing*. Hertfordshire: Research Studies Press, 2000.

[10] Silva JMM, Maia NMM, Ribeiro AMR. Cancellation of massloading effects of transducers and evaluation of unmeasured frequency response functions. J Sound Vib 2000;236(5):761–779. https://doi.org/10.1006/jsvi.1999.2993

[11] Y. Chen, "A study of the cutting forces and vibration characteristics in titanium machining," Ph.D. dissertation, New South Wales, 2015.

[12] G. Quintana and J. Ciurana, "Chatter in machining processes: A review," *International Journal of Machine Tools and Manufacture*, No. 51(5), 2011. pp.363–376 [Online]. https://doi.org/10.1016/j.ijmachtools.2011.01.001

[13] B.Powalka *et al*., "Identification of machining force model parameters from acceleration measurement," *International Journal of Manufacturing Research*, . No. 3, 2008, pp. 1–30. https://doi.org/10.1504/IJMR.2008.019210

[14] E. Wahyuni and T. Ji, "Relationship between Static Stiffness and Modal Stiffness of Structures," *The Journal for Technology and Science*, 2010. https://doi.org/10.12962/j20882033.v21i2.36

[15] J. Munoa *et al*., "Chatter suppression techniques in metal cutting," *CIRP Annals Manufacturing technology*, 2016, pp. 785–808. https://doi.org/10.1016/j.cirp.2016.06.004

Моделирование функции частотной характеристики и её оценка при расточке

М. Шихалеев, В. Медведев

Аннотация. *В данной работе исследовано методом конечных элементов функцию частотной характеристики для расточного инструмента с использованием высоко-нелинейного решателя LS-Dyna. Компьютерное численное моделирование позволяет получить амплитудно-фазовую частотную характеристику технической обрабатывающей системы с использованием модели различных материалов, с высокой точностью по сравнению с реальными экспериментами с датчиками, такими как испытания ударным молотком. Эта функция используется при построении лепестковых диаграмм устойчивости, позволяет оператору обрабатывающего центра избегать автоколебаний обрабатывающей системы. Такие вибрации приводит к снижению производительности и качества резки металлов и других материалов. Амплитуду и фазовый угол для модели получено из интерпретатора результатов LS-Dyna, который считывает двоичные файлы, созданные при моделировании программой. Они зависят от динамической жесткости системы обработки.*

Ключевые слова: *функция частотной характеристики; растачивание; моделирование; вибрации; виброакустичиские явления в обрабатывающих центрах.*

Моделювання функції частотної характеристики і її оцінка при розточуванні

М. Шихалєєв, В. Медведєв

Анотація. *У цій роботі досліджено методом скінченних елементів моделювання амплітудно-фазової частотної функції для розточувального інструменту в високонелінійному вирішувачі LS-Dyna. Комп'ютерне чисельне моделювання дозволяє отримати амплітудно-фазову частотну характеристику технологічної оброблюючої системи (ТОС) з використанням моделі різних матеріалів, з високою точністю порівняно з реальними експериментами з датчиками, такими як випробування ударним молотком. Ця функція використовується при побудові пелюсткових діаграм стійкості, що дозволяє оператору обробного центру уникати автоколивань системи. Такі вібрації призводить до зниження продуктивності та якості різання металів та інших матеріалів. Амплітуду і фазовий кут для моделі отримано з інтерпретатора результатів LS-Dyna, який зчитує двійкові файли, створені під час моделювання програмою. Вони залежать від динамічної жорсткості технологічної оброблюючої системи.*

Ключові слова: *амплітудно-фазова частотна характеристика; розточування; моделювання; вібрації; віброакустичні явища в обробляючих центрах.*