Simulation of Oscillation Frequency Effects on Complex Shape Part Milling

A S Yamnikov and M N Bogomolov

Dept. of Manufacturing Technology, Tula State University Tula, Russia

Abstract. The study shows that auto-oscillations occur even at low-speed turning. The paper includes a literature review revealing the reasons for machining vibrations. It is difficult to analytically describe the physical processes in the cutting area, so we used simulation. Moreover, milling is the most difficult process to describe and research. The variable magnitudes/directions of the cutting forces resulting in extra auto-oscillations make the analysis even more complex. It is shown that reducing the cutting speed leads to lower vibrations, and to lower output, higher rigidity, and damping capacity. It is important to know the natural frequency of the workpiece to choose the right cutting mode. The finite element method and SolidWorks CAD models were used for simulating the complex shape part dynamic behavior. We found out that the workpiece natural frequency is 6.5 times higher than the forced oscillation frequency.

1. Introduction

Cutting oscillations occur even in turning leading to lower part quality [1-3]. Methods to stabilize and control the cutting process are intensively studied [4, 5]. The available papers do not offer an analytical definition or the cutting process since the chip material is in its boundary state. So, most researchers apply numerical simulation of the physical phenomena in the cutting area [5-14]. The most complex process is milling because the magnitude and direction of the cutting forces oscillate systematically, so forced oscillation complement the auto-oscillations [8, 9]. Reduced cutting modes, higher rigidity and damping capacity can reduce cutting vibrations. Moreover, it can be achieved by changing the workpiece location and using vibration-resistant cutters [3-14].

Figure 1. Shaped sleeve
Another interference factor affecting thin-walled sleeve milling is the periodical variations of the cutting force magnitude and direction [13, 14]. Moreover, re-machining a workpiece with cutting traces produces a regenerative effect that boosts the oscillations.

Figure 1 shows a shaped sleeve used as an example. The wall thickness is 2.5 mm. The upper surface has two steps 1. Eight fillets 2, Rmax15-1 radius are to be machined. The surface 2 roughness should be Ra 6.3 and the cylindricity deviation should be 0.1 mm.

The workpiece is a 121×7 tube, GOST 8732-78 made of steel grade 30 (GOST 1050-88.) Such a configuration (large sleeve size to the wall thickness ratio) reduces the machining rigidity. It is another factor inducing vibrations [3-5, 13, 14]. That is why the non-rigid sleeve milling results in intense oscillations.

“In serial production at SPLAV Corp., the tube workpiece is milled by the RBH BCF3028S32-160 ball cutters with the ZCET 150CE replaceable inserts. The equipment is the Akira seiki performa v4 xp4-axis machining center. The workpiece was clearance-placed onto the rigid mandrel. The arbor was clamped into the dividing head of the 4th machine axis. A washer, a nut and a stud screwed into the rigid mandrel were used for clamping. The cutting mode was: cutting speed \( V = 282.6 \text{ m/min} \), milling RPM \( n = 3,000 \text{ min}^{-1} \), feed \( S_{\text{min}} = 450 \text{ mm/min} \), cutting depth \( t = 2.5 \) mm, milling width \( B = 3 \) mm” [3]. Correspondingly, the mill tooth impact frequency was \( \omega = (2 \times 3,000): 60 = 300 \text{ Hz} \).

## 2. Theoretical background

It is important to know the natural frequency of the workpiece to choose the right cutting mode. The finite element method (FEM) was used for the complex shape part dynamic modeling [10-14]. According to the FEM concept, the workpiece is divided into elements that form its finite element model. Refer to Figure 2.

![Figure 2. A cylinder model and its partitioning into finite elements](image)

A finite element model of the part was made in the SolidWorks CAD software. The key finite element motion displacement equation can be expressed as matrices [15]:

\[
\begin{align*}
\mathbf{m}^e \ddot{\mathbf{q}}_e + \mathbf{c}^e \dot{\mathbf{q}}_e + \mathbf{k}^e \mathbf{q}_e &= \mathbf{f}^e \\
\end{align*}
\]

(1)

Where \( \mathbf{m}^e = \int_V [\mathbf{N}]^T \rho [\mathbf{N}] dV \) is the finite elements mass matrix; \( \mathbf{c}^e = \int_V [\mathbf{N}]^T \mathbf{c} [\mathbf{N}] dV \) is the finite element damping matrix; \( \mathbf{k}^e \) is the finite element rigidity matrix; \( \mathbf{f}^e \) is the finite element nodal force vector; \( \mathbf{q}_e \) is the finite element nodal displacement vector; \( [\mathbf{N}] \) are the finite element shape functions; \( \rho \) is the density; \( c \) is the damping coefficient.
By summing up the contributions of each finite element we obtained a system of differential linear motion equations relative to the nodal displacement vector of the entire finite element model:

$$M_d \ddot{q} + C_d \dot{q} + K_d q = F$$

(2)

where $q$ is the finite element model nodal displacement vector; $F$ is the nodal force vector; $M_d$ is the mass matrix, $C_d$ is the damping matrix; $K_d$ is the structural rigidity [15].

3. Solution

A law of mechanics is: “A body disturbed from its rest position tends to vibrate at certain frequencies called natural, or resonant frequencies. For each natural frequency, the body takes a certain shape called mode shape. Frequency analysis calculates the natural frequencies and the associated mode shapes.”

We used the frequency analysis to estimate the part deformation at its natural frequencies, and the fraction of the part mass associated with these frequencies. The total effective mass in each direction can also be estimated. Refer to Table 1 for the frequency analysis results. The table lists mass contribution coefficients for the global X-Y-Z directions normalized to the total mass. The use of sections to divide the text of the paper is optional and left as a decision for the author. Where the author wishes to divide the paper into sections the formatting shown in table 2 should be used.

**Table 1. Mass contribution coefficients for the global directions**

| Mode No. | Frequency (Hz) | X axis % | Y axis % | Z axis % |
|----------|----------------|----------|----------|----------|
| 1        | 1,059.8        | 0        | 0        | 0        |
| 2        | 1,203.5        | 0        | 0        | 0        |
| 3        | **1,963.4**    | **0**    | **19**   | **52**   |
| 4        | **1,963.6**    | **0**    | **52**   | **19**   |
| 5        | 2,228.7        | 0        | 0        | 0        |
| 6        | 2,231.2        | 0        | 0        | 0        |
| Total effective mass, % | 71 | 71 |

The results show that the mode No. 3 at 1,963.4 Hz along the Z-axis is the most destructive one: 52% of the part mass is affected (refer to Figure 3).

**Figure 3. Stress distribution curve for mode No.3**
For the Y axis the most dangerous is mode No. 4 at 1,963.6 Hz: 52% of the part mass is affected (refer to Figure 4).

![Stress distribution curve for mode No.4](image)

**Figure 4.** Stress distribution curve for mode No.4

We can claim that central portion of the part is most vulnerable to deformations and vibrations.

4. **Conclusion**

The resonance oscillation frequencies along the Y and Z axis are almost equal because the material and the wall thickness distribution are nearly identical in two mutually perpendicular directions (ref. to Figure 1).

Natural frequencies of a specific workpiece are 6.5 times higher than the forcing dynamic load oscillation frequency, so no resonance can occur.

**Acknowledgments**

The reported study was funded by RFBR according to the research project «Postgraduate» №20-38-90248-RFBR.

**References**

[1] Lazarev G. Metal Cutting Stability. Moscow: Vysshaia Shkola Publishing, 1973. 184 p.
[2] Amosov V. Scrigan Turning: Accuracy, Vibrations, and Surface Finish [in Russian] Moscow, Leningrad. MASHGIZ Publishing. 1958. 91 p.
[3] Leontiev B.V., Leontieva A.N. Cutting Process Management to Remove Vibrations (Far East Federal University Vladivostok, 2012), pp. 159-162.
[4] Kuznetsov V., Yannikova O. Machining System Stability for Threading with Multi-Cutter Heads [in Russian] // STIN Journal, 2004. No. 2 Pp. 12-14
[5] Bykov G.T. Vibrational stability in turning thin-walled pipe by multicutter heads / G.T. Bykov, A.S. Yannikov, O.A. Yannikova, etc.// Russian Engineering Research. 2010. V. 30. No 3. Pp. 296-299.
[6] Kalinski K., Mazur M., Galewski M. High speed milling vibration surveillance with the use of the map of optimal spindle speeds // Proceedings of the 8th International Conference on High Speed Machining, 2010. ENIM, Metz, France. Pp. 300-305.
[7] Wu D.W. Comprehensive Dynamic Cutting Force Model and Its Application to Wave-Removing Processes // Journal of Engineering for Industry. 1989. No 2. Pp. 155-164.
[8] Yannikova O. A Simulation Model of a Non-Rigid Shaft Oscillations under Cutting // STIN Journal, Moscow: 2003, No. 1, pp. 18-21
[9] Zagórski I., Kulisz M., Semeniuk A., Malec A. *Artificial neural network modelling of vibration in the milling of AZ91D alloy*. Advances in Science and Technology // Research Journal. Sep. 2017. Vol. 11, Issue 3. Pp. 261-269. DOI: 10.12913/22998624/76546

[10] Comak A., Budak E. *Modeling dynamics and stability of variable pitch and helix milling tools for development of a design method to maximize chatter stability* // Precision Engineering, 2017, 47, 459-468.

[11] Yamnikov A.S., Chuprikov A.O. *Chucks for Thin-Walled Blanks* // Russian Engineering Research. Vol. 35 No. 11 2015, Pp. 838-840.

[12] Voronov S., Kiselev I. Identification of the Cutting Forces *Coefficients via Milling Process Simulation* // Proceedings of the ASME 2011 International Design Engineering Technical Conference and Computer and Information in Engineering Conference IDETC/CIE, USA. 2011. V 1. P. 127-133.

[13] Yamnikov A., Bogomolov M. *High Vibration Stability Centering Mandrel for the Thin Walled Sleeves* // Ferrous Metals, No. 5, 2019. Pp. 52-57

[14] Bogomolov M., Chuprikov A., Yamnikov A. Mandrel for Fixing Thin-Wall Cylindrical Workpieces // Russian Federation Patent No. 2688019, B23B31/40. Published on May 17, 2019 No. 14.

[15] Zenkevich O. *The Finite-Element Method in Engineering*. Moscow: MIR Publishing. 1975. 541 p. ing the last numbered section of the paper.