Thin-Walled Part Machining Process Parameters Optimization based on Finite-Element Modeling of Workpiece Vibrations

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Abstract

Arising vibrations when milling thin-walled parts are the most essential factor that limits machining productivity and surface location accuracy. Such vibrations have large amplitude and often result in irreversible workpiece damage on finishing stage. Vibrations appear as a result of resonance of workpiece natural frequencies and cutting force spectrum harmonics from tooth-passing impacts.

To eliminate resonance vibrations it is suggested to tune spindle speed in such a way, that workpiece natural frequencies differs from tooth-passing frequency harmonics. In practice changing of natural frequencies due to stock removal and changing of cutter position during machining process need to be considered.

To estimate natural frequencies and mode shapes of workpiece mounted on NC machine worktable and its evolution during the process it is reasonable to apply widespread finite element modeling software. Using of FEM methods for vibration process modeling is considered. Also practical example of machining thin blade of compressor’s aerodynamic model is considered.

Keywords: milling process, workpiece vibrations, thin-walled part, finite-element modelling.

1. Introduction

At present further progress in high-speed machining efficiency is mainly restricted with vibrations of machine-tool-part system, limiting productivity, accuracy and quality of the part. The issue becomes critical while finishing thin-walled parts with tightly restricted accuracy. The trial-and-error method—which is time consuming and expensive—is still widely used to optimise the machining process.

The example of thin-walled parts in TsAGI’s aerodynamic models production department are GTE (Gas-Turbine-Engine) aerodynamic model compressor blades. Typical thickness changes from 5 to 1 mm from root to end section (section length changes from 100 to 50 mm) for wingspan varying from 200 to 300 mm (figure 1). Intensive vibrations arises when semifinishing and finishing machining the part at recommended cutting speed and process parameters (depth of cut, width of cut, feedrate) resulting in sufficient part damage.

The study of the vibration effect started many years ago with the first work by Tobias [1] and Tlusty [2]. This theory was extended to the milling process [3-5]. Proposed mathematical methods and industrial solutions [6], using special software and equipment for measuring spindle-cutter frequency response function and computation of optimal process parameters prove their effectiveness for tool and spindle vibration elimination. In case of thin-wall milling, proposed method of stability lobes cannot be applied directly because of dynamic change during machining. Workpiece natural frequencies are changing during stock removal process, which lowers workpiece mass and stiffness.

To solve the problem it was proposed to study the part machining by small zones. In these zones, the part
could be modeled with constant dynamic properties and with rigid body motion [7]. The stability characteristics change during machining leads to a third dimension on the stability lobe [8]. This approach was also extended by taking into account machine and tool flexibility [9].

Nevertheless workpiece vibrations are still a problem in practical applications. Here we consider simple practically oriented technique based on the approach proposed in [7] to eliminate part vibrations when tool is assumed to have high stiffness, as it is applied in TsAGI’s aerodynamic model production department. An example part will be GTE compressor aerodynamic model blade.

Workpiece vibrations usually rise from resonance between natural frequencies of part under machining and components of cutting force frequency spectrum, defined by tool rotation frequency and cutting tooth impacts. Typical result of part vibrations is shown on figure 2. If workpiece natural frequencies are known one could select tool rotation frequency, preventing resonance on the process planning stage.

The core idea is to calculate workpiece natural frequencies and their changing during machining process by using Finite Element Modeling.

2. Nomenclature

\begin{align*}
F_0 & \quad \text{Average force in Fourier series [N].} \\
F_j & \quad j\text{-th harmonic amplitude in Fourier series [N].} \\
\omega_t & \quad \text{angular cutter rotation frequency [rad/s].} \\
\phi_j & \quad \text{phase shift of } j\text{-th Fourier series harmonic [rad].} \\
j & \quad \text{cutting force spectrum frequency index.} \\
c & \quad \text{linear oscillator damping coefficient, determined mostly by the environment conditions.} \\
k & \quad \text{linear oscillator stiffness coefficient.} \\
m & \quad \text{linear oscillator system mass.} \\
\omega & \quad \text{oscillator natural frequency.} \\
F(t) & \quad \text{external force acting on the system.} \\
f_t & \quad \text{tooth passing frequency.} \\
f_i & \quad \text{workpiece natural frequency.} \\
i & \quad \text{workpiece natural frequency index.} \\
\mu & \quad \text{allowance coefficient.} \\
Z & \quad \text{number of cutter teeth.} \\
n & \quad \text{spindle speed.}
\end{align*}

Fig. 1. GTE compressor aerodynamic model blade.

Fig. 2. Influence of vibrations on surface quality after milling: machined part and vibration form.

3. Cutter Impact Influence on the Part to be Machined

Cutting force profile function is calculated using mechanistic model [2] (Figure 3a). Frequency spectrum is obtained by Fourier series expansion (Figure 3b):

\[
F(t) = \frac{F_i}{2} + \sum_{j=1}^{\infty} F_j \cdot \sin(\omega_t \cdot t + \phi_j) \\
\]

Harmonics (1) interact with workpiece frequency spectrum. Close or match values of cutting force harmonics and one or several workpiece natural frequencies result in vibrations rising.
Computation of Workpiece Natural Frequencies and Vibration Forms and Experimental Verification

Computation of workpiece natural frequencies and vibration forms was carried out using industrial FEM software NASTRAN and CATIA-Analysis.

Basis idea is to use the simplest FEM algorithms to reduce engineer qualification demands.

So tet-element mesh in CATIA-Analysis and hex-element mesh in NASTRAN were automatically generated according to geometrical part model using one end fixing conditions.

To validate FEM results natural frequencies were experimentally measured. In the experiment vibrations were excited using impact hammer. Vibration frequencies were measured by miniature accelerometers (m ≤ 0.4 g). Mounting position for the accelerometer were in vibration form loop as it given by FEM software (figure 4).

Computation (using both software packages) and experimental results for the blade modes 1-th to 10-th combined in table 1. Column 5 shows difference between computation and experiment.

<table>
<thead>
<tr>
<th>Form #</th>
<th>Natural frequencies [Hz]</th>
<th>Difference of experimental measurements and CATIA-Analysis results, %</th>
</tr>
</thead>
<tbody>
<tr>
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<td>Experimental measurement</td>
<td>Computed CATIA-Analysis</td>
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<tr>
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<td>118.5</td>
<td>122.4</td>
</tr>
<tr>
<td>2</td>
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</table>

One can see measured experimental frequencies are very close to frequency values obtained using FEM. Maximum difference is about 5-6%.

Resonance Elimination Condition

The blade performs oscillations in linear elasticity zone of load-extension diagram. In a certain point of blade part its oscillations could be simplified to linear oscillator motion (with some allowance). Let us consider forced oscillations result from external force action (cutting force acting on the part):

\[
\ddot{x} + 2 \cdot s \cdot \dot{x} + \omega^2 x = F(t) \tag{2}
\]

\[
s = \frac{c}{2m}, \quad \omega^2 = \frac{k}{m}
\]

Substituting expression (1) in (2), we obtain:

\[
\ddot{x} + 2 \cdot s \cdot \dot{x} + \omega^2 x = F_0/2 + \sum_{j=1}^{m} F_j \cdot \sin(j \cdot \omega \cdot t + \varphi_j) \tag{3}
\]
For the linear oscillatory circuit vibrational process could be presented as sum of elementary oscillation processes. So let us write down equation for j-th oscillatory circuit in sum (3), consider phase \( \varphi_j = 0 \):

\[
\ddot{x} + 2 \cdot s \cdot \dot{x} + \omega^2 \cdot x = F_j \cdot \sin(j \cdot \omega_t \cdot t)
\] (4)

Varying starting time moment one could set zero phase \( \varphi_j = 0 \).

Differential equation (4) solution is the sum of decayed free oscillations and forced oscillations, excited from cyclic force action [3].

Forced oscillations amplitude could be written as:

\[
A = F_j / \sqrt{\left[ \omega^2 - (\omega_t \cdot j)^2 \right]^2 + 4 \cdot s^2 (\omega_t \cdot j)^2}
\] (5)

It can be seen from (5), vibration amplitude achieves its maximum value when natural frequency and frequency of acting force are equal. So that resonance condition would written as follows:

\[
\omega = \omega_t \cdot j
\] (6)

In a similar way for the workpiece fixed on NC machine worktable, there is no resonance when:

\[
f_j \neq f_t \cdot j
\] (7)

Taking into account inaccuracy of FEM computation expression (7) can be written as:

\[
f_i \in [(1 - \mu) \cdot f_t \cdot j, (1 + \mu) \cdot f_t \cdot j]
\] (8)

Allowance coefficient \( \mu \approx 0,06 \) is approximately equal to the sum of maximum relative difference between experimental measurements and computation results from table 1 and half-width of resonance curve.

If workpiece natural frequencies \( f_i \) are known safe (when there is no resonance) tooth-passing frequency and respective spindle speed can be found from equation (8) and the following:

\[
n = f_t \cdot 60 / Z
\] (9)

Thereby, the problem is reduced to determination of reasonable tooth-passing frequency \( f_t \). For any stage of stock removal while machining condition (8) should follow taking into account continuing variation of workpiece natural frequencies.

5. Selecting Cutter Rotation Frequency

Starting cutter rotation frequency is usually defined from tool catalogue to provide optimal cutting speed for given tool and workpiece materials.

To define spindle speed more exactly the following sequence of operations is accomplished. Process planner sets positions for technological mounts added to the part to fix it on worktable (figure 5a).

The blade is divided in a sequence of technological zones, corresponding to stock removal.

The blade under consideration was divided into five zones along the wing midline (\( \Delta L = 0–50 \text{ mm}; 50–100 \text{ mm}; 100–168 \text{ mm}; 168–190 \text{ mm}; 190–209 \text{ mm} \)). For each zone geometrical model and FEM mesh with appropriate boundary conditions were created (figure 5b). Computation of interesting vibration forms is accomplished (five forms were considered for the blade). The number of natural vibration modes is defined by preliminary amplitude analysis. For higher modes vibration amplitude is relatively small and do not influence machined surface quality.

Fig. 5. (a) geometrical workpiece model including technological added mounts. Green shows machined surface, red – surface before machining (stock); (b) FEM model mesh and computed natural vibration forms

Table 2 contains natural frequencies computation results for examined vibration forms.
Table 2. Changing of blade workpiece natural frequencies due to stock removal

<table>
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<th>Reference points, L [mm]</th>
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<th>Natural frequency</th>
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</tr>
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<td>553</td>
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<td>604</td>
</tr>
<tr>
<td></td>
<td>209</td>
<td>584</td>
</tr>
</tbody>
</table>

It appears vibration form visualization is useful for general oscillation analysis and workpiece fixation effectiveness. Particularly, it can be seen vibrations have flexural-and-torsional form (figure 5b). Amplitude values allow us to use two point part fixation scheme, as showed on figure 5. The figure illustrates stock removal process modeling using consecutive zones when machined surface (L – characteristic length along the wing) becomes larger while stock shortens.

Natural frequency values for modes and their changing while removing stock (along the wingspan) showed on figure 6. Green lines denote cutting force spectrum harmonic amplitudes (the lowest corresponds to the tooth-passing frequency).

Fig. 6. Blade part resonance frequencies in machining process (L – length of machined zone, figure 5).

Two of them cross blade natural frequency curves at the distance of 150 mm from its right end (figure 6). Obtained corruption of machined surface and break-up of blade edge resulting from appearing vibrations can be seen on figure 2a. To eliminate resonance vibrations it is reasonable to alter spindle rotation frequency that results in corresponding exciting force spectrum modification.

6. Conclusions

For the considered blade part performed spindle speed correction allowed sufficiently (more than 10 times) to lower vibration amplitudes and provided achieving necessary surface roughness.

This confirms efficiency of the proposed practical approach. Since 2008 to 2012 it was applied in machining of 14 variants of blades of different geometry (wing span from 250 to 350 mm).

Using of CATIA-Analysis and NASTRAN is laborious in practical applications. When preparing computational mesh, there are a lot of standard geometrical and node modification operations that could be done automatically. Special FEM software intended for machining part vibrations computation could simplify applying of the proposed approach in manufacturing industry.

Acknowledgements

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References