Dynamic stability of grinding machines – potentials and risks

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Products and Services:
Measurement
- Displacement / Cutting forces
- Vibrations / Stiffness
- Modal analysis / Frequency response
- Geometry / Dyn. spindle concentricity
- Analysis of floor and base
- Long-term monitoring of machine tools

Simulation
- Simulation of structural mechanics
- Mechatronic simulations
- Simulation of machining
- Base analysis
- Tool calculation

Design & development
- Vibration exciter and damper
- Measurement systems
- Measurement and calculation software
Structure

- Dynamic stability and vibrations during grinding
- Risks of dynamics in design of grinding machines
- Possibilities for detecting causes of vibration
- Potentials for avoidance of dynamical weaknesses
- Summary
Chatter vibration – a permanent risk in fine finishing

Characteristic of Chatter
- Self-energizing vibration
- Caused by a resonance point of the machine tool/workpiece/clamping device
- Vibration frequency depends on properties of mass and stiffness without any rotation speed dependency
- Appearance slightly above the dominant resonance of the machine tool
- Short wavelength partly visible as surface waves, long wavelength only measurable in the majority of cases
Self-excited vibration – the regenerative effect on workpiece side

The mechanism of chatter
- **T₁**: Outer disturbance forces cause a relative displacement between workpiece and grinding wheel.
- **T₂**: The resulting vibration fades away because of damping effects.
- **T₃**: After one revolution of the workpiece the waviness causes a change of the cutting forces.
- **T₄**: The deviation of the cutting forces results in a new and increased shifting which intensifies the forming of waves.

**Note:**
- A similar mechanism is also possible on grinding wheel side during grinding and dressing!
Flexibility areas – amplifying of the residual imbalance of the grinding wheel

Characteristic of vibration
- The rotary frequency of the grinding wheel is located in an area where the machine tool has a higher flexibility.
- The increasing of the vibration level is not proportional to the rising rotation speed (resonance crossing).
- No self-energizing or swing up.
- The wave numbers on the grinded workpiece can always be calculated back to the rotary frequency of the grinding wheel respectively its multiples.

Related problems:
- Installation vibrations caused by external excitation transferred via the floor.
- Vibration caused by internal excitation from aggregates.
Consequence of dynamic weak points of grinding machines

Consequence
- Visible and/or measurable waviness on grinded surfaces.
- Abrasion/wave forming on the grinding wheel.
- Tolerance problems in FFT order analysis.
- Reduced life period of spindles, bearings, guidance.

Questions
- **Forecast:** Is the risk of vibration calculable?
- **Classification:** Is it possible to classify vibration problems, e.g. point of origin?
- **Statistics:** Which kind of vibration problems frequently appear depending on the type of machine tool?
Dynamic stability and vibrations during grinding

Risks of dynamics in design of grinding machines

Possibilities for detecting vibration causes

Potentials for avoidance of dynamical weaknesses

Summary
Vibration problems are complex and appear in various ways – but can be classified by the point of origin

**Dressing unit:**
- Tilting/sliding of the dressing slides / housings
- Bending/sliding/torsion of the dressing spindle
- Self-deformation of the dressing tool

**Workpiece unit:**
- Tilting/sliding of slides/housings/head stock
- Bending/sliding/torsion of the spindle/chuck/workpiece
- Self-deformation of the workpiece

**Basic machine:**
- Tilting/weaving of columns
- Installation vibration of the machine bed

**Grinding unit:**
- Tilting/sliding of slides/housing/head stock
- Bending/sliding/torsion of spindle/flange/grinding wheel
- Plate oscillation of grinding wheel/tool

**Other causes:**
- Unbalanced excitation
- Geometrical runout
- Pitch error of ball screw
- Damages of bearings
- Belt vibrations
- Pump pulsation
- …
Vibration problems appear in all types of grinding machines – but the causes are unevenly distributed.

**Percentaged distribution of chatter causes:**

- **Other (unbalancing/runout)**: 19.8%
- **Vibration of the dressing unit**: 9.3%
- **Vibration of the workpiece unit**: 17.4%
- **Vibration of the grinding unit**: 46.5%
- **Vibration of the basic machine**: 7%

**Statistics:** Metrological problem analyses of 62 grinding machines of different sizes and types:

- 16 External cylindrical grinding machines (between centers)
- 4 Internal cylindrical grinding machines
- 11 Portal or column grinding machines (Large designs)
- 9 Surface or profile grinding machine (with single- or double-spindle)
- 12 Centerless grinding machines
- 10 Other grinding machines (gear / blade grinding etc.)

**Note:** Vibration problems with multiple causes were also classified in more than one field.
Overlayed vibration problems can appear during grinding and dressing - but in common cases there is only one problem during grinding

**Statistics:** Metrological problem analyses of 62 grinding machines of different sizes and types:

**Chatter vibration appears:**
- during grinding: 77.6%
- during dressing: 22.4%

**Chatter vibration is caused by:**
- three sources: 3.3%
- two sources: 34.4%
- one source: 62.3%

**Conclusion:**
- Because of higher cutting forces, the chatter vibration appeared during grinding in about 78% of considered cases.
- In about 38% of considered cases, the work piece deviance is caused by more than one dynamic weak point.
General risks of vibration of a grinding unit

Statistics: Evaluation of 42 grinding machines, where grinding units caused the chatter vibration:

- Sliding/tilting of the grinding unit: 40.4%
- Bending of the spindle-bearing-system: 45.2%
- Tilting of spindle and flange/grinding wheel: 9.5%
- Plate oscillation of the grinding wheel: 4.9%

percentage of vibration caused by grinding unit [%]
**General risks of vibration of a workpiece unit**

<table>
<thead>
<tr>
<th>Sliding/tilting of the workpiece unit</th>
<th>Bending of the whole unit (spindle/chuck/workpiece and sleeve in some cases)</th>
<th>Self-deformation of the workpiece</th>
</tr>
</thead>
</table>

**Statistics:** Evaluation of 14 grinding machines, where workpiece units caused the chatter vibration:

- Sliding/tilting of the workpiece unit: 14.3%
- Bending of spindle/chuck/workpiece: 71.4%
- Self-deformation of the workpiece: 14.3%
### General risks of vibration of a dressing unit

<table>
<thead>
<tr>
<th>Sliding/tilting of the dressing unit</th>
<th>Bending of spindle-bearing-system</th>
<th>Self-deformation of the dressing tool (dressing roll / plate / diamond etc.)</th>
</tr>
</thead>
</table>

### Statistics: Evaluation of 8 grinding machines, where dressing units caused the chatter vibrations:

- Sliding/tilting of the dressing unit: 0%
- Bending of spindle-bearing-system: 50%
- Self-deformation of the dressing tool: 50%

Percentage of vibration caused by dressing unit [%]
Design-specific risks: cylindrical grinding machines, grinding machines for crankshafts and camshafts

Overswining, caused by differences in mass and inertia between tool and workpiece side, is caused by:
- Fast change in direction of the slides due to the cam shape
- High mounting above the machine bed
- Similar natural frequencies of the tool unit and workpiece unit

Natural vibration of a clamped crankshaft is problematic at:
- Crankshafts with a high stroke (weak profile)
- Clamping concepts without distortion (joints in chuck)
- First grinding without stationary support

A waviness at bearing position A can be transferred to bearing position B via contact of a stationary support.
Design specific risks: Portal and vertical grinding machines

The weaving of the slide/grinding head is caused by:
- Wide throat depth of the slide
- Heavy tool heads / grinding heads
- Small/weak profile of the slide

Statistics: Metrological problem analysis of 42 large machine tools in portal-, gantry- or floor stooded column layout with vertical slide

- Tilting of column/portal: 9,6%
- Natural vibration of workpiece/table: 21,4%
- Weaving of the slide: 50%
- Other causes: 19%

Percentages of vibration causes [\%]
Dynamic stability and vibrations during grinding

Risks of dynamics in design of grinding machines

Possibilities for detecting vibration causes

Potentials for avoidance of dynamical weaknesses

Summary
Metrological process and machine analysis: systematic detection of dynamic weak points

Analysis of machined surface

Complete examination:
Measurement on site: 1 - 2 Days
Pre-evaluation: instantly
Profitability report: 2 – 5 Days
Total: 4 – 7 Days

Of these:
Machine downtime: 1 – 2 Days
Results after: 4 – 7 Days

Expenditure of time

Vibration analysis of processes

Stiffness analysis of machine tool

Evaluation of enhancement measures:
- A: Process changing
- B: Component stiffening
- C: Damping of machine structure
- D: ...

Weakness analysis / measures

Complete, detailed modal analysis
Shaker: Important measurement system for acquisition of the „dynamical fingerprint“ of a machine structure

**Vibration exciter (Shaker):**
- Easy relative fixation between grinding wheel and workpiece side
- Relative excitation of the machine structure close to the process with a static preload
- Flexible choice of the excitation spectrum

**Application:**
- Reproducible measuring of the quasi-static and dynamic flexibility behavior of machines
- Separation of the flexibility-shares of grinding wheel and workpiece side
- Continuous excitation and force acquisition for modal analysis
Direct time domain analysis (TDA):

- Non-linear and non-stationary processes (swing up, timed changes of vibration direction) can not be analysed with methods of the frequency domain (operational vibration / modal analysis).
- The measured vibration accelerations are integrated two-fold and displayed as vibration displacement.
- Generally, a simultaneous measurement with multiple triaxial sensors is necessary.
Evaluation of vibration processes starts with the grinding results: Proper estimation of FFT-order-analysis

Challenges:

- Fine wavinesses often have amplitudes within the scope of roughness.
- Low tolerance limits are often exceeded by a multiple of the wave order of the actual problem.
- The speed of wheel rotation necessary to differentiate chatter problems from imbalances/bearings of the grinding is often not documented.
- Frequencies the workpiece rotation speed needs to be documented.
- In order to estimate the exciting of waviness shares the future installation has to be considered (e.g. the amount of rolling elements of a bearing)

**Tolerance curves**

Evaluation of vibration processes starts with the grinding results: Proper estimation of FFT-order-analysis

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Calculational concept- and machine analysis: Early detection and avoidance of dynamical weak points

Example of a surface grinding machine:
- Working space (X/Y/Z): 800/2000/600 mm³
- X-Axes
  - Guidance: 2x2 INA RUE 35 D
  - Ball screw: Steinmeyer 10.40.7,5.4
  - Fixed bearing: INA ZKLF 3080
  - Engine: Siemens 1FT6086-1AF71-2AH1
- Y/-Z-Axis
  - Guidance: INA RUE 45 D (Y = 2x3, Z = 2x2)
  - Ball screw: Steinmeyer 20.50.9.4
  - Fixed bearing: INA ZKLF 3590
  - Engine: Siemens 1FT6108-8AF71-4PG4
- Main spindle
  - Front: 4x spindle bearing FAG B7020C.UL
  - Rear: Cylinder roller bearing FAG NN3018.ASK
  - Engine: Siemens 1PM6138-2VF81-1BR1
- Materials
  - Machine bed: Epument 145B
  - Column, X-slide, table slide: GG25
  - All other parts: Steel
- Machine installation: 12x Isoloc UMS5-ASF/30

Machine simulation:
- FE-modeling and flexible multiple-body-simulation of the machine
- Consideration of all components of structure, spindles, guideways, bearings and the machine installation
- Linking of the drive control

Calculation possibilities:
- Static stiffness
- Frequency response functions
- Modal analysis / natural frequencies
- Consideration of the base
- Wall thickness optimization
- Rating of tuned-mass dampers

Not very time-consuming:
Time for calculation of an entire machine takes: about 5 – 7 days
Static stiffness in X-direction:

- $k_{XX,\text{REL}} = 17.9 \text{ N/\mu m}$
- $k_{XX,\text{TOOL}} = 18.1 \text{ N/\mu m}$
- $k_{XX,\text{WP}} = 2117.9 \text{ N/\mu m}$

Shares of the individual components in the overall deformation:

- Ball screws: 14.9%
- Column: 15.6%
- Guideways: 12.5%
- Grinding spindle: 10.8%
- Table slide: 0.5%
- X-slide: 43.6%

X-slide, the column and the X-ball screw have the largest shares of the deformation.
Optimization of component weight and stiffness by wall thickness optimization

Starting points of the optimization:

Objective:
- Decrease in weight of component but with equal stiffness
- Increase of the stiffness with equal weight of component

Example machine column:
- Through a weight-neutral optimization of the column’s wall thickness, the stiffness of the surface grinding machine can be increased in X- and Y-direction by about 3 to 4%.
- With a deformation share of 16 - 21% (X- and Y-direction) this represents a quite a good improvement.
- Additional (minor) improvements require considerably more material (cmp. variation 2: +285 kg).
Optimization of the static machine stiffnesses

Objective:
- Systematic increase of static stiffness of components with largest shares in the overall deformation

Example X-Slide:
- **Variation 1**: The enhancement of the base plate by about 40 mm with a simultaneous change of material from GG25 to GGG40 causes a considerable gain of stiffnesses of between 4% (Z) and 33% (X).
- **Variation 2**: The increase of the square bar dimensions to 220 mm appears to be a good compromise between disturbance outline and stiffness. It increases the stiffness in grinding-sensitive direction (X) by about another 12%.

### Relative static stiffnesses

<table>
<thead>
<tr>
<th>Variation</th>
<th>$k_{XX,REL}$</th>
<th>$k_{YY,REL}$</th>
<th>$k_{ZZ,REL}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Variation 0</td>
<td>17.9</td>
<td>17.6</td>
<td>18.1-18.8-18.9</td>
</tr>
<tr>
<td>Variation 1</td>
<td>26.6</td>
<td>22.2</td>
<td>18.8</td>
</tr>
<tr>
<td>Variation 2</td>
<td>30</td>
<td>25.3</td>
<td>18.9</td>
</tr>
</tbody>
</table>

**Graph:**
- **$k_{XX,REL}$**: Variation 0 vs. Variation 1 vs. Variation 2
- **$k_{YY,REL}$**: Variation 0 vs. Variation 1 vs. Variation 2
- **$k_{ZZ,REL}$**: Variation 0 vs. Variation 1 vs. Variation 2

**Legend:**
- Black: Variation 0
- Blue: Variation 1
- Red: Variation 2
Optimization of the dynamic machine rigidity

Damping of the X-slide:
- The square bar also executes a nodding vibration in X-direction because of the increased mass in the statically enhanced variation 2.
- Variation 3: With a tuned mass damper at the square bar the maximum dynamic flexibility can be reduced by about 70% compared to variation 2.
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- The quality of a workpiece can be reduced by a multitude of different vibration influences.

- However, not all vibration problems appear equally often. Certain machine types are, depending on the design, more or less prone to chatter vibrations, whereas the cause is often found in the same assembly group.

- Nevertheless, well-tried measurement strategies for vibration and stiffness acquisition allow for a quick and systematic cause analysis.

- State-of-the-art simulation methods help in early detection and avoidance of dynamical weaknesses.
Thank you for your attention!

If you are interested in this topic:

You can see an animated 3D-view of the presented machine simulations and modal analysis with our free software, planlauf/VIEW,

You can download the software and this presentation at:

www.planlauf.com