Noise Evaluation Process for Gearboxes

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Outline

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Motivation

Gearing Data → gear engagement forces → noise level

MBS simulation in time domain is substituted by MNOISE gear module

Simulation can be done by FE expert

FE frequency response analysis for several rpm steps
Simulation - process

MBS simulation in time domain is substituted by MNOISE gear module

→ Simulation can be done by FE expert

Gearing Data

gear engagement forces

Finite Element Run-up Simulation

MNOISE

FE frequency response analysis for several rpm steps

noise level
Simulation model

Model build up:

- spur gear pair
- shafts
- bearings
- additional spring-mass system at in- and output (representation of additional power train components located at in- and output)

Assumptions / simplifications:
- Gears are considered as rigid body
- bending of shafts is described by deflection curve (elastic deformation)
- bearings are represented by spring-damper system (stiffness-damping value)
- gear tooth contact is represented by “mesh stiffness curve”

Keep in mind: The “exact” representation of the contact is very hard, because this is a complex physical phenomenon (micro plasticity, lubrication, friction, …).

⇒ A simplified phenomenological approximation by a “mesh stiffness” description is chosen
Simulation model – governing equations

**Equilibrium of forces:**
\[
\bar{M}\ddot{x} + \bar{C}\dot{x} + \bar{K}x = F
\]

**Rotational DOFs** (related to center of mass):
\[
\varphi = (\varphi_{1,x} \quad \varphi_{1,y} \quad \varphi_{1,z} \quad \varphi_{2,x} \quad \varphi_{2,y} \quad \varphi_{2,z} \quad \varphi_{in} \quad \varphi_{out})
\]

**Mass matrix** \(\bar{M}\)  \(\ldots\) \(\text{mass-matrix}\)

**Damping matrix** \(\bar{C}\)  \(\ldots\) \(\text{damping-matrix}\)

**Stiffness matrix** \(\bar{K}\)  \(\ldots\) \(\text{stiffness-matrix}\)

**Mass moment of inertia matrix** \(\bar{I}\)  \(\ldots\) \(\text{mass moment of inertia-matrix}\)

**Rotational damping matrix** \(\bar{C}_\varphi\)  \(\ldots\) \(\text{rotational damping-matrix}\)

**Rotational stiffness matrix** \(\bar{K}_\varphi\)  \(\ldots\) \(\text{rotational stiffness-matrix}\)

**Moment equilibrium:**
\[
\bar{I}\ddot{\varphi} + \bar{C}_\varphi \dot{\varphi} + \bar{K}_\varphi \varphi = T
\]
Simulation model – governing equations

Equilibrium of forces:
\[ \overline{M} \ddot{x} + \overline{C} \dot{x} + \overline{K} x = F + F_{coup} \]

Moment equilibrium:
\[ \overline{I} \ddot{\varphi} + \overline{C}_\varphi \dot{\varphi} + \overline{K}_\varphi \varphi = T + T_{coup} \]

Gear tooth contact is simplified by mesh stiffness curve

DOFs:
\[ x = (x_{1,x} \quad x_{1,y} \quad x_{1,z} \quad x_{2,x} \quad x_{2,y} \quad x_{2,z}) \]
\[ \varphi = (\varphi_{1,x} \quad \varphi_{1,y} \quad \varphi_{1,z} \quad \varphi_{2,x} \quad \varphi_{2,y} \quad \varphi_{2,z} \quad \varphi_{in} \quad \varphi_{out}) \]

Separation of linear- and nonlinear expressions leads to a semi-implicit numerical solution scheme. ➔ Implicit solution for linear parts, explicit solution for nonlinear parts ➔ fast and stable
Simulation model - 14 DOF governing equations

Handled effects:
• Crowning (included in mesh stiffness curve)
• Inclination of shafts (geometrically included in governing equations, stiffness curve)
• Bending of shafts (analytical deflection curve)
• Stiffness of bearings (linearized stiffness at operating point)
• Displacement-, Velocity- and Acceleration Boundary conditions represented by Lagrange Formulation → numerical stability
• Infinite stiffness behavior possible → rigid bodies in case of missing data
• Pitch fluctuation (360°stiffness curve)
• Addendum modification (LVR – output → parameter modification in governing equations)
• Input-Output substitute stiffness/mass
• …
Simulation model – boundary conditions

Boundary condition:
1. \( \varphi \) angle
2. \( \dot{\varphi} = \omega = 2\pi \cdot \frac{n_{out}}{60} \)
3. \( \ddot{\varphi} = \dot{\omega} \)

Run-up:

\[
\frac{d\varphi(t)}{dt} = \omega_1 + \frac{\omega_2 - \omega_1}{T} \cdot t
\]

\[
\varphi(t) = \omega_1 \cdot t + \frac{\omega_2 - \omega_1}{T} \cdot \frac{t^2}{2}
\]
Lagrange multipliers are used for coupling and deactivation of DOFs

- combinations of different DOFs can be deactivated by use of Lagrange multipliers
  - deactivation of bearings/supports (due to missing data, axial, radial)
  - inclination angle (simplification- missing data)
  - input/output substitute stiffness/mass

- Lagrange multipliers can also be used for extension to multiple gear stages

- Boundary condition (rotational speed) is applied with Lagrange multipliers
Simulation model - numerical schema

\[
\begin{pmatrix}
\dot{x} \\
\dot{y} \\
\dot{\lambda}
\end{pmatrix}
= \begin{pmatrix}
\ddot{x} \\
\ddot{y} \\
\ddot{\lambda}
\end{pmatrix}
= \begin{pmatrix}
0 \\
F + F_{coup} \\
b
\end{pmatrix}
\]

+ Boundary conditions are strictly fulfilled with Lagrange multipliers (compared to penalty method)

+ Lagrange multiplier can directly be interpreted as cutting force.

+ simple coupling and extension of the equation system to multi stage design

- Lagrange multiplier increase the number of unknowns.

- Lagrange multiplier technique can course to an over constrained system (compared to penalty method)

3 - point - difference scheme (2\textsuperscript{nd} order accuracy):

\[
\frac{\partial}{\partial t}Q = \frac{3 \cdot Q^{n+1} - 4 \cdot Q^n + Q^{n-1}}{2 \cdot \Delta t} + o((\Delta t)^2)
\]

Reduced numeric damping!
Simulation model – mesh stiffness

- Characteristics – meshing ratio
  - Stiffness characteristics of SIMPACK again shows the nominal meshing ratio
  - Similar characteristic with LVR (on unloaded gear pair)
  - Considering the nominal load of 190Nm tangential engagement force, KissSoft and LVR obtain an real overlap ratio of approximately 1.6

- Magnitude
  - KissSoft obtains a considerably higher magnitude than LVR and SIMPACK
  - The lower magnitude of the unloaded LVR curve is due to dominating local tooth contact influences → nonlinear support (Hertzian pressure/stress)
• Good compliance of the tooth force characteristics. FEM in blue (Abaqus) and MBS in red (SIMPACK)

• The spikes in Abaqus results are mainly caused by surface-to-node contact method. An additional influence has the (rough) mesh resolution.

• The results of SIMPACK show the nominal overlap ratio, because elastic tooth deformation is not considered.

• Abaqus involves the elastic tooth deformation, so there is an increase in the overlap ratio of about 12%.
Simulation model – mesh stiffness

- Quasi-static roll off simulations show good conformity with SIMPACK and MNOISE
- The differences in the roll off simulation reflect the differences in the stiffness curves mentioned before
  - MNOISE uses the stiffness curve of LVR
- MNOISE and SIMPACK obtain feasible input data for acoustic evaluation, the Abaqus results are too noisy
Example – mesh stiffness curve

low load

high load

“exact” method

“simplified” method
Simulation model – mesh stiffness

Roll-off simulation:

- exact tooth geometry necessary (micro and macro)
- fine FE mesh (contact problem)
- complex stiffness curve extraction (polygon effect → smoothing, definition of reference for stiffness extraction e.g. root circle)
- expansive static simulation (roll-off simulation in several discrete steps)
The FE simulation is performed in the frequency domain.

Fourier-transform:\n\[
X(\omega) = \mathcal{F}(x(t)) = \int_{-\infty}^{\infty} x(t)e^{-i\omega t} \, dt
\]

Properties of the Fourier-transform:

• Linearity (Superposition)

• temporal derivative ⇔ Multiplication with frequency

• convolution of time dependent functions ⇔ product of frequency dependent functions

• (linear) differential equation ⇔ algebraic equation
FE run-up simulation - Linear time-invariant (LTI) systems

\[ u(t) = \int_{-\infty}^{+\infty} h(t - \tau) f(\tau) \, d\tau \]

Output signal = Superposition of time-shifted impulse response functions

\[ u(t) = h(t) \ast f(t) \]

\[ h(t) \text{...Impulse response} \]

(System property)

\[ \mathcal{U}(\omega) = \mathcal{H}(\omega) \mathcal{F}(\omega) \]

Fourier transform:

Convolution in time domain ⇔
Product in frequency domain

Advantage:

- high stability
- no transient start-up effect
- efficient numerical solution technique
FE run-up simulation - MNOISE

Gear stage run-up simulation with MNOISE gear
- loads obtained in time domain
- FFT
- loads applied in frequency domain

MNOISE software

load spectrum for rpm steps

acoustic results for full rpm range

FE model with surface definition

FE-Solver

automated call
results

frequency response for one rpm step

Source: http://upload.wikimedia.org/wikipedia/de/1/1b/Stirnradler.JPG
Example: AMPV transfer case

* AMPV is a joint venture of Krauss-Maffei Wegmann and RMMV

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Example: AMPV transfer case
Example: AMPV transfer case

- Input shaft
- Intermediate shaft
- Front axle drive
- Rear axle drive
- High stage
- Low stage
- Differential
Example: AMPV transfer case

Dynamically reduced system. The multi stage spur gear is reduced to a single stage spur gear.
Experimental set up

P1
acceleration sensor

P2
microphone
Campbell diagram – P1

\[ a_{0dB} = \frac{1 \mu m}{s^2} \]

46th order dominant!
≡
gear order
Acceleration 1\textsuperscript{st} gear-tooth order - P1
Acceleration $1^{st}$ gear-tooth order - P1

Sensor P1

46$^{th}$ order frequency
Mode shape

- Axial deflexion of input and intermediate shaft
- No local deflexion of housing

604 Hz
Mode shape

- Axial deflexion of input and intermediate shaft
- No local deflexion of housing

1087 Hz
Mode shape

1087 Hz

- Axial and radial deflexion of input shaft
- No local deflexion of housing
Mode shape

1449 Hz

- Bending deflexion of intermediate shaft
- Bending of output shafts
- No local deflexion of housing
Mode shape

- Inclination of intermediate shaft
- Small local deflexion of housing

2174 Hz
Mode shape

- Bending deflexion of intermediate shaft
- No local deflexion of housing

2174 Hz
Mode shape

- Local deflexion of housing

2537 Hz
Summary and conclusion

• A reliable method for noise prediction of gearboxes has been presented. An integrated MBS solver generates the load data for the FE simulation.

Future extension and completion of the method:

• Friction in tangential contact direction
• Consideration of more gear stages
• Extension to bevel – and hypoid gears (stiffness curve)
• …
Thank you for your kind attention