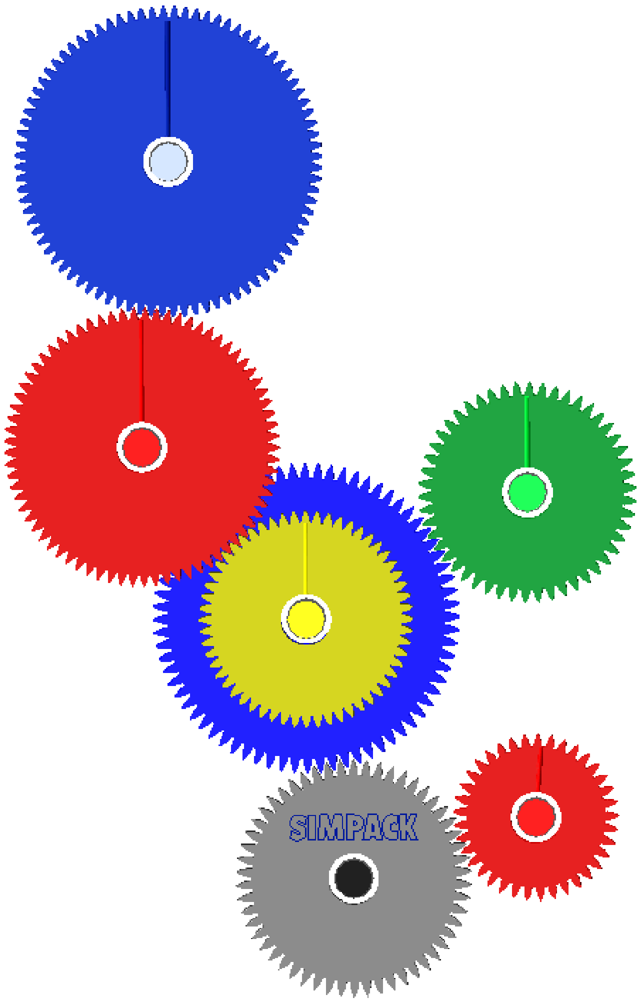


SIMPACK Usermeeting 2006

Baden-Baden

21. – 22. March 2006

The New, Powerful Gearwheel Module

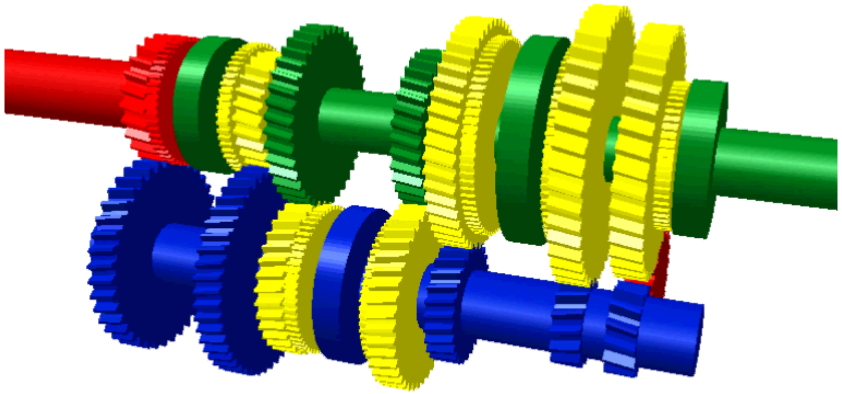
L. Mauer

INTEC GmbH Wessling

The New, Powerful Gearwheel Module

L. Mauer, INTEC GmbH

Outline



Method of Multy Body System Dynamics

Contact modelling for the gearwheel element

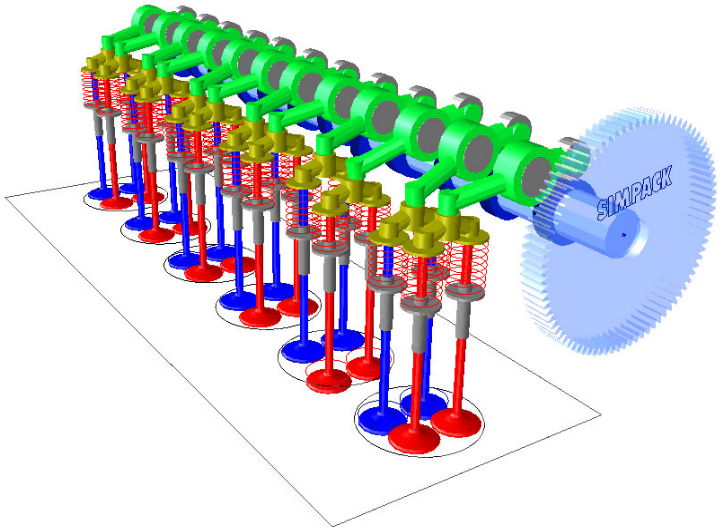
Application examples of powertrain systems

- gear trains in combustion engines

- Drive train with a planetary gears and two spur gear stages in wind energy machines

MBS-Characteristics

Characteristics of Multy Body Systems (MBS)



mechanical system, containing:

- rigid and flexible bodies
- non-linear kinematic Joints
- moved reference systems
- massless force elements with flexibility and/or damping, also with states describing dynamic eigen-behaviour
- closing loop constraints
  - formulation in relative coordinates
  - contact point to curve
  - contact point to surface
  - planar contact curve to curve
  - 3D contact surface to surface
- applied forces depending on constraint forces (friction forces)
- actuators and sensors

$$\dot{\mathbf{p}} = \mathbf{T}(\mathbf{p})\mathbf{v}$$

$$\mathbf{M}(\mathbf{p})\dot{\mathbf{v}} = \mathbf{f}(\mathbf{p},\mathbf{v},\mathbf{c},\mathbf{s},\mathbf{u},\boldsymbol{\lambda}) - \mathbf{G}^T(\mathbf{p},\mathbf{s},\mathbf{u})\boldsymbol{\lambda}$$

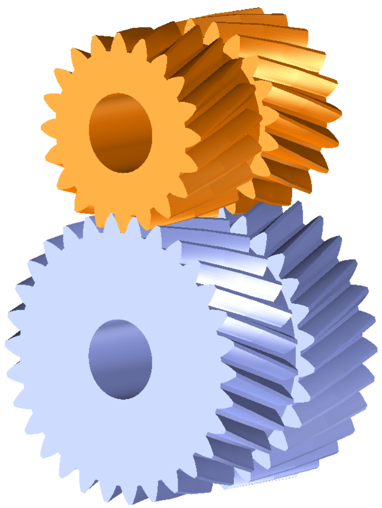
$$\dot{\mathbf{c}} = \mathbf{f}_c(\mathbf{p},\mathbf{v},\mathbf{c},\mathbf{s},\mathbf{u},\boldsymbol{\lambda})$$

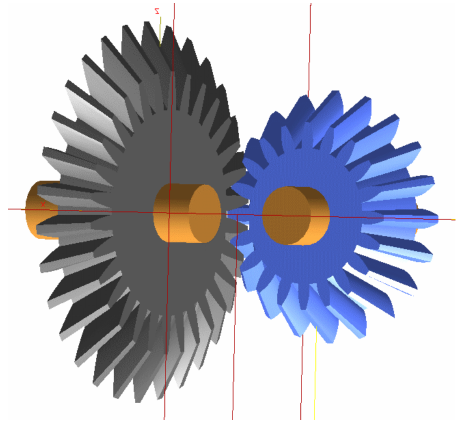
$$\mathbf{0} = \mathbf{g}(\mathbf{p},\mathbf{s},\mathbf{u})$$

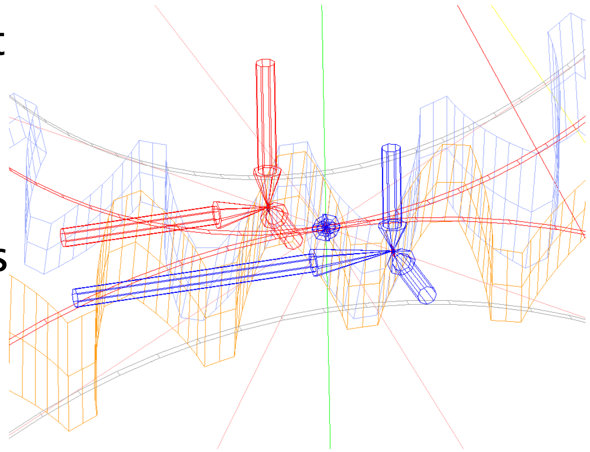
$$\mathbf{G}(\mathbf{p},\mathbf{u}) = \frac{d\mathbf{g}}{d\mathbf{p}}$$

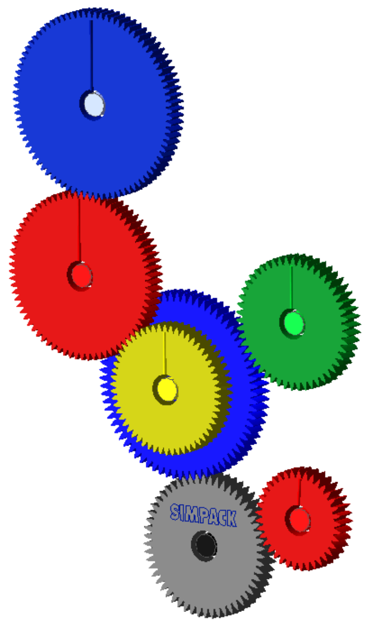
Force Element Gear Wheel

- evolute tooth profile
- spur gears and helical gears
- external and internal - toothing
- profile shift
- profile modification (tip relief)
- backlash
- parabolic function of the single tooth pair contact stiffness
- fluctuation of the total meshing st
- dynamic change in axle distance
- dynamic change in axial direction
- visualisation of the meshing forces components x, y, and z





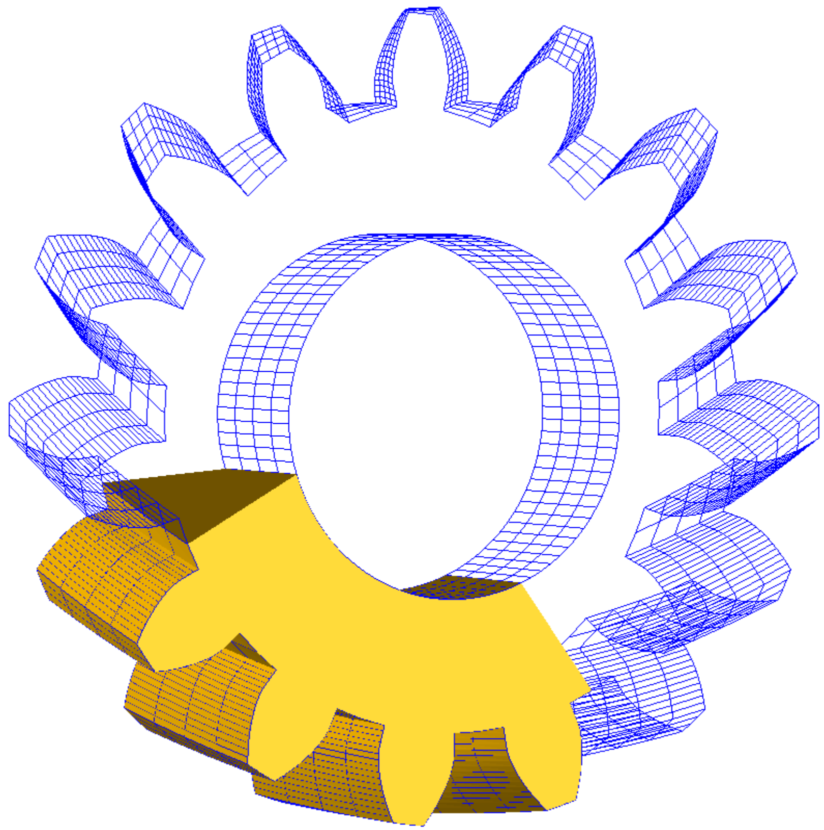




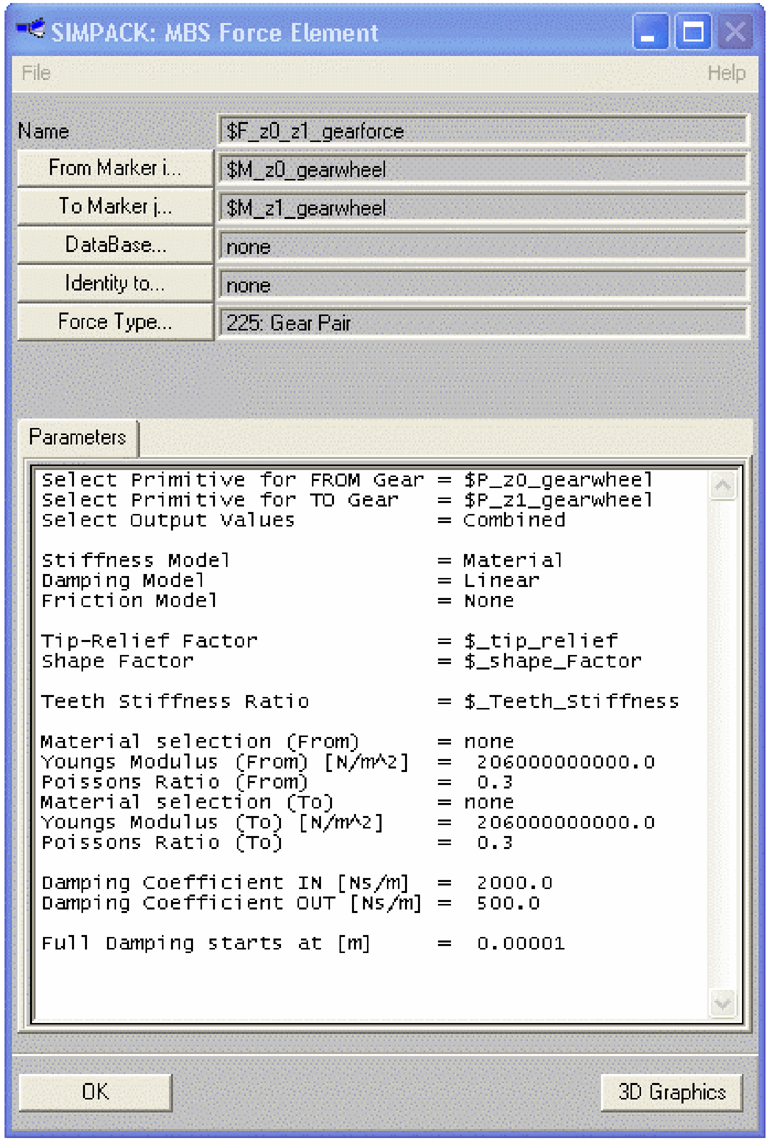
Force Element Gear Wheel

Geometrical input parameters for tooth gear primitives

- flag for setting external or internal gearwheels
- number of teeth
- normal module
- normal angle of attack
- addendum and dedendum height
- helix angle
- bevel angle
- profile shift factor
- backlash or backlash factor
- face width
- discretisation of the graphical representation
- initial rotation angle of the toothing



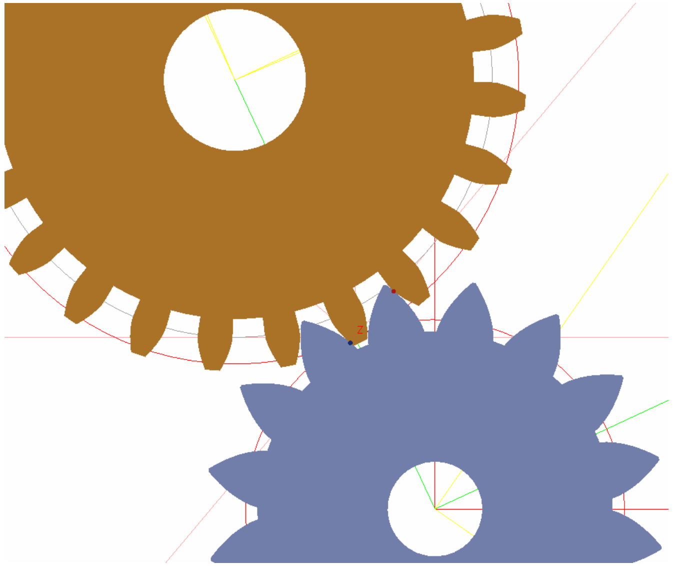
Force Element Gear Wheel

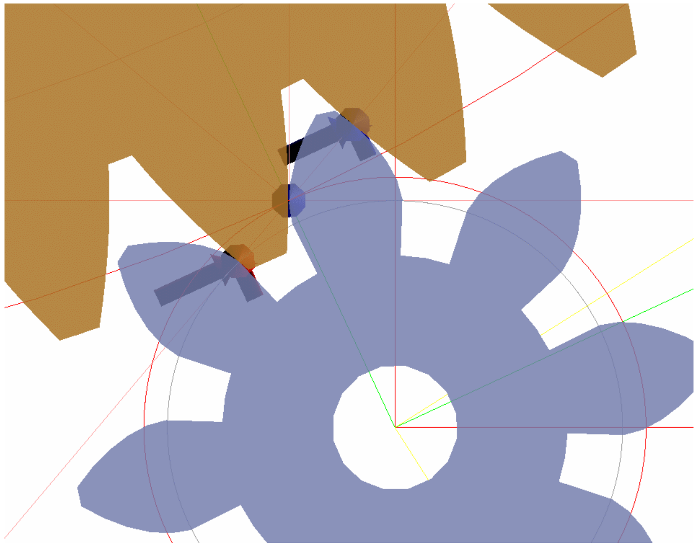


Definition gearwheel force element

- stiffness model
  - linear / non-linear
- damping model
  - linear / non-linear
- friction model
  - non / coulombic
- tip relief factor
- shape factor
- material properties
  - Young modulus, Poisson ratio
- damping parameters

Calculation of the Contact Stiffness





Calculation of the contact stiffness

- calculation of the nominal contact stiffness according to DIN 3990
- parabolic function for the contact stiffness Parameter: *Stiffness Ratio*
- super positioning of the tooth pairing forces considering *Tip Relief*
- flank backlash is depending on the actual centre distance
- if the actual backlash becomes negative, double sided flank contact will be considered

Calculation of the theoretical contact stiffness of a single tooth pair in accordance to DIN 3990

$$q' = C_1 + C_2 / z_{n1} + C_3 / z_{n2} + C_4 x_1 + C_5 x_1 / z_{n1} + C_6 x_2 + C_7 x_2 / z_{n2} + C_8 x_1^2 + C_9 x_2^2$$

z_{n1} number of teeth gear 1

z_{n2} number of teeth gear 2

x_1 profile shift factor gear 1

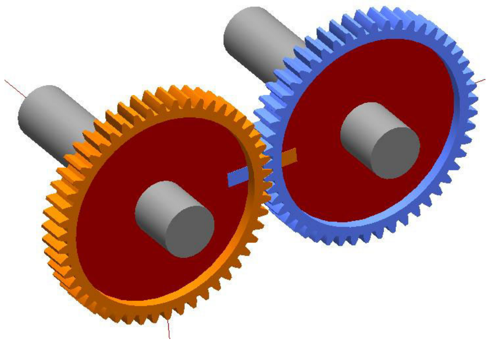
x_2 profile shift factor gear 2

$$z_{n1} \approx \frac{z_1}{\cos^3 \beta}$$

C_1	C_2	C_3	C_4	C_5	C_6	C_7	C_8	C_9
0.04723	0.15551	0.25791	-0.00635	-0.11654	-0.00193	-0.24188	0.00529	0.00182

$$c'_{th} = \frac{1}{q'}$$ theoretical tooth pairing stiffness [N/(mm µm)]

Calculation of the nominal contact stiffness
for the single toot pairing in accordance to DIN 3990



$$c' = c_{th} \cdot C_M \cdot C_R \cdot C_B \cdot \cos\beta$$

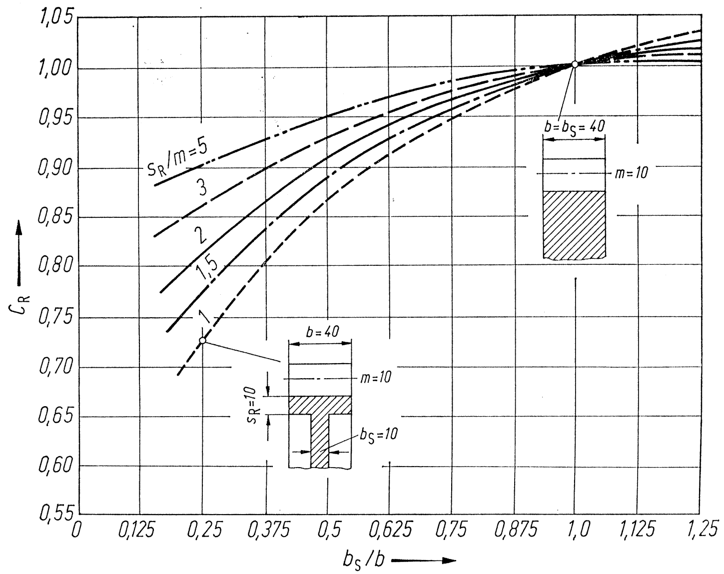
c_{th}	theoretical contact stiffness [N/(mm µm)]		
C_M	correction factor [-]	standard value:	$C_M = 0.8$
C_R	shape factor [-]	for solid gears:	$C_R = 1.0$
C_B	reference profile factor against norm reference profile [-]		
β	helix angle		

standard value for the nominal contact stiffness
(Niemann/Winter, Maschinenelemente II)

$$c' = 14 \qquad [\mathrm{N/(mm\,\mu m)}]$$

Gearwheel shape factor C_R

Source:
Niemann/Winter: Maschinenelemente



Reference profile factor C_B

$$C_B = \left\{1 + 0.5 \cdot \left(1.2 - h_f^* / m_n\right)\right\} \cdot \left\{1 - 0.02 \cdot \left(20^o - \alpha_n\right)\right\}$$

where the standard reference profile is defined with the following properties:

dedendum height factor $h_f^* = 1.2$

angle of attack $\alpha_n = 20 \quad [\text{deg}]$

Parabolic function of the stiffness for a single tooth pair contact

defined with the stiffness ratio $\boldsymbol{S_R}$

$$S_R = \frac{c_{\min}}{c_{\max}}$$

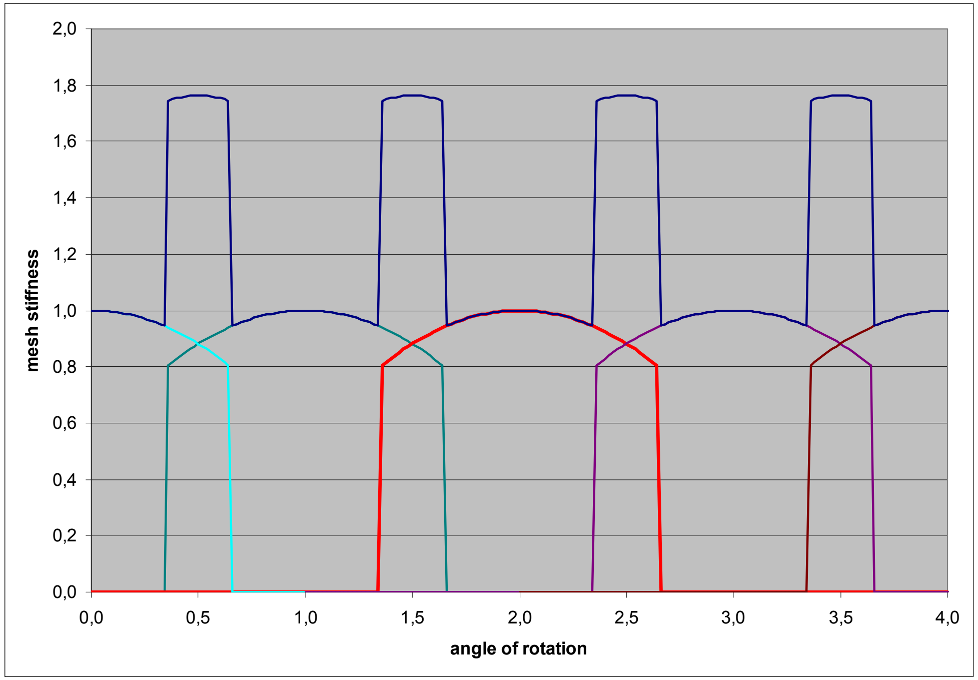
$$S_R = 0.80$$

where: $c_{\max} = c'$

$$c_{\min} = c' \cdot S_R$$

stiffness function

$$c(\varsigma) = c' \cdot \left(1 - (1 - S_R) \cdot \varsigma^2\right)$$



Contact Stiffness depending on Tip Relief

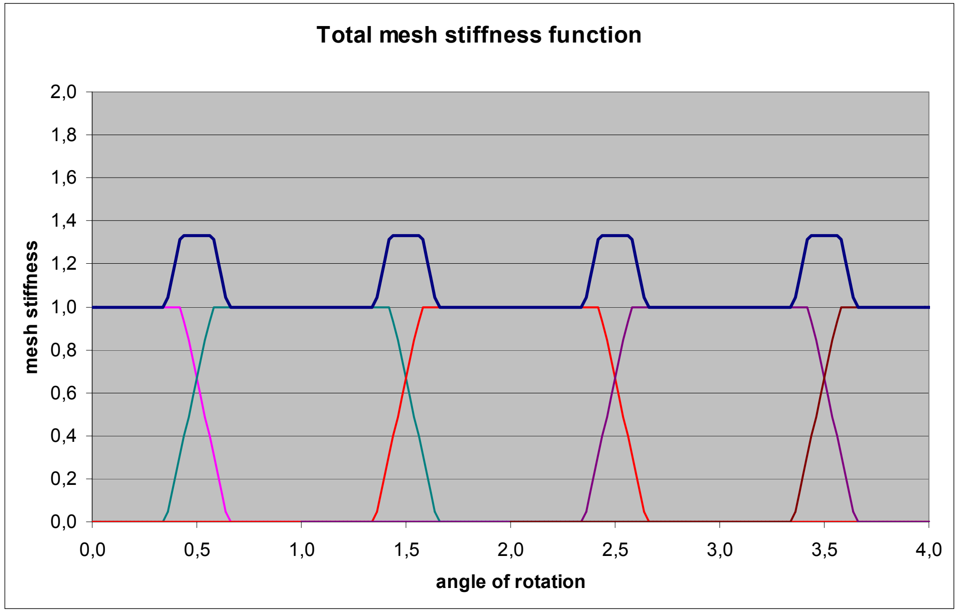
Using tip relief factor for modification of the total mesh stiffness function

example spur gear: $\varepsilon_\alpha = 1.3$

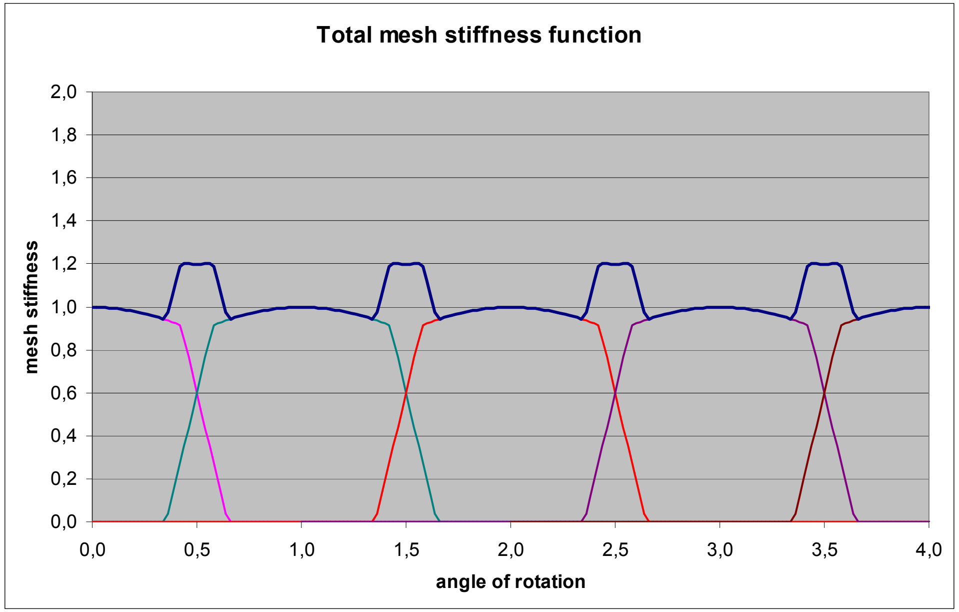
75 % tip relief

$$T_R = 0.75$$

$S_R = 1.0$



$S_R = 0.8$

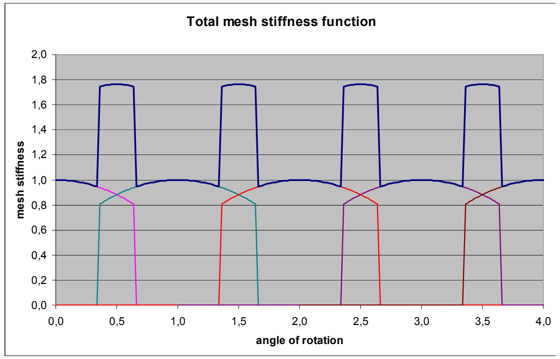


Finding the Contact Points

Special hints for modelling of spur gears

Why tip relief should be used

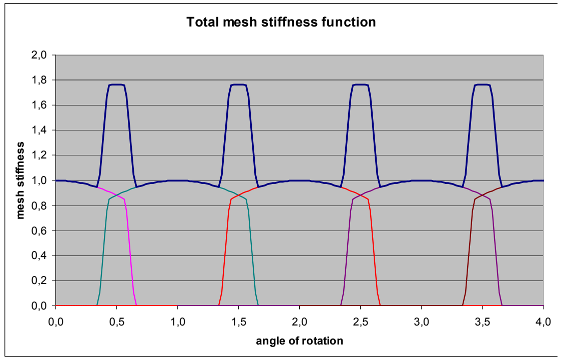
➢Without use of tip relief, each new tooth pair which is coming into contact, invokes a jump in the normal contact forces



➢If we would like to deal with this jumps, we must set Root functions for the gearwheel

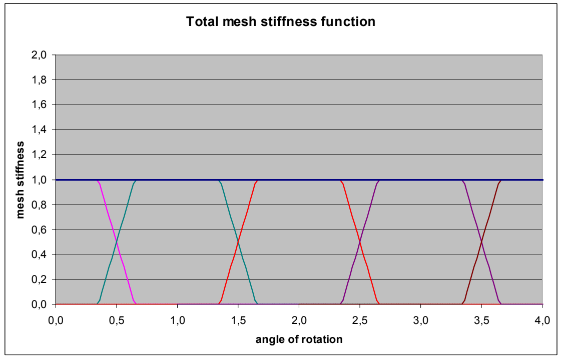
➢Use of tip relief involves an smooth steadily beginning of the contact forces

➢For spur gears a minimum tip relief factor of 0.1 is recommended



➢Linear contact stiffness relations are given for

$$S_R = 1.0 \qquad T_R = 1.00$$



Helix gears, function of the contact stiffness

The contact stiffness function of helix gears depends on the helix overlap ratio ε_β

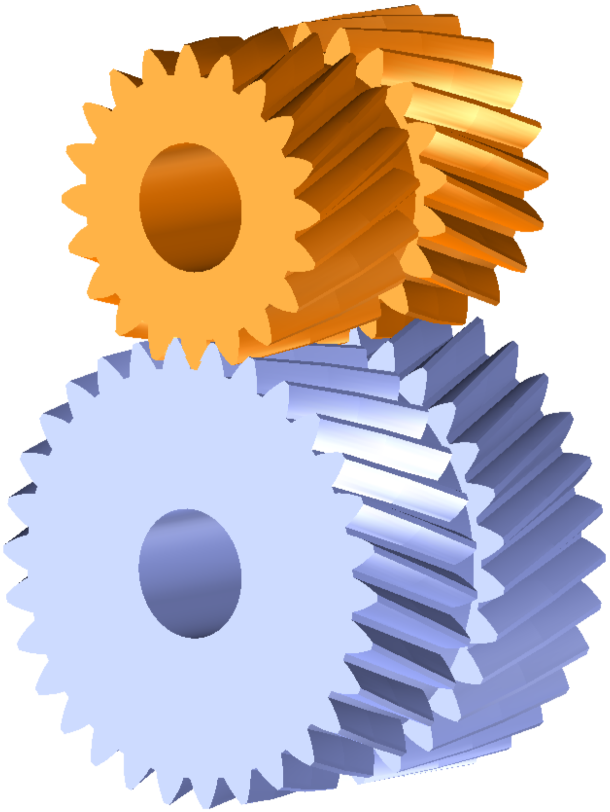
$$\varepsilon_\beta = b \cdot \frac{\sin \beta}{m_n \pi}$$

Using the function of the tooth pairing stiffness for spur gears,

$$c(\varsigma) = c' \cdot \left(1 - (1 - S_R) \cdot \varsigma^2\right)$$

the pairing stiffness function for helical gears may found as an integral of this function.

The mean axial position of the resulting stiffness function depends also on the scaled angel of rotation ς



Contact Stiffness depending on overlap ratio

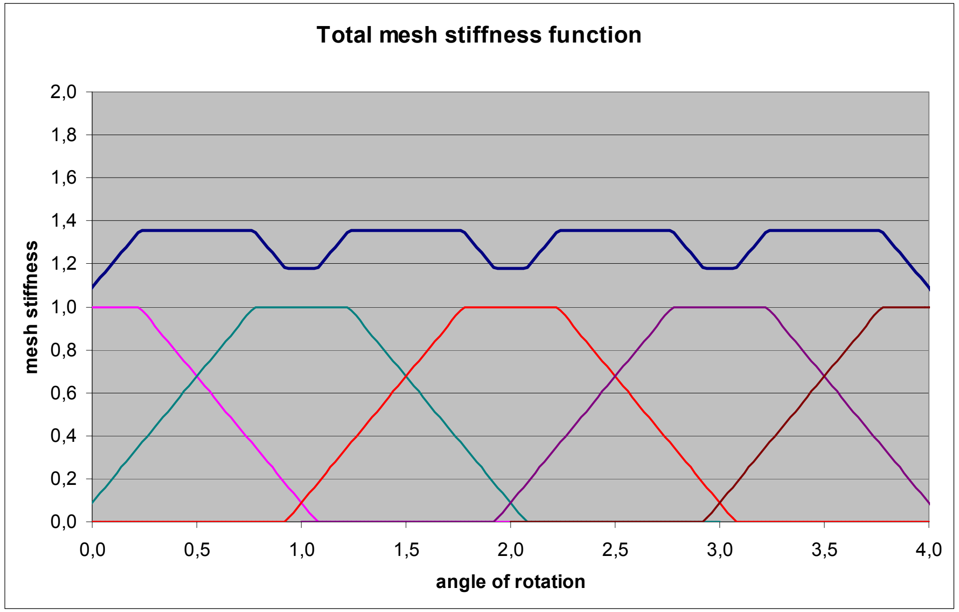
Helix gears, influence of the overlap ratio

overlap ratio $\varepsilon_\beta = b \cdot \frac{\sin\beta}{m_n \pi}$

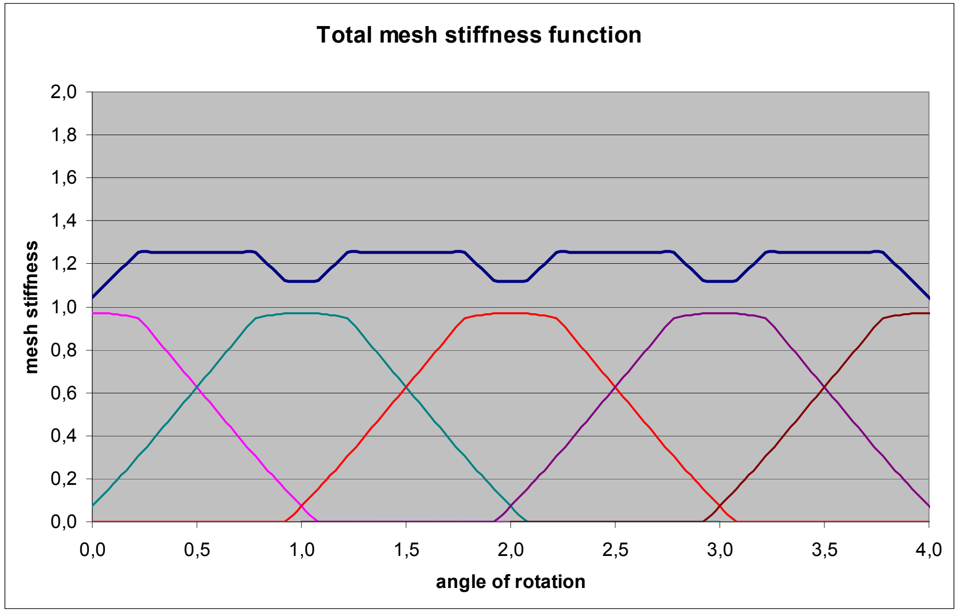
Example: contact ratio $\varepsilon_\alpha = 1.3$

$$\varepsilon_\beta = 0.85$$

$S_R = 1.0$

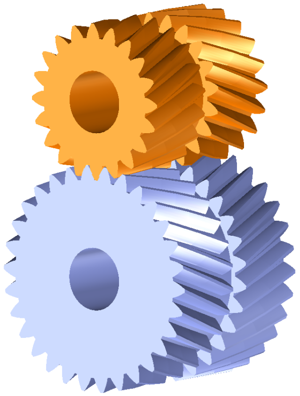


$S_R = 0.8$



Finding the Contact Points

What is the best Overlap Ratio?



The function of the total mesh stiffness depends on the overlap ratio strongly:

sharp upper edges for

$$\varepsilon_\beta = \varepsilon_\alpha - 1 + n$$

sharp lower edges for

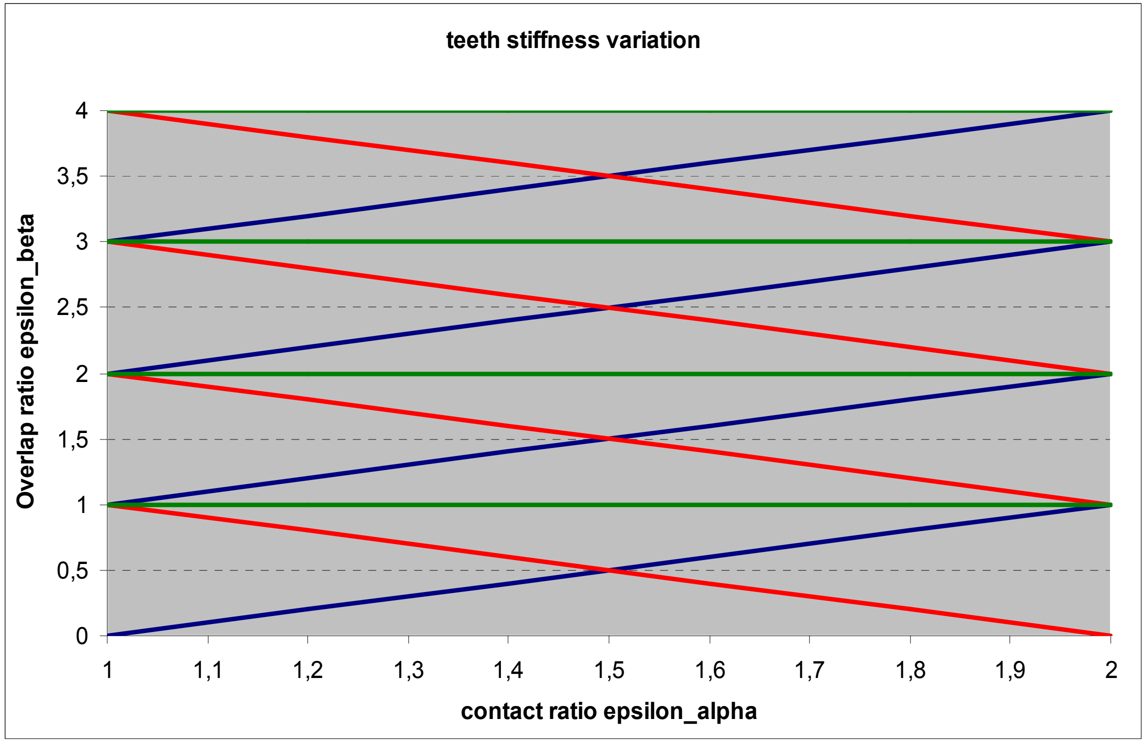
$$\varepsilon_\beta = 2 - \varepsilon_\alpha + n$$

constant function for

$$\varepsilon_\beta = 1 + n$$

where

$$n = 0, 1, \ldots, m$$

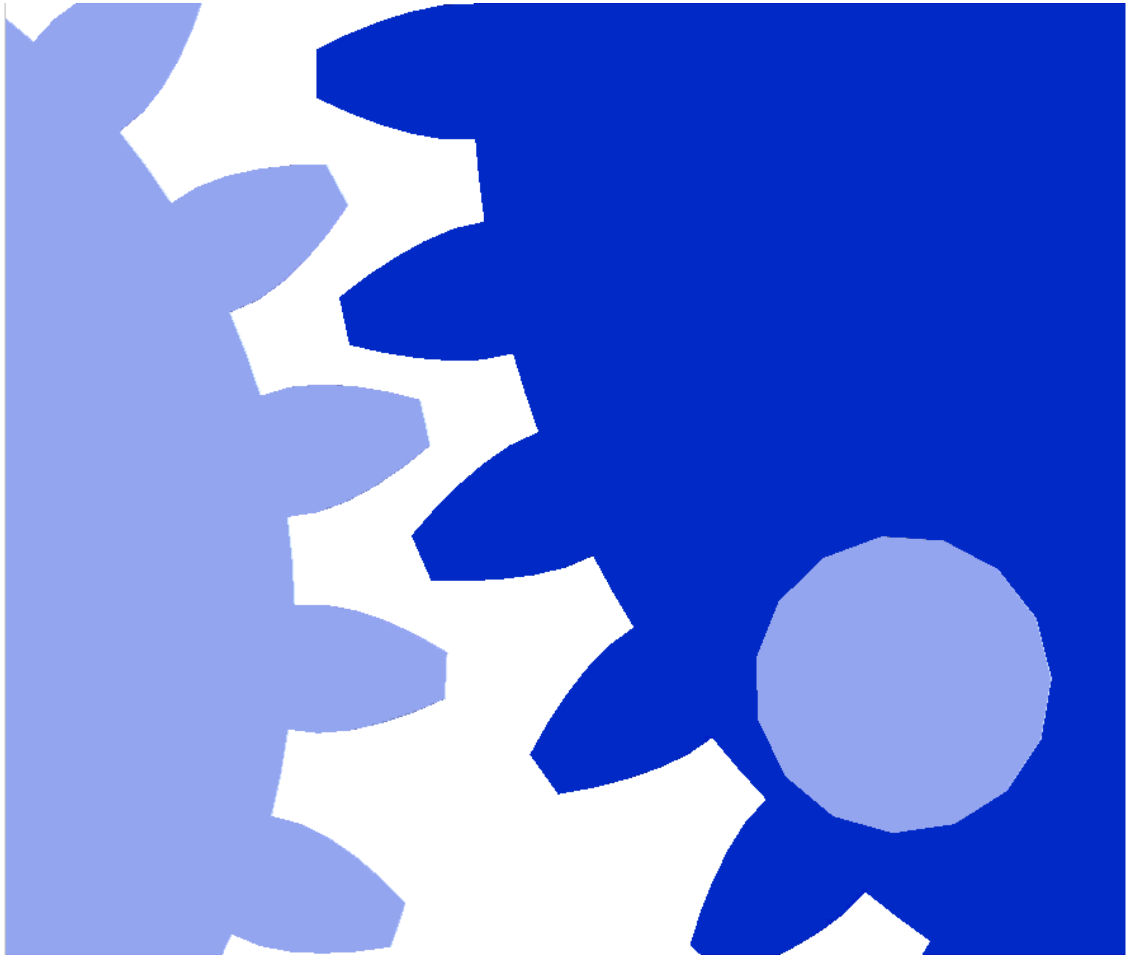


Dynamic input to the force element gear wheel

- rotational angle of both gears
- rotational velocities
- actual centre distance
- relative axial displacement (important for bevel gears)

Finding the locations of flank contact

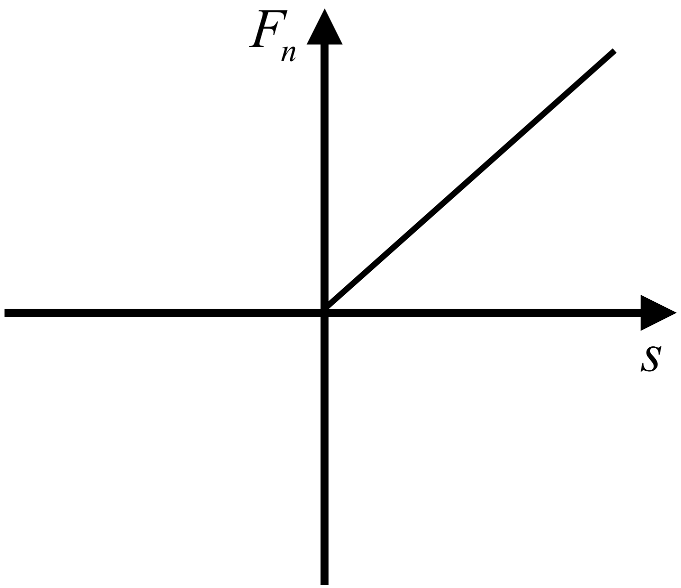
- the analytical determination of the contact point locations makes the numerical time integration fast, robust and reliable
- no discretisation errors

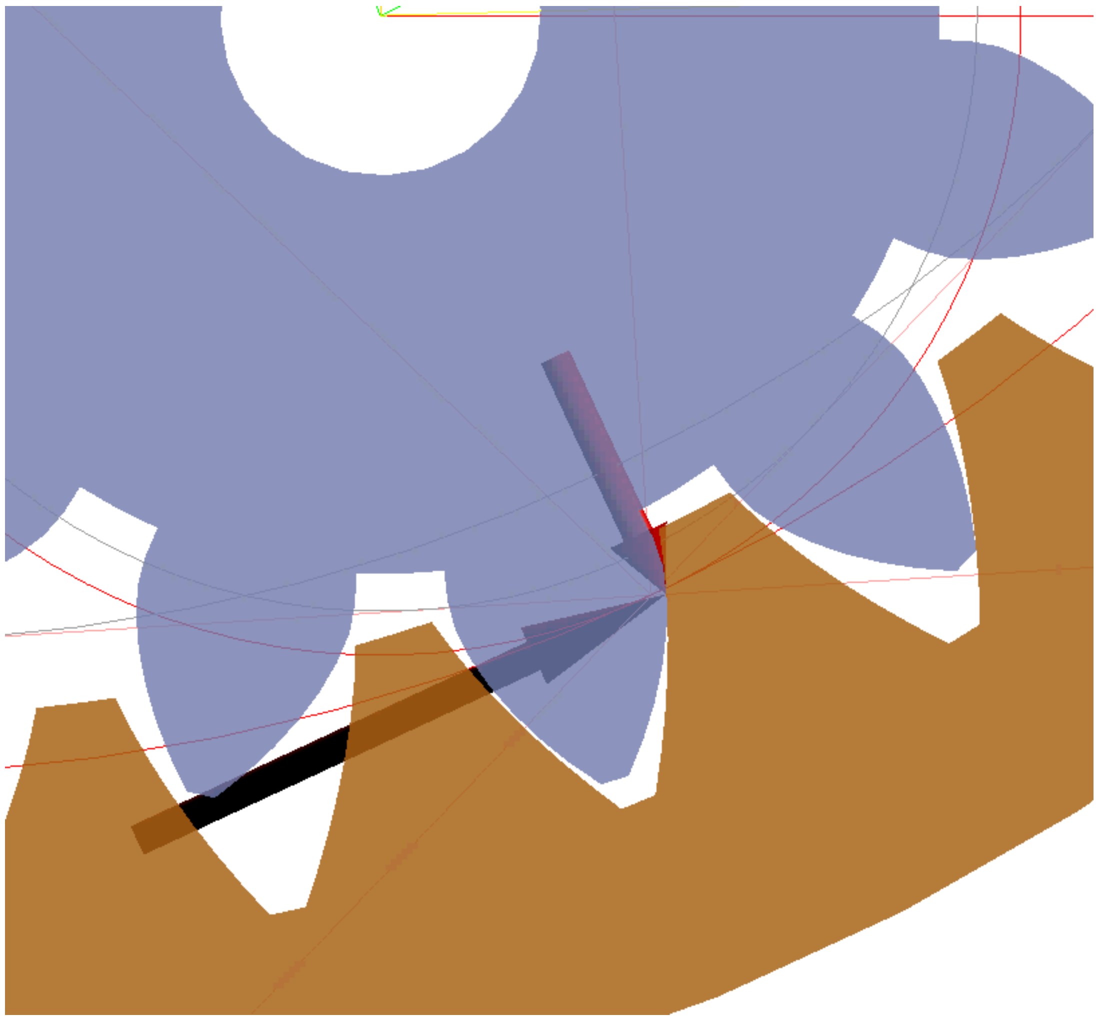


Contact Force Calculation

Impacts in tooth contact

All tooth contacts are modelled as one side acting springs. The impact forces are depending on the amount of flexible penetration.





Contact Force Calculation

Damping during tooth contact

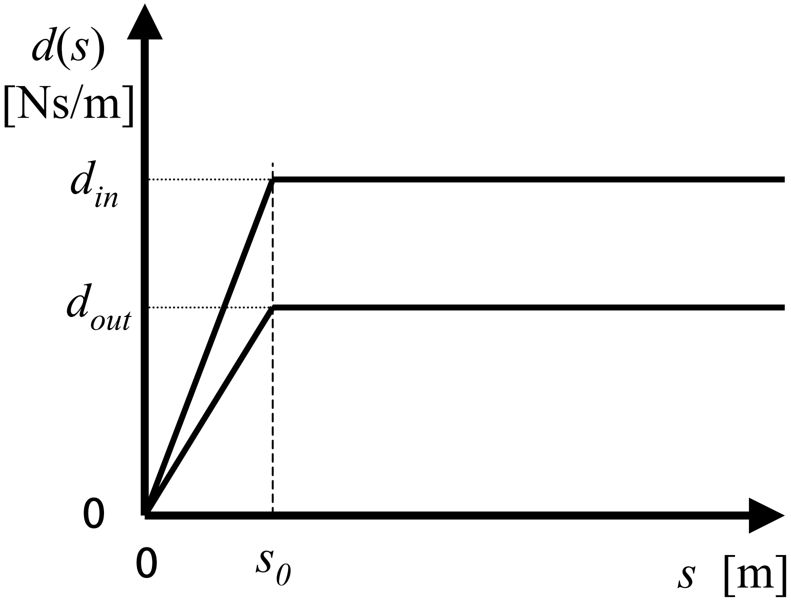
in normal direction

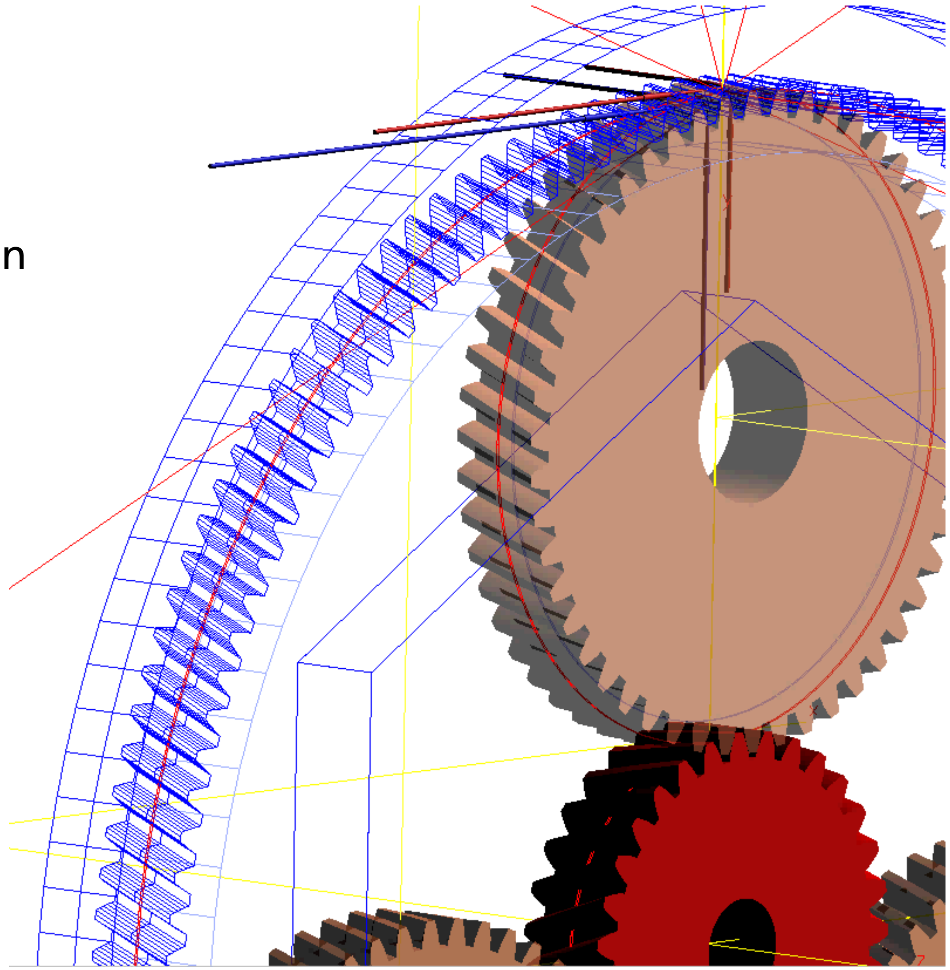
- viscous damping linear

d_{in} damping constant for compression

d_{out} damping constant for decompression

s_0 value of flexible penetration, where the full damping acts





Contact Force Calculation

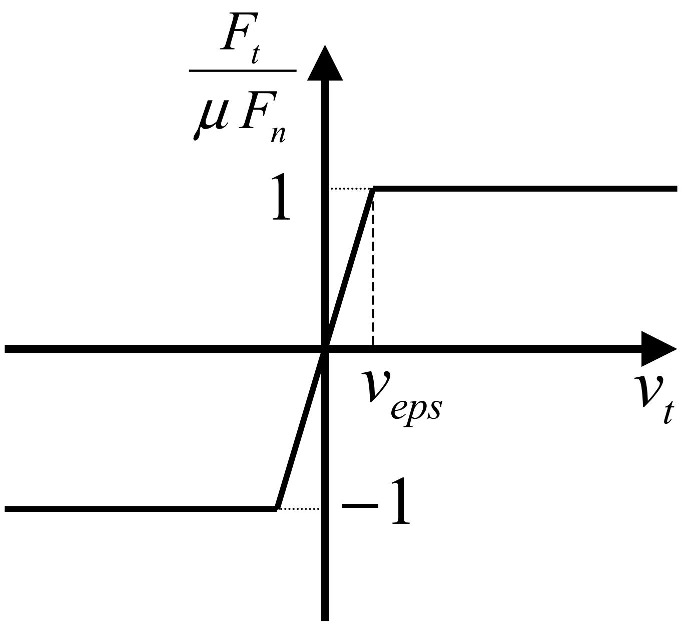
Damping during tooth contact in tangential direction

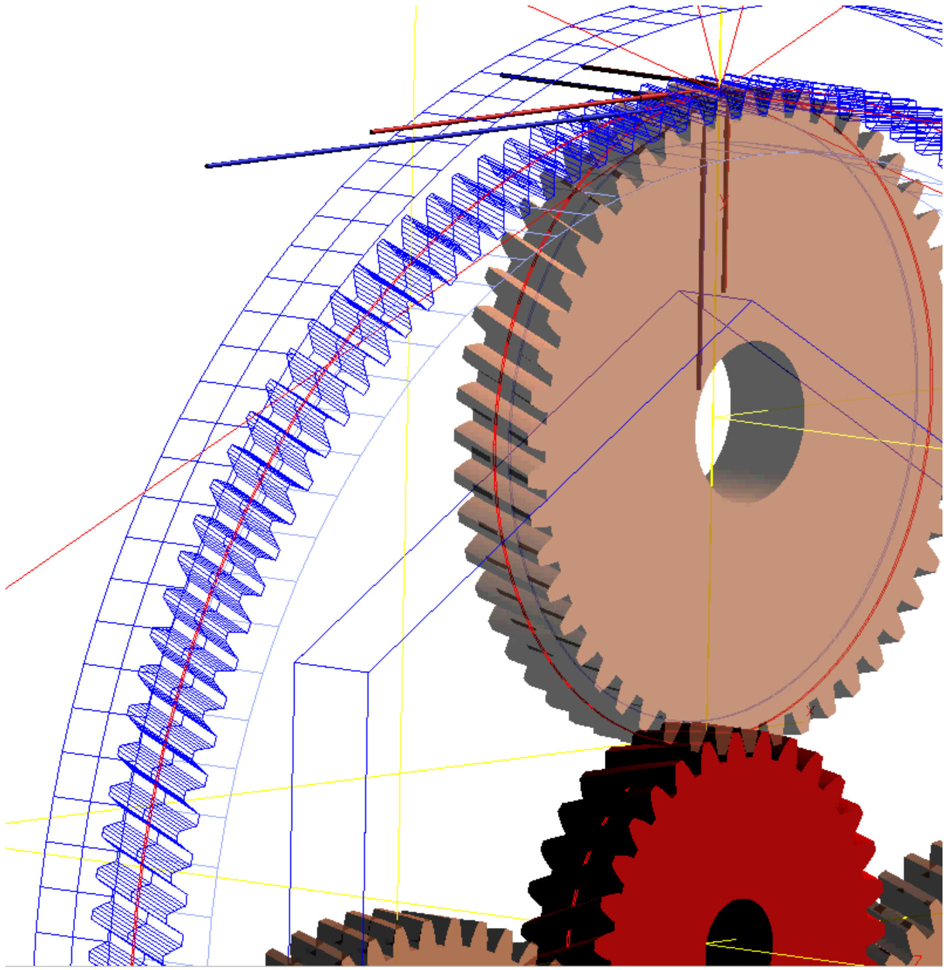
- Coulombic friction

v_t tangential velocity

v_{eps} Coulomb transition velocity

μ coefficient of friction

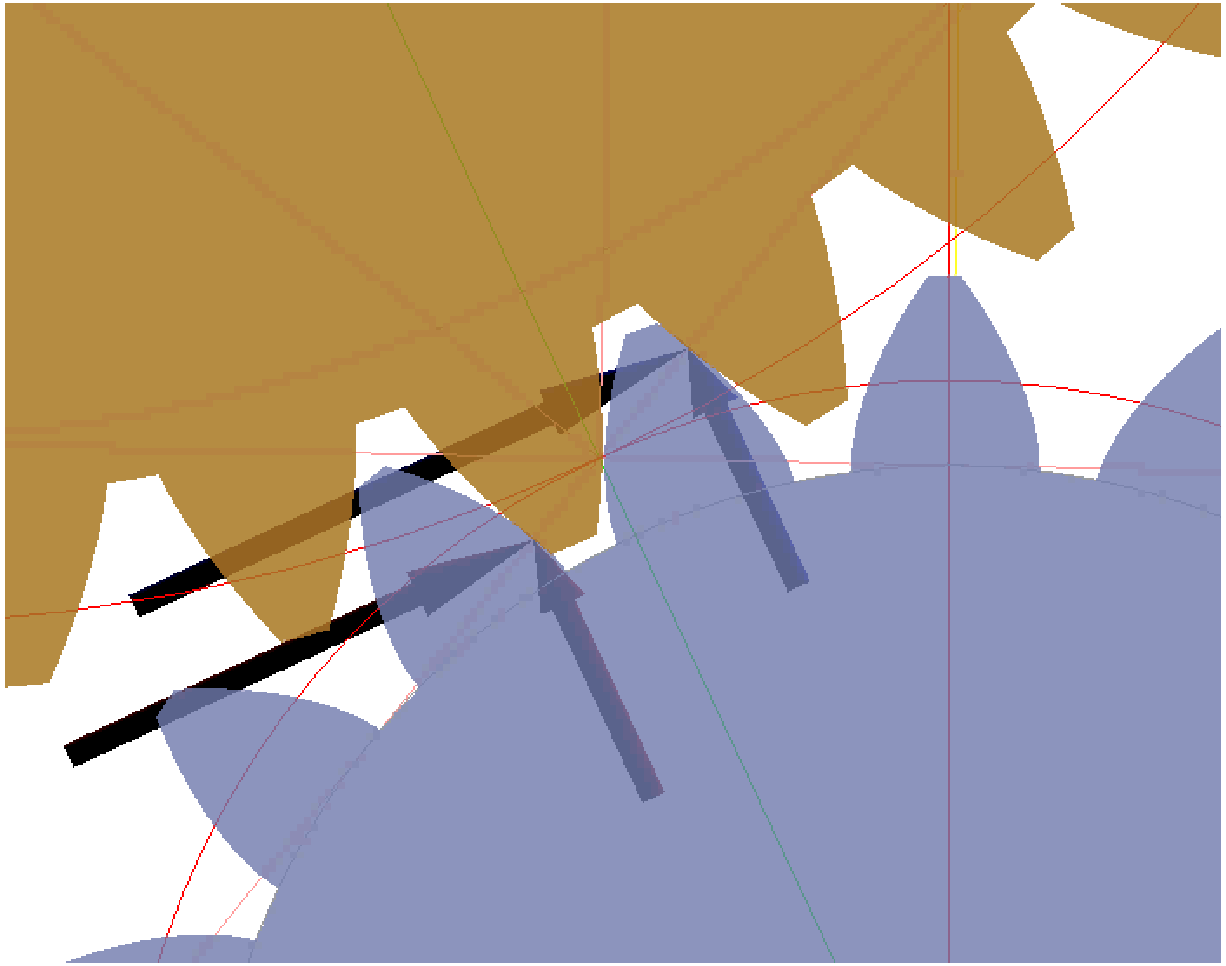




Animation of simulation results

The tooth contact forces may be represented in the animation of the MBS as scaled arrows in the following three components:

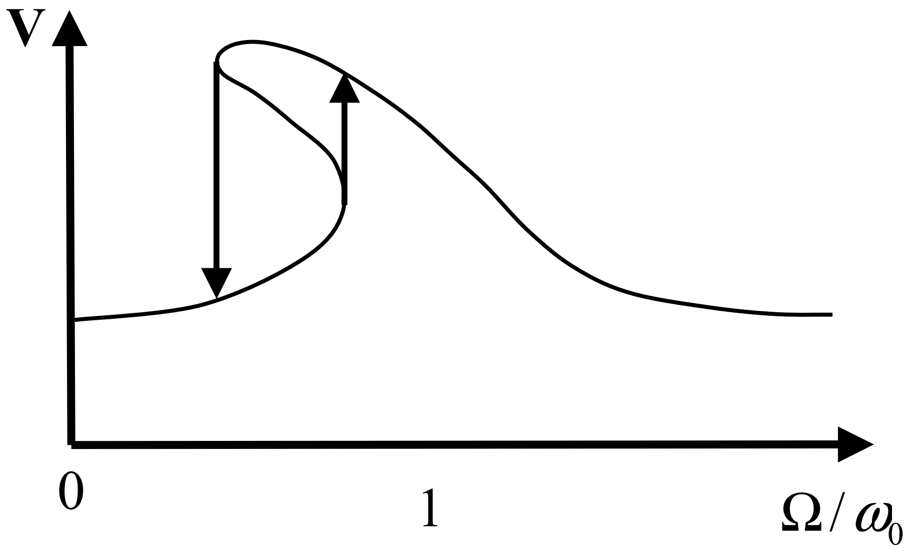
- circumferential force
- radial force
- axial force

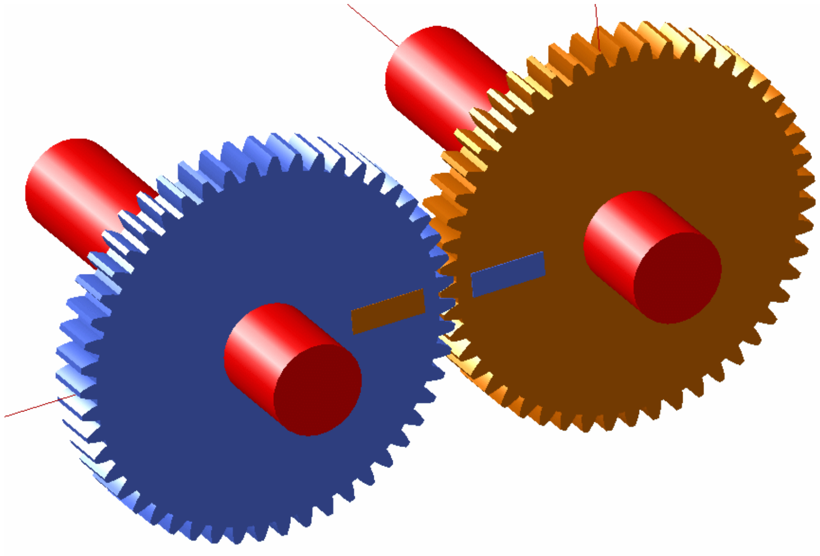


Example:
External pair of spur gears. Both gears are kinematical driven by a transmission ratio which is not exactly the ratio of the teeth numbers

Non-linear effect of gear pairings in the presence of backlash

Tooth gear pairings having backlash represents an oscillator with an under-linear stiffness function.



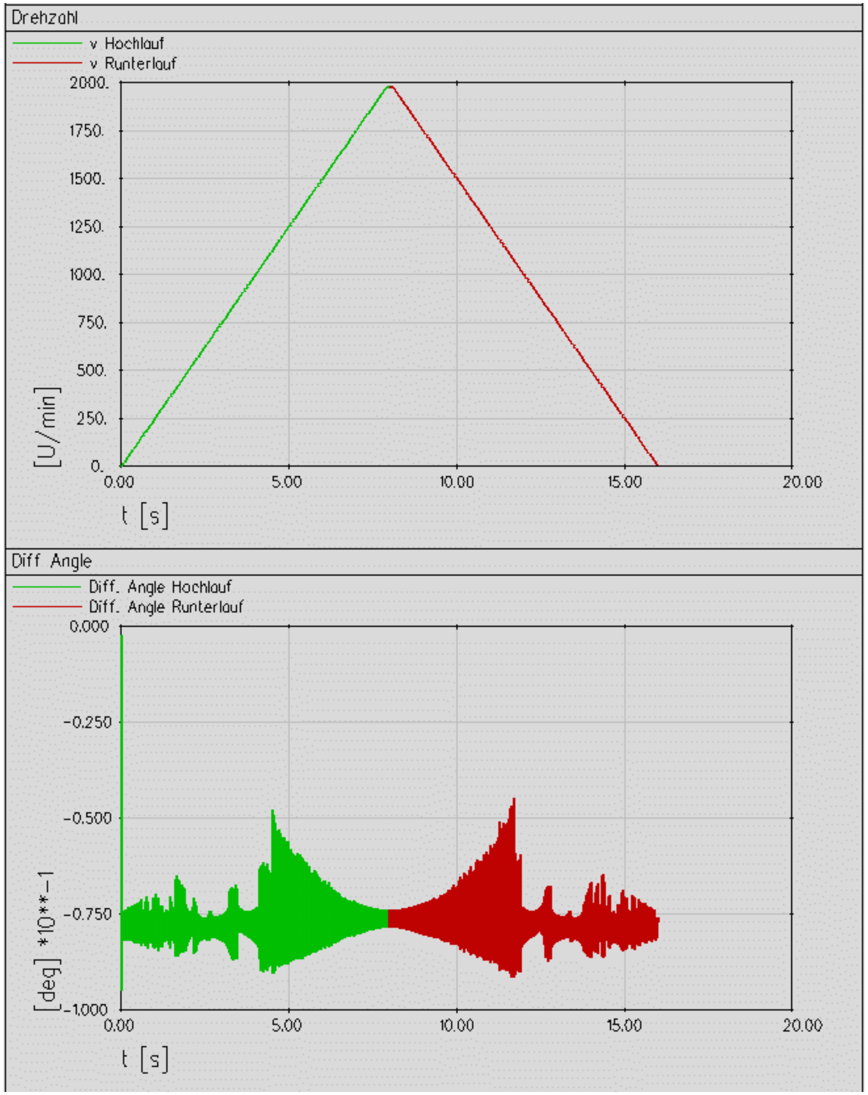


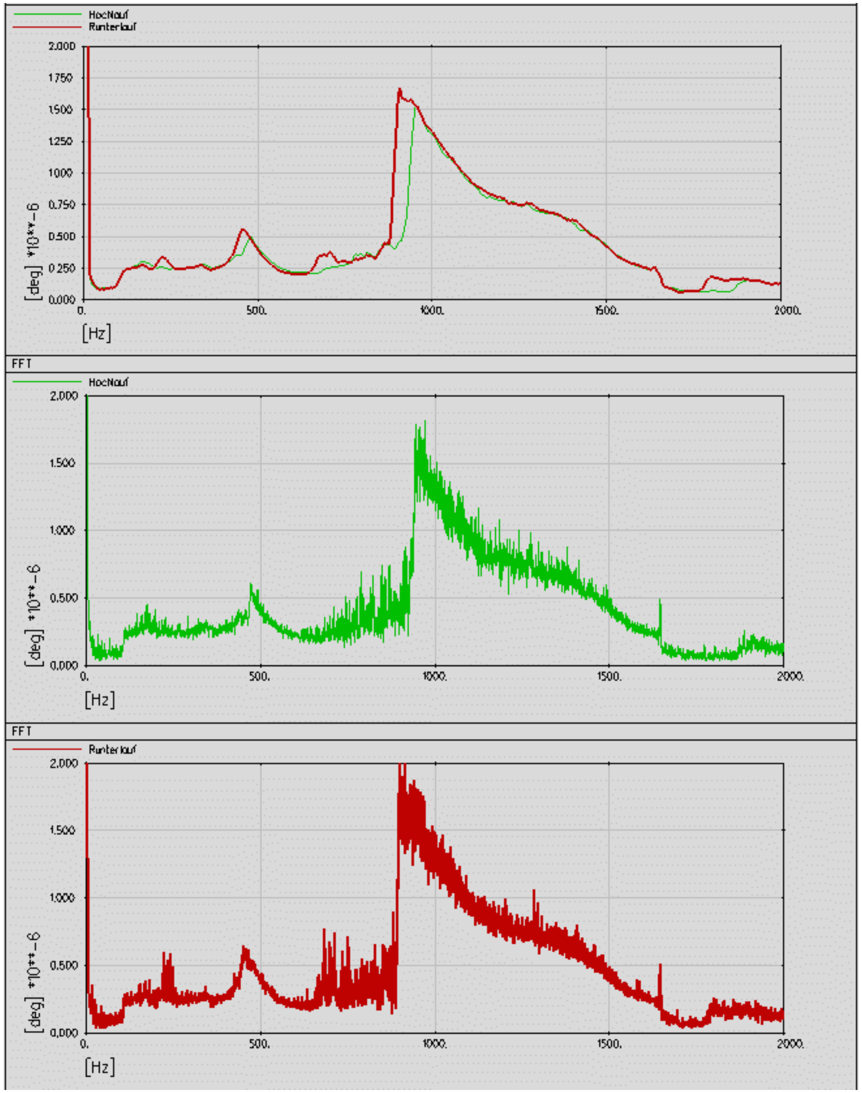
Literatur:
G. W. Blankenship, A. Kahrman: Steady State Forces Response of a Mechanical Oscillator with Combined Parametric Excitation and Clearance Type Non-Linearity.
Journal of Sound and Vibration (1995) 185(5), 743-765

Steady State Force Response

SIMPACK User Meeting 2006 in Baden-Baden

Frequenz-Sweep upwards green, downwards red





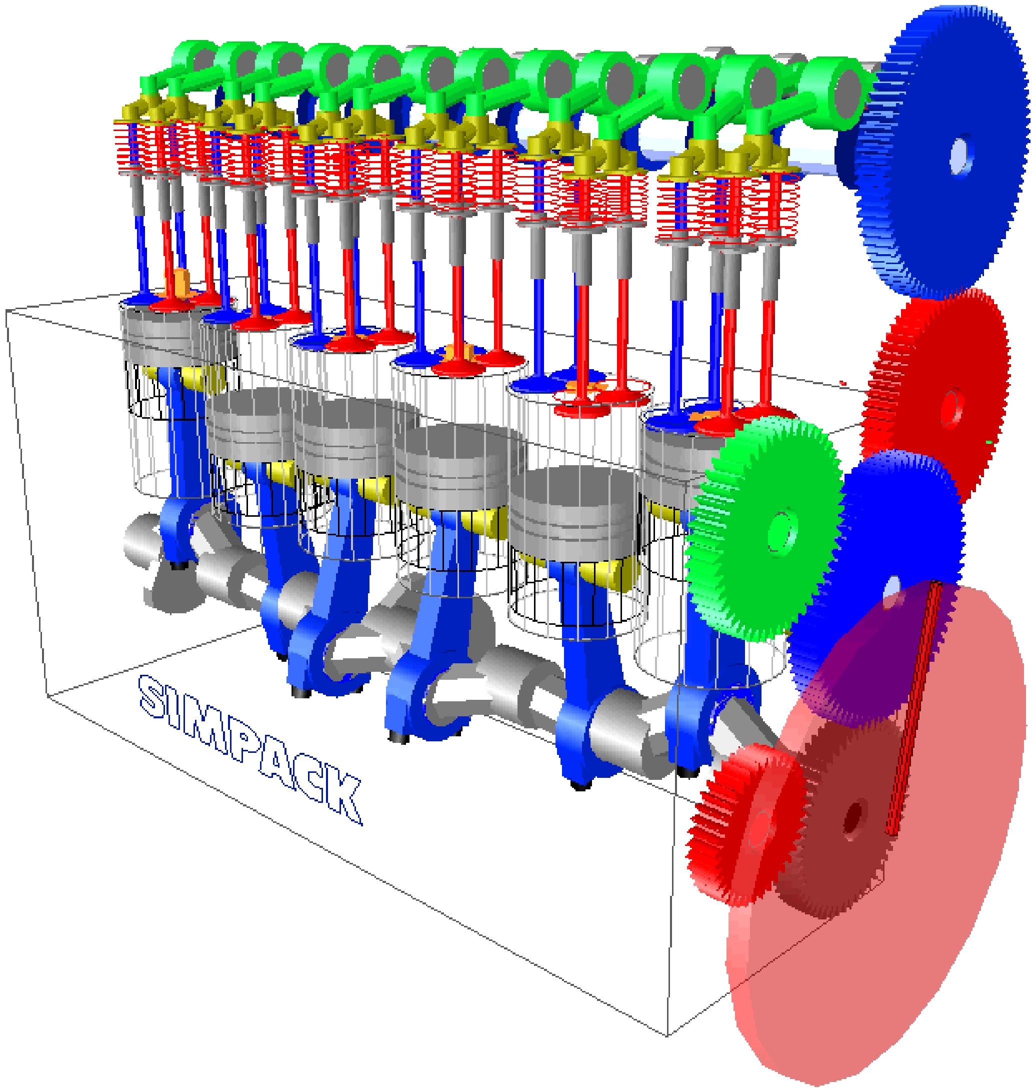
Application Example Timing Mechanism

Timing mechanism using gear trains

given problem
- high number of revolutions
- high dynamic loads

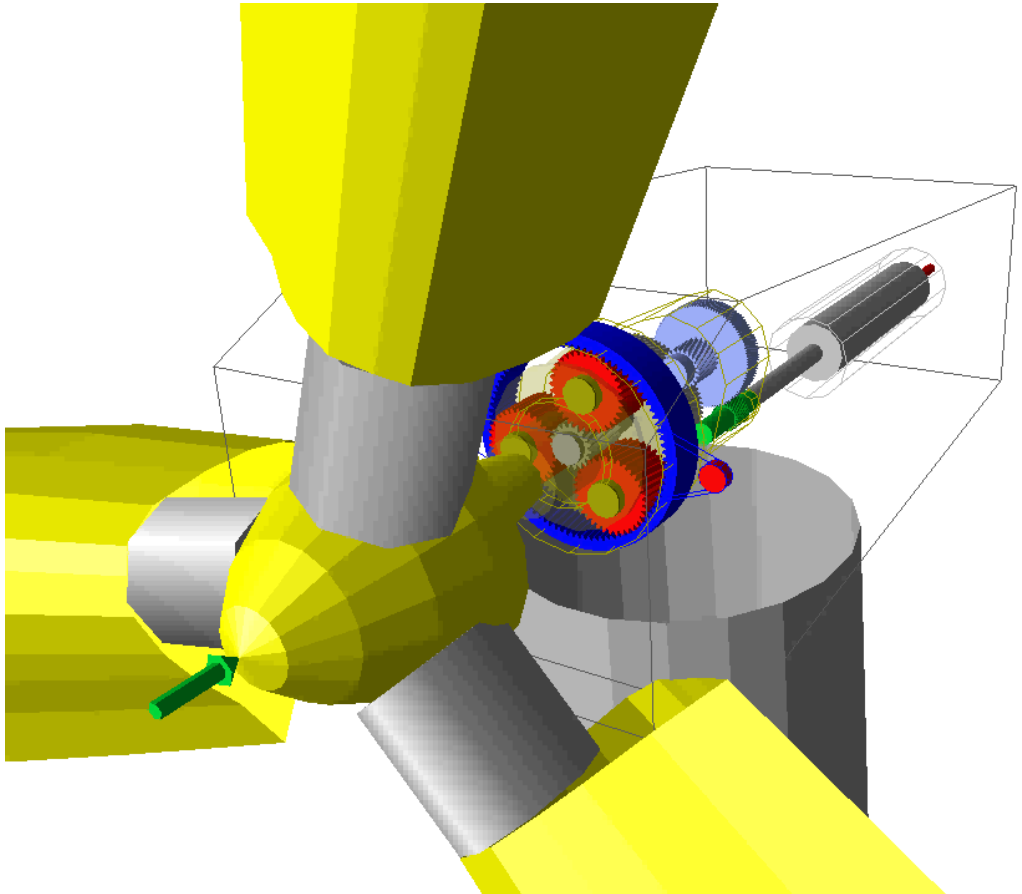
why gearwheels instead of chains
gear trains are stable for highest numbers of revolution

simulation technique
- Tooth meshing frequencies with more than 5000 Hz have to be processed.
- All tooth meshing interactions have to be described with the proper phase relations.



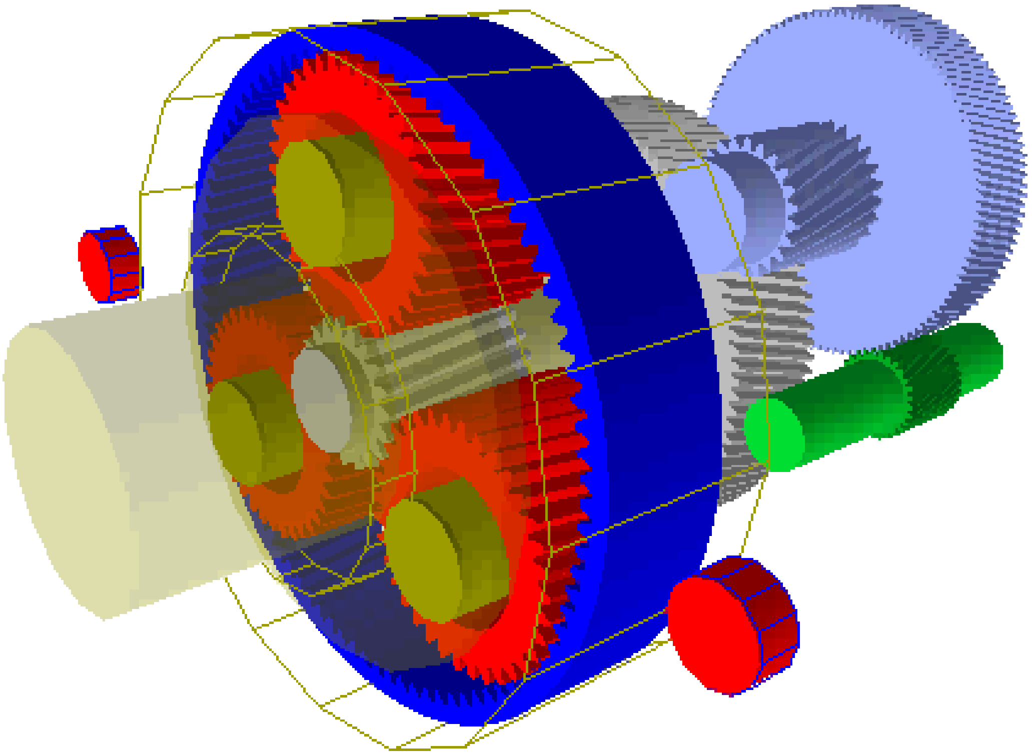
Wind turbine plant, total system models

- flexible components (tower, rotor blades, machine frame)
- detailed dynamic model of the power train including all gear stages, flexible axle couplings, brake and generator



Wind turbine plant, total system models

- generator controller and grid coupling (User fct., embedded DLL, or Matlab/Simulink s-function)
- Aero dynamic force calculation using blade element-theory (e.g. AeroDyn)
- active control of the blade pitch angle (e.g. co-simulation together with Matlab/Simulink)



Conclusion

- recursive order(n) algorithm in relative coordinates ⇒ set of minimal coordinates
- analytical description of the tooth profile geometry ⇒ no discretisation errors
  ⇒ no iterative algorithms
- consideration of changes in centre distance and in axial movement of the gears
- Parameterisation of the function of mesh stiffness ⇒ easy fit to static FEA
- contact force calculation for each individual toot contact
- complete coupling of drive train models within the three dimensional MBS
  - flexible bearing of the gear shafts
  - resilient moment strut mount
  - investigation of the overall system dynamics
- modellisation in substructure technique ⇒ analysis of sub models
- complete parameterisation of the models ⇒ easy change of model properties
- use of solvers working without numerical damping ⇒ reliable simulation results
- efficient solver technology ⇒ MBS-models > 1000 states
- wide reaching industrial application experience ⇒ high process reliability.