

Reference Point, Mesh Stiffness and Dynamic Behavior of Solid, Semi-Solid and Thin Rimmed Spur Gears

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ABSTRACT:

Most of the models proposed to predict the dynamic behavior of gears do not take into account the blank flexibility and its natural modes of vibration. In those models, a gears pair is reduced essentially to two rigid disks coupled by flexible teeth, which we call "Rigid Disks – Flexible Teeth" models (RD-FT). Contacting teeth pairs are replaced by a spring attached to the disks and acting in the normal direction at the contact points. The mesh stiffness, deduced from the stiffness value of each contacting tooth, depends on the position of the contact point and the number of teeth pairs in contact. Regardless of the intrinsic principle of the method and the boundary conditions applied, the displacement of the loaded point is measured relative to a reference point. The compliance of the tooth is largely affected by the choice of the reference point and so is the prediction of the dynamic behavior of the gearset.

This paper presents an investigation of the reference point in relation to mesh stiffness calculation and RD-FT dynamic models. The influence of the gear body flexibility is considered and the paper presents results for solid, semi-solid and thin rimmed spur gearsets. The analysis is made on the basis of the natural frequencies, when finite element models of meshing spur gears are taken as reference. Results show that a reference point inside the gear blank yields excellent correlation between the natural frequencies extracted from the RD-FT and F.E.A. models. That conclusion is true as long as no coupling effect between rigid body and gear body modes occurred. The reference point is located at the same place whether the gear body is solid or thin rimmed

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**REFERENCE POINT, MESH STIFFNESS AND DYNAMIC
BEHAVIOR OF SOLID, SEMI-SOLID AND
THIN-RIMMED SPUR GEARS.**

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NOTATION

b	= Half of Weber contact width
F	= Face width
F_n	= Normal load between teeth pairs
J_p, J_g	= Polar inertia of the pinion and the gear
K	= Mesh stiffness
K_0, K_{-1}	= Stiffness of teeth pairs 0 and -1
K_{TP}, K_{TG}	= Torsional stiffness of pinion and gear shafts
K_{BP}, K_{BG}	= Bending stiffness of pinion and gear shafts
K_{body}	= Gear body stiffness
m_p, m_g	= Mass of the pinion and the gear
r_{bp}, r_{bg}	= Base pitch radius (pinion and gear)
U_n	= Normal deformation at the contact point
X_j, Y_j	= Inertial coordinate (pinion and gear)
y'	= Distance from the line of action
z	= Axial coordinate
θ_{x3}, θ_{y3}	= Rotation of the pinion and the gear

1.0 INTRODUCTION

Gears have been recognized for a long time as an internal excitation source [2] of vibration and as a radiating medium [13]. Gears cause vibration mainly because of mass unbalance, transmission error, variable mesh stiffness, and backlash. In running condition, the pinion and gear mass centers move around a static equilibrium position because of the variable loads between teeth pairs in contact. In addition, elastic waves are generated and propagate through the gear bodies; resonance phenomena may occur if the gear blank is excited at certain frequencies [4, 8].

Özguven and Houser [12] have reviewed more than 200 hundred mathematical models used in gear dynamics. Most of the models proposed to predict the dynamic behavior of gears do not take into account the blank flexibility and its natural modes of vibration. In those models, a gear is reduced essentially to two rigid disks coupled by flexible teeth, which we call "Rigid Disks - Flexible Teeth" models (RD-FT). Contacting teeth are replaced by a spring attached to two rigid disks and acting in the normal direction at the contact points.

The mesh stiffness, deduced from the stiffness of each contacting tooth, depends on the position of the contact point and the number of teeth in contact. Elementary tooth stiffness is obtained from the ratio of the applied load to the displacement of the contact point. For spur gears, the problem is normally considered two-dimensional rather than three-dimensional as it is the case for more complex gear types. Several methods are available to calculate the compliance of gear teeth: the complex potential method [5], the finite element analysis (F.E.A.) [14], boundary element method [9], and other methods based on the strength of material and elasticity theory using or not the energy principle [6, 16]. Whatever the intrinsic principle of the method and the boundary conditions applied, the displacement of the loaded point is measured relative to a reference point. The compliance of the tooth is largely affected by the choice of the reference point and so is the prediction of the dynamic behaviour of the gear pair. If the reference point is chosen at the root radius, the compliance is smaller than if the reference point is situated at the bore radius.

The determination of tooth stiffness is also an essential step in the calculation of load sharing. For static load sharing predictions, every displacement that makes teeth move relative to each other must be taken into consideration [14, 15]. If the question of the reference point can be answered for static calculations, it is not yet