21st International Conference ENGINEERING MECHANICS 2015
Svratka, Czech Republic, May 11 –14, 2015
Paper #037, pp. 246–247

Dynamics of Simple Planet Gearing Model

Ladislav Půst[a] *, Luděk Pešek[b]

[1]Institute of Thermomechanics AS CR, Dolejškova 5, Prague 8, Czech Republic

[a]pust@it.cas.cz, [b]pesek@it.cas.cz

Keywords: planet gearing box, flexible pin, free and forced vibration, response curves

Abstract: Planetary gears have several advantages over parallel axis gears. Flexible pins of planets enable better power flow distribution in planet subsystems and increase the input torque density. Analysis of proposed simple mathematical model explains basic dynamical properties of planetary gearing with flexible pins, influence of pins' stiffness on frequencies and modes of free vibrations as well as on the response curves on external harmonic excitation.

Introduction

Planetary gearboxes have in comparison to the single parallel-axes mechanical gear transmission systems advantage in minimizing of weight of produce at given power due to distribution of internal force flow into several planet branch circuits [1, 2]. However the modal and spectral dynamic properties of such gearings are more complicated and therefore they need detailed dynamic analysis. The increased compliance of planetary gears axes can have essential influence on internal dynamic properties and very strongly can change also the reliability and service live of the whole product.

Presented paper is oriented on the first stage of mathematical model design, in which a detailed analysis of influence of compliance pivots of planetary gears on dynamic properties is mainly studied. The simplest plane type of gearings with three planetary subsystems has been selected from the many possibilities of basic types of planetary gearings. It is supposed that the central sun wheel, planet wheels as well as outer annulus ring wheel have cylindrical double helical (herring-bone) gearings. As this kind of gearings has an essential smaller variation of contact stiffness against the direct spur gearings, this stiffness variability is not taken into account and in addition also teeth contact compliance can be neglected with respect to the elastic deformation of planetary gears pins.

The steady contact in gearings without any interruption is asserted by means of preloading on the sun and outer rig gears. External dynamic excitation load is realized by means of harmonic moment acting on outer annulus gear.

The development of dynamic mathematical model of planetary gearing set starts on the simplest structure and will continue with the successive improvement. This procedure will be coordinated with the development of laboratory experimental investigation of gearing set prototype. The first stage of dynamic mathematical model of planetary gearing set is based on these assumptions:

a) Plane motion of all gearing wheels.
b) Rigid wheels, with inertia parameters Θ_1, Θ_2, Θ_3, m.
c) Rigid, exact gearings.
d) Compliant planet wheels pivots, stiffness k.
e) Entire system is without damping, linear.
f) Axes of the central sun and outer annulus rig wheels are fixed, wheels can only rotate.
g) All planetary gear wheels move identically.
h) Radial components of teeth contact forces acting on the planet gears are in balance, so that forces in teeth contacts can be replaced by the tangential components.

Equations of motion

Planetary wheels work as idle gear wheels in this transversal-torsion dynamic model, the motion of which can be described by means of three angles of rotation $\varphi_1, \varphi_2, \varphi_3$ of ring, planet and sun wheels and of tangential translation of planet wheel described by $r_4\varphi_4$. Graphical representation of kinematical situation is plotted in Fig. 1 and expressed by equations containing also two contact forces F_1, F_2.

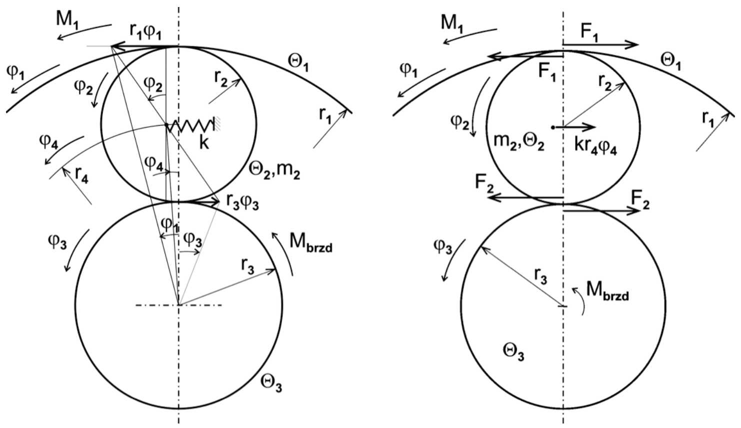

Fig. 1: Kinematical situation

$$\begin{aligned}
r_1\varphi_1 &= \quad r_2\varphi_2 + r_4\varphi_4 \\
r_3\varphi_3 &= -r_2\varphi_2 + r_4\varphi_4 \\
\Theta_1\ddot{\varphi}_1 &= -F_1 r_1 + M_{inp}(t) \\
\Theta_2\ddot{\varphi}_2 &= \quad F_1 r_2 - F_2 r_2 \\
m r_4\ddot{\varphi}_4 + k r_4\varphi_4 &= F_2 + F_1 \\
\Theta_3\ddot{\varphi}_3 &= -F_2 r_3 + M_{outp}
\end{aligned} \tag{1}$$

Solution

After eliminating contact forces F_1, F_2 and angles φ_1, φ_3 we get couple of equations with two unknown variables φ_2, φ_4. Theirs rearrangement gets expressions for modes and frequencies of free vibrations as well as for forced vibrations and corresponding response curves.

Summary

In the study of flexible pins stiffness effects on the dynamic behaviour of planetary gears, the following conclusions have been obtained:

Eigenfrequencies and eigenmodes of gearing set with fixed planetary carrier were ascertained for several values of flexible pin's stiffness, for several magnitudes of inertia moment of ring gear and of central sun gear.

Properties of forced vibrations were analysed by means of response curves. Comparison of responses obtained by application of sinusoidal excitation with constant amplitude of force moment with responses obtained at sinusoidal excitation with constant amplitude of displacement showed differences both in positions of resonance zones and in amplitudes of vibrations.

Acknowledgements: This work has been elaborated in a frame of the grant project TA04011656.

References

[1] Z. Doležal, Řešení planetových vysokootáčkových převodů metodou komplexních dynamických poddajností. In: Proc. Dynamics of machines, Prague, Liblice, (1977), 85-90.

[2] M. Hortel, A. Škuderová,: Dynamika nelineárních parametrických planetových soustav s kinematickými vazbami. Report ÚT AVČR Z-1512/14, (2014)