SIMULATION OF BACKLASH-FREE GEAR RUN USING TOOTH WHEELS CONTAINING FLEXIBLE ELEMENTS

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Abstract

The paper describes one of the principles of the backlash-free gear using tooth wheels. A design of a backlash-free gearbox with countershafts is solved in detail. Backlashes in a gear of tooth wheels and bearings are determined by a preloaded torsion-bar spring. A dependence of reaction forces applied to the tooth system and bearings on the torque bar preload is examined. Using a computer program created in the Maple environment, a loaded gear run is simulated in the speed reversal. The results are used to checkthe strength, material fatigue, and service life of dynamically loaded gear components .

Keywords: Backlash-free gear, gearing, torsion-bar spring, preload, accurate position control system

Introduction

In engineering practice, it is often necessary to position accurately physical objects such as work pieces, tools, mounted parts, conveyed (handled, transported) materials, finished products etc. A movable axis of a machine tool, a rotary positioning table, a robotic manipulator and many others can serve as a positioning device. In such cases it is necessary to accelerate and decelerate objects having a considerable weight, which implies that the drives of positioning systems work with relatively great forces and torques, and at relatively small speeds. According to the motion type, it is possible to classify drives into a) rotary ones, and b) linear ones. Electric motors of different designs belong to the most widespread rotary drives. However, these motors reach optimal efficiency and load characteristics at higher speeds and lower torques than the positioning mechanisms require. Therefore it is necessary to interpose a suitably designed reducing gear between the motor and the driven mechanism. Similar to any moving mechanism, the gearbox is produced with clearances and dimension tolerances. Between its input and output shafts, there are several series arranged clearances like this; therefore their sizes are added up. An adjustment of these backlashes during a reversal of the sense of the shaft rotation is a problem all the time. The paper describes one of the methods how to adjust these backlashes and to simulate the run of the loaded backlash-free gearbox.

1 Design

1.1 Principle of the backlash-free gearbox

Fig. 1a represents schematically the principle of the backlash-free gear with countershafts. The torsion-bar spring is mounted preloaded at a pre-determined value of the torque so that pinions act on meshing gears in opposite directions. When the rotary motion is transmitted, the load is transferred either by one path or by the other one.

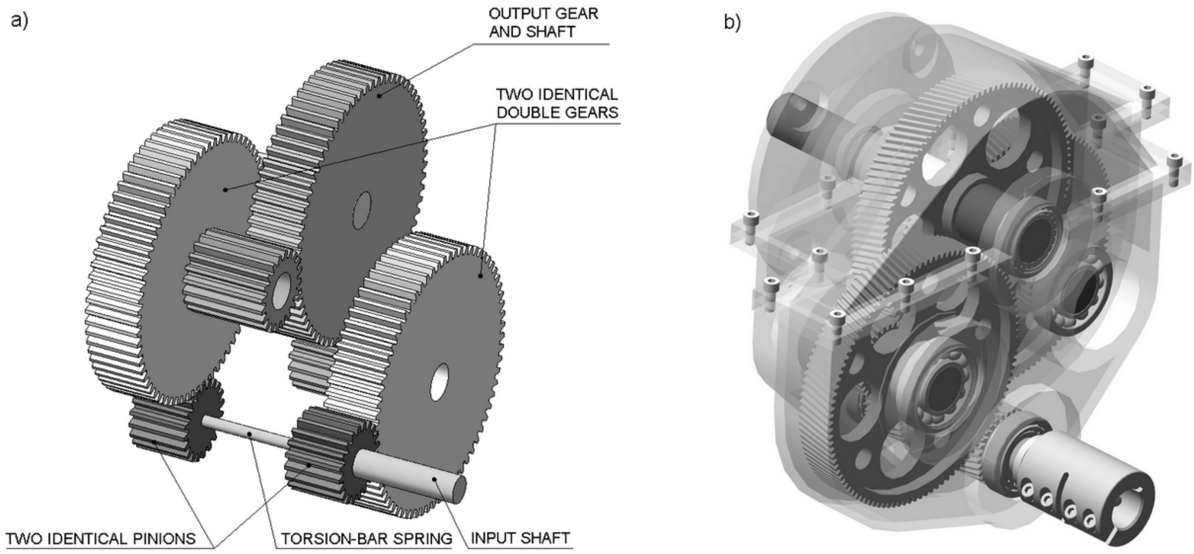

Fig. 1 a) anti-backlash gear assembly, b) prototype design of the backlash-free gearbox

1.2 Mounting conditions of gearing

The gearing (see *Fig. 1a)* is a closed range of tooth wheels. Therefore, when designing itis necessary to respect the limits given by the distances between the axes and the peripheral velocities of the tooth wheels. In the general case (*Fig. 2a*) the condition (1) has to be fulfilled

$$\frac{d32}{d31} = \frac{d22}{d12} \cdot \frac{d11}{d21} \tag{1}$$

Providing the same module of the tooth system in all the gears, it is possible to replace the pitch diameters by the number of teeth (2) in the equation (1)

$$\frac{z32}{z31} = \frac{z22}{z12} \cdot \frac{z11}{z21} \tag{2}$$

The advantage of this type of gearing is ahigh variability in dimensions and therefore a large area for optimization of specific applications. The following case was chosen for the prototype design and the subsequent calculations: $d11 = d12 = d31 = d32 = d1$ and $d21 = d22 = d4 = d2$ (*Fig. 2b*).

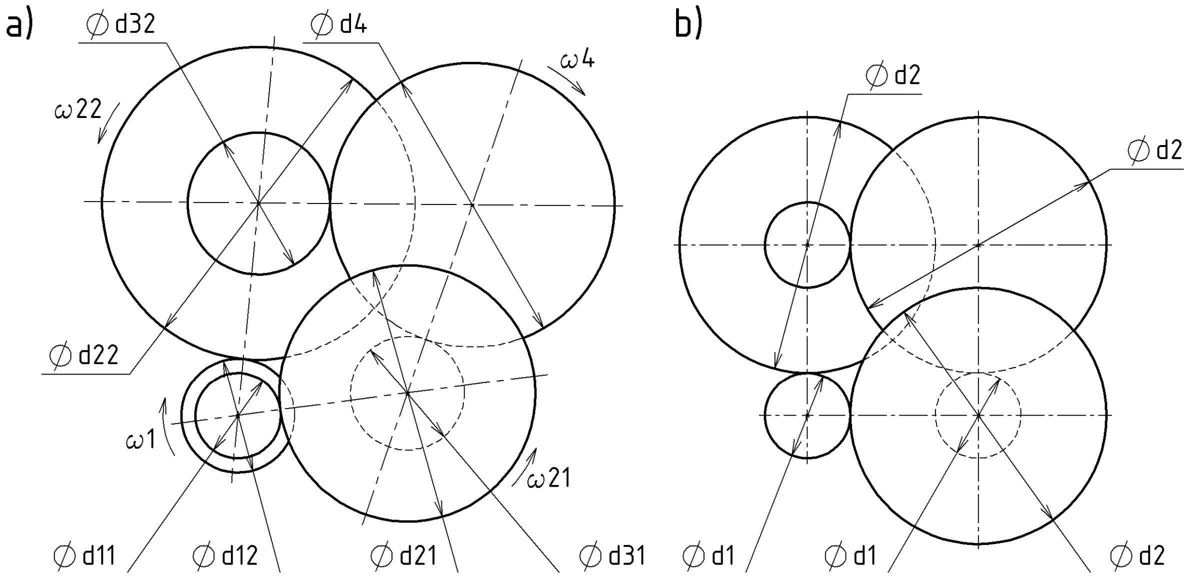

Fig. 2 Mounting conditions a) general case, b) prototype design

Tab. 1 lists technical parameters of the gearbox, which are used for subsequent calculations.

Tab. 1 Technical parameters of gearbox

rated power	P	2.9	[kW]
input torque	$M1$	18.6	[Nm]
input speed	$n1$	1500	[rpm]
gear module	m	1.5	[mm]
tooth number	$z1$	31	[–]
tooth number	$z2$	108	[–]
reduction ratio	i	12.1	[–]

2 Force conditions

2.1 Static equilibrium

The gearing according to *Fig. 1*a is a six-partcompound mechanism (including a frame). In *Fig. 3* particular parts are released and drawn in with all theacting forces. It is necessary to determine 38 unknown forces and moments; the moments of Mtk (preload of torsion-bar spring) and M2 (output shaft load) are the parameters of the equation system. Asequilibrium of the system of bodies in a space is concerned, 30 equations of equilibrium are available; remaining 8 equations result from the tooth system geometry (four gears of tooth wheels, 2 equations for each of them).

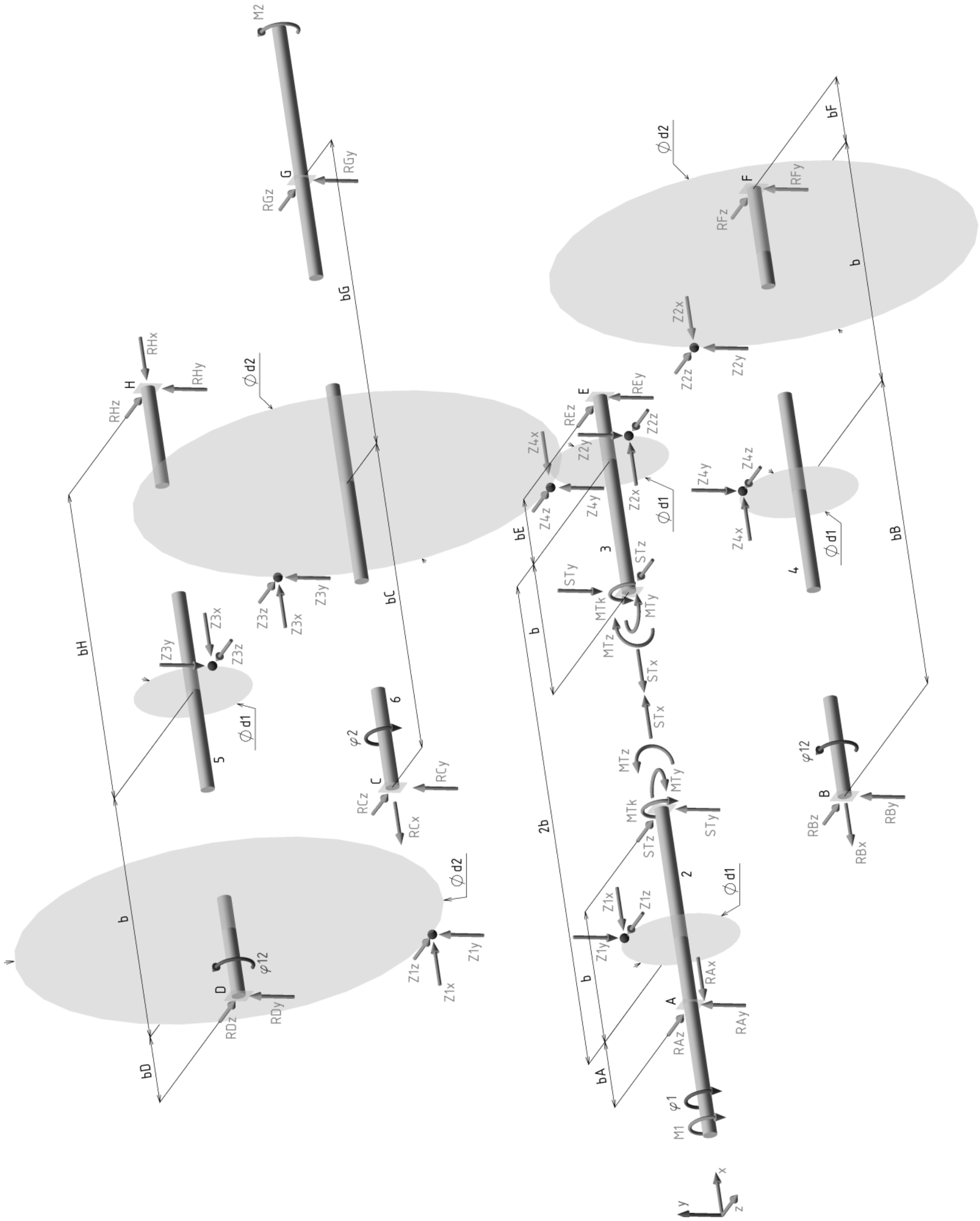

Fig. 3   External forces and forces on bearings, gear meshes and within torsion-bar spring

2.2 Tests of dynamics

A model according to *Fig. 4* was created in order to test dynamics. The gearbox is connected by a ball screw with a weight that is supported by a linear ball guide. A known desired kinematics of the weight motion is the model input. Reaction forces in the gearbox are calculated depending on the time. Especially tangential forces in gears of pinions are checked

– to keep the correct function of the gearbox , they have to be positive under all circumstances.

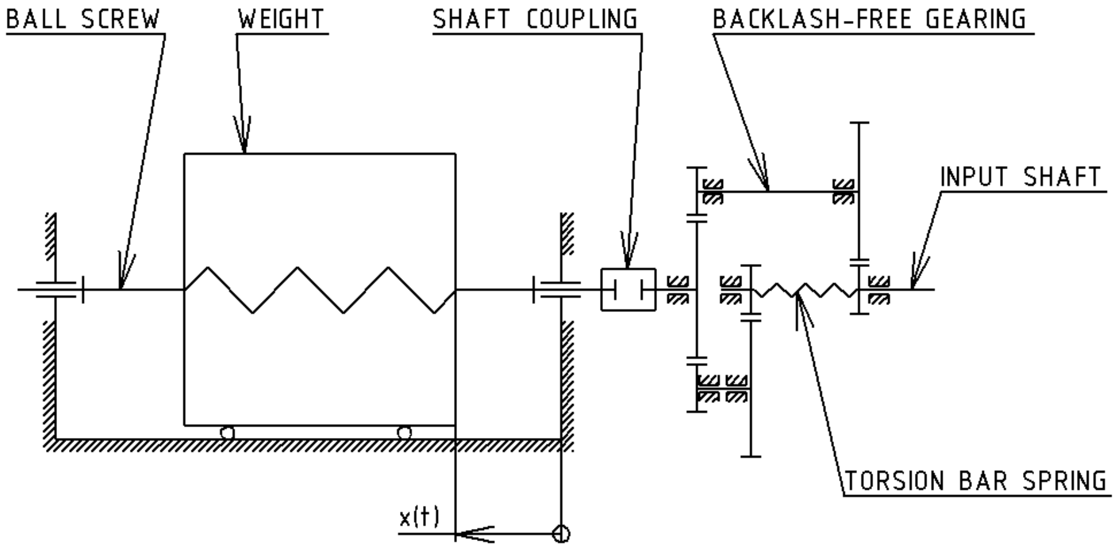

Fig. 4 Dynamic model

For purpose of a dynamic solution, it is necessary to know not only geometric characteristics but also mass ones. The weight mass is 100kg. Moments of inertia for tooth wheels, shafts and a ball screw were determined using a 3D model in the SolidWorks environment. For calculating, the universal calculation algorithm was made up in the Maple environment.

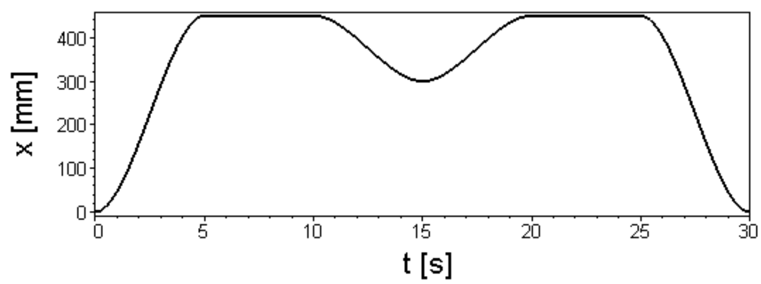

Fig. 5 Weight position versus time

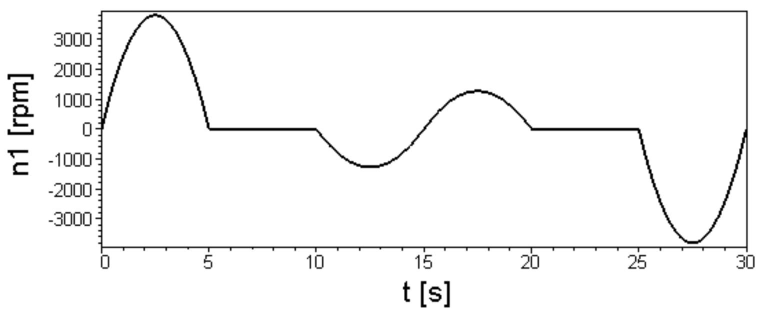

Fig. 6 Gearbox input shaft speed versus time

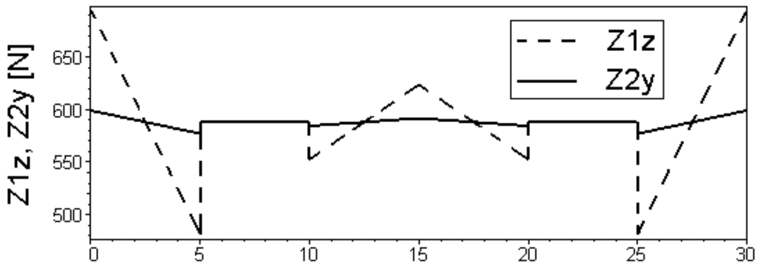

Fig. 7 Reaction forces in mesh of pinions versus time (Fig. 3)

Conclusion

The above mentioned method can be used for optimizing the gearbox dimensions. All the parameters of the created algorithm of calculation are established as variables, i.e. this algorithm is universal and applicable to all gearboxes having the same design. Each force applied to the calculation can be expressed separately as a function of time. Therefore, results are substantial for dimensioning of stressed parts as well as for checking of material fatigue. Also, the kinematics of an output mass is freely modifiable.

Acknowledgements:

This work was supported by MPO ČR, research project TIP, no. FR–TI1/594, Research of sophisticated methods of design and development of special-purpose machines, components and peripherals of industry machines.

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SIMULACE PROVOZU BEZVŮLOVÉHO PŘEVODU OZUBENÝMI KOLY S ELASTICKÝMI PRVKY

Článek obsahuje popis jednoho z principů bezvůlového převodu ozubenými koly. Detailně je řešena konstrukce bezvůlové převodovky s předlohovými hřídeli. Vůle v záběrech ozubených kol a v ložiscích jsou vymezeny pomocí předepjaté torzní tyče. Vyšetřena je závislost reakčních sil, působících na ozubení a ložiska, na předpětí torzní tyče. Pomocí výpočetního programu, vytvořeného v prostředí Maple, je simulován provoz zatíženého převodu při reverzaci otáček. Výsledky jsou použity pro kontrolu pevnosti, únavy materiálu a životnosti dynamicky zatížených komponent převodu.

SIMULATION DES BETRIEBS EINES ÜBERSETZUNGSGETRIEBES MIT HILFE VON ZAHNRÄDERN MIT ELASTISCHEN ELEMENTEN

Der Artikel enthält eine Beschreibung eines der Grundsätze einer spielraumfreien Übertragung durch Zahnräder. Dabei wird detailliert die Konstruktion eines spielraumfreien Getriebes mit einer Zwischenvorlegewelle gelöst. Der Spielraum beim Ineinandergreifen der Zahnräder und in den Lagern wird mit Hilfe eines vorgespannten Torsionsstabs definiert. Untersucht wird die Abhängigkeit der Reaktionskräfte, welche auf die Zahnräder wirken, und des Lagers von der Vorspannung des Torsionsstabs. Mit Hilfe eines Computer-Programms, das in der Maple-Umgebung geschaffen worden ist, wird der Betrieb der belasteten Übertragung bei einer Reversierung der Drehzahl simuliert. Die Ergebnisse werden verwendet, um die Festigkeit, Materialermüdung und Lebensdauer der dynamisch belasteten Bauteile der Übertragung zu kontrollieren.

SYMULACJA PRACY BEZLUZOWEJ PRZEKŁADNI Z KOŁAMI ZĘBATYMI Z ELEMENTAMI ELASTYCZNYMI

Artykuł opisuje jedną z zasad działania bezluzowej przekładni z kołami zębatymi. Szczegółowo omówiono konstrukcję przekładni z wałami pośrednimi. Luz przy ruchu kół zębatych oraz w łożyskach określony jest przy pomocy naprężonego skrętnego drążka. Na naprężeniu drążka skrętnego zbadano zależność sił reakcji działających na przekładnie i łożyska. Przy pomocy programu komputerowego, stworzonego w środowisku Maple, przeprowadzono symulację pracy obciążonej przekładni przy nawrotności obrotów. Wyniki wykorzystano do sprawdzenia wytrzymałości, "zmęczenia" materiału i żywotności dynamicznie obciążanych elementów przekładni.