Towards Intelligent Planocentric Gear Train for Robotic Industry – Part 1

Gorazd Hlebanja, Miha Erjavec, Luka Knez, Simon Kulovec, Jože Hlebanja

Abstract:
A design and development of a planocentric gearbox to be used in robot arm joints, namely for a collaborative robot, is presented in the papers. Besides strict limitations regarding the near zero backlash, the output position and torque should be available as well. So, a spatial-awareness encoder and a torque sensor are incorporated in a gearbox. The first assures an accurate absolute output position and the latter ensures information on actual output torque. The presented solution is based on the S-shaped tooth flank geometry. The near zero backlash requirement requires a tolerance analysis, which was accomplished by simulations in KissSoft software. The results of the analysis enabled a successful modification of the gearbox. A sophisticated testing rig was developed to verify actual gearbox characteristics, and to test its short-term and long-term behaviour. Backlash, stiffness, kinematic error, and dynamic behaviour of produced gear trains are measured in this way.

Keywords:
planocentric gear train, backlash, tolerance analysis, S-gearing, torque sensor, spatial-awareness sensor, testing rig

1 Introduction

Planocentric gearboxes are used in robotics, machine tools, aeronautics, aircraft, marine and many other industries. The efficiency can be above 90 % and gear ratios can achieve up to 160 : 1 (160 rotations of the input shaft for a single turn of the output shaft). Basic arrangements of this type are described in well-known references [1]. The available industrial solutions include *Sumitomo* cyclo gearboxes [2], *Spinea* drives [3], *Nabtesco* [4], *Onvio* [5] and many others. Gearings are usually cycloidal or lantern. Some solutions combine a classic planetary gear train and a cycloidal stage with three eccentrics having origins in the centers of the planets. Some other devices incorporate a threefold eccentric rotating three planets. Such gearboxes are more compact but also more complex. All producers claim near zero or zero backlash. There is also a strong patent activity all over the world, especially in Russia and China, which indicates the importance of the field. The mentioned companies are holders of such patents as well. The expected characteristics are low hysteresis for accurate positioning, low lost motion, compact design, high torsional stiffness, low inertia, high efficiency, overload capacity, easy assembly, and lifetime lubrication. Some robot producers develop their own gearboxes in order to achieve the best possible accuracy and reliability.

Potential advantages of planocentric gear boxes, namely a high-speed reduction ratio combined with

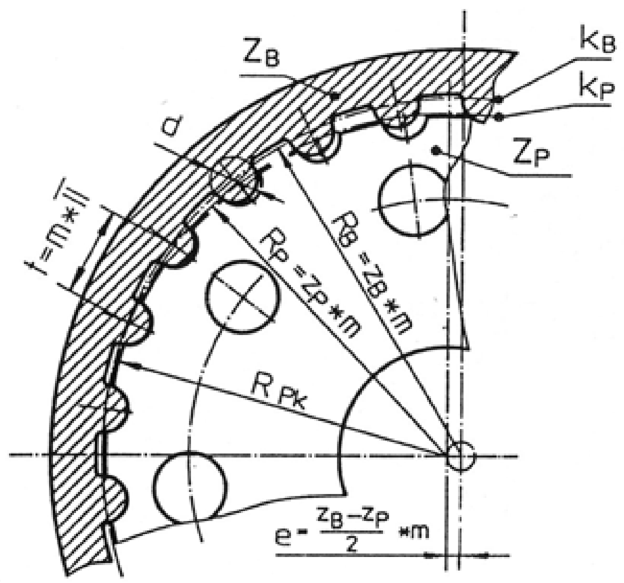

Figure 1 : *Gear ring and planet gear during meshing [6].*

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high output torque in a relatively small volume, were a driving force for a new development already twenty years ago. So, the modified lantern gears were developed for use in planocentric gear drives with eccentric [6] and patented [7]. The internal gear pair during meshing is illustrated in *Fig. 1*, clearly indicating the value of eccentricity.

The teeth of the ring gear are designed as semi-circular extremities, whereas the planetary gears are designed with corresponding semi-circular spaces, adapted in size for a tolerance. Design aims were focused in automatic production lines and CNC--machinery. The planocentric lantern gear box design was robust. The transmission of rotation from the planet gears through a pin composition to the output shaft was provided by a single sided cage with double bearing output shaft. Small series were produced with various gear ratios up to 100 and with various modules, down to m = 0.5 mm.

Pure mechanical drives can conform to high tech industry requirements with regard to backlash, lost motion, stiffness, hysteresis, etc. However, supplementary features based on sensorics can add additional functionalities to such a gearbox. So, an accurate output shaft positioning and an output torque sensorics are installed in the device as an option. However new functionalities enable the incorporation of such devices in collaborative robot's arm joints and adaptive control. An upgrade to a self-aware condition monitoring system could increase the overall reliability of the drive and the effective predictive maintenance whereas a corresponding condition monitoring could enable safe human interactions which is of special importance e.g. in the field of robotics.

Thus, the paper describes how the planet gear moves based on the eccentric rotation and induces the output rotation. Since the gearing geometry is based on an S-gear flank profile, some ideas about this gear type are explained. Next, the gearbox prototype is revealed. The described testing rig enables all necessary tests, e.g. stiffness, hysteresis, kinematic error and durability measurements, which are necessary in confirming the prototype design or indicating possible improvements. A longer chapter discusses influences of tolerances. Findings of tolerance analysis helped in improving the gearbox design furthermore. In the final chapter, the torque flange is described. The torque sensor is in its final stage of development, so the flange design, the strain-gauges, and electronics and signal transmission are already defined.

2 Kinematic Circumstances of a Planocentric Gear Train

The planocentric gearbox has coaxial input and output shafts, and large transmission ratios can be achieved based on a gear ring with internal gearing in combination with usually two planet gears with external gearing, where the difference in the numbers of teeth between the gear ring zv and the planet gears z_p rules the output gear ratio, Eq. (1). The difference in ring and planet numbers of teeth should be one.

$$i_{out} = \frac{z_p - z_v}{z_p} \tag{1}$$

The planet gears are mounted on an eccentric shaft, where bearings separate the planet gears

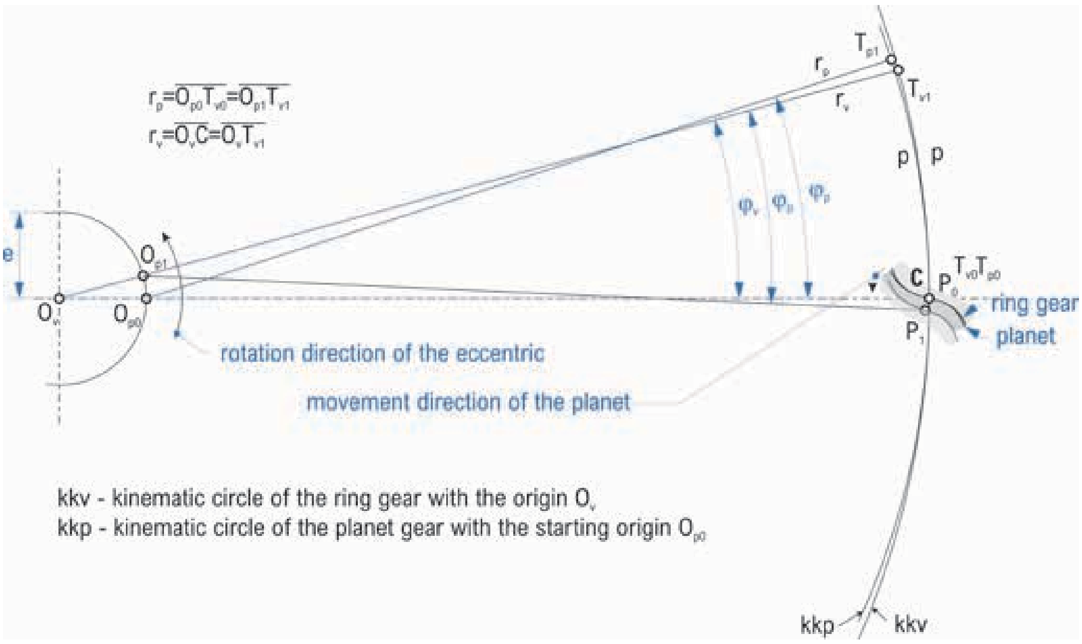

Figure 2 : *Planetary gear movement in accordance with hypocycloid generated on the kinematic circle of the ring gear [8].*

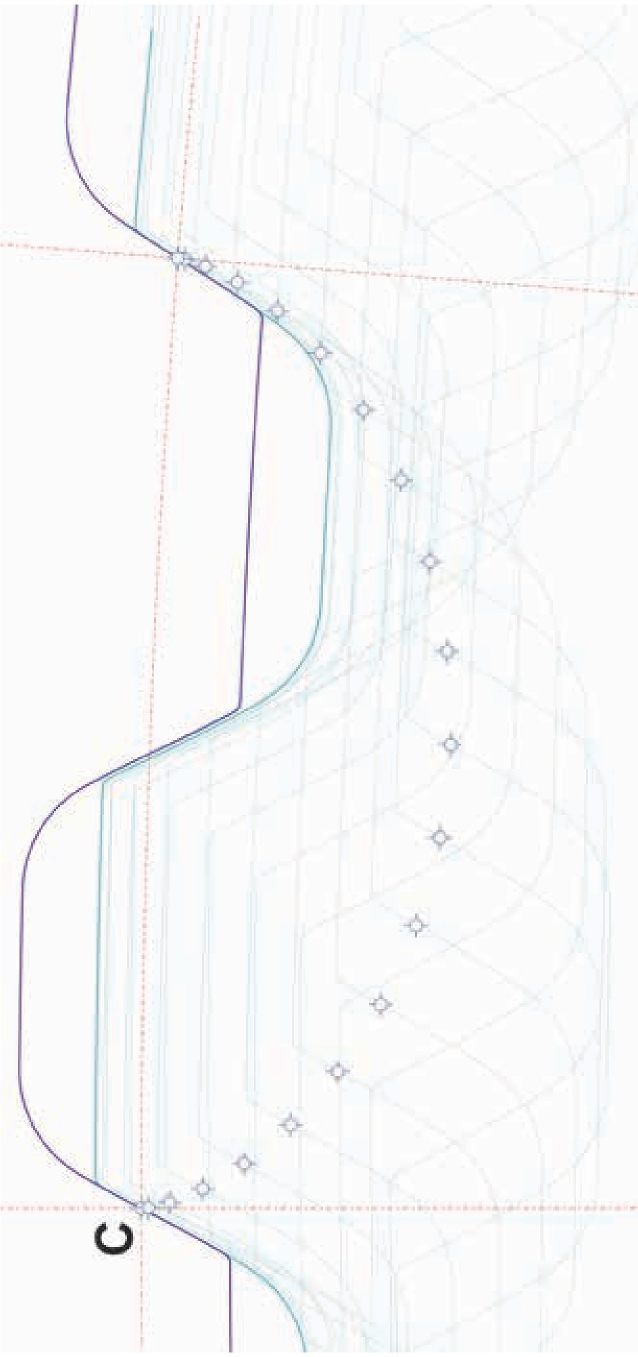

***Figure 3 :** Planetary gear movement in accordance with hypocycloid generated on the kinematic circle of the ring gear [8].*

from the eccentric. The planet gears wobble around the gear ring, that is, they reverse for one tooth in each revolution of the eccentric. The wobbling movement is in accordance with a hypocycloidal movement where the generating circle with the radius of the eccentric is rolling on the kinematic circle of the ring gear. At the same time, the planet gear kinematic circle rolls in the inner side of the ring gear kinematic circle, which is simultaneous with the rotation of the eccentric. In this way the planetary gears develop rotation superimposed on the wobble. So, the input rotation of the eccentric is transformed into the reduced output rotation of the cage with the pins according to the gear ratio in the reverse direction of the input shaft in the same axis. And the gear ring is fixed to the housing.

The eccentric driven planocentric gear train can be regarded as a simple mechanism with two links. The first link size is the radius of the eccentric and its joint indicates its position. The second one connects the eccentric with a point on the planet gear (a rigid body), e.g. the contact point. The eccentric link rotates and induces movement of the chosen point on the planet gear, which is restricted by the following rule:

$$r_v = r_p \frac{\varphi_p}{\varphi_v} \tag{2}$$

r_v and r_p are the radii of the kinematic circles of the ring gear and the planet gear, respectively. If the ring gear rotates for φ_v the planet rotates for φ_p. *Fig. 2* illustrates movement of the planet based on the rotation of the eccentric and rolling of the planet kinematic circle on the fixed ring circle.

A simple algorithm can be used to define movement of the planet based on the rotation of the eccentric and limited by Eq. (2).

- T_{p0} and T_{v0} coincide with C. P_0 is a point on the planet also coinciding with C.
- T_{p1} and T_{v1} are calculated according to Eq. (2). It is true: $p = \pi\, m = \widehat{CT_{v1}} = \widehat{CT_{p1}}$.
- Eccentric turns for φ_v to the new point O_{p1}. kkp rolls on kkv in such a way that T_{p1} coincides with T_{v1}. So, tangents and normals of kkv and kkp coincide in T_{v1}.
- The normal of the planet in this point runs through O_v and O_{p1}.
- Since the planet is a rigid body, the right leg of the angle φ_p rotates around O_{p1} in CW direction for the difference $\Delta\varphi = \varphi_v - \varphi_p$.
- The procedure is continuous, but it can be numerically calculated by any adequate number of steps.

The above procedure can be formalized. Thus, successive points on the ring gear kinematic circle T_{vi} are defined as follows:

$$x_{Tvi} = r_v cos\varphi_{vi} \text{ and } y_{Tvi} = r_v sin\varphi_{vi}. \tag{3}$$

Similarly, successive position points O_{pi} of the eccentric are

$$x_{Opi} = ecos\varphi_{vi} \text{ and } y_{Opi} = esin\varphi_{vi}. \tag{4}$$

The coordinates of the moving point Pi on the planet gear are

$$x_{Pi} = x_{Opi} + r_p cos\Delta\varphi_i \text{ and } x_{Pi} = y_{Opi} + r_p sin\Delta\varphi_i. \tag{5}$$

The eccentricity e is defined by Eq. (6):

$$e = \frac{z_v - z_p}{2} \cdot m \qquad (6)$$

The planet gear tooth movement into a new ring gear tooth space is illustrated in *Fig. 3* by 20 iterations. So, each point and planet gear position in Fig. (2) is based on successive rotations of the eccentric for 18°.

3 Gear Tooth Flank Geometry

Beside semicircular, many other gear flank geometries have been proposed for planocentric gear trains, all in search for an optimal geometry.

The involute gear shape can be used to compose a planocentric gear box. A research [9] with the goal to produce a robotic gear box in order to replace an existing cycloidal planocentric gearbox was conducted. The key point was also to be economic in manufacture due to little influence of manufacturing and assembly errors. However, due to possible gear interference, such gears exhibit rather high pressure angle α_w around 30°, and Δ_z cannot amount to less than 5, or 4 based on the condition that the numbers of teeth of pinion and ring gear are rather high, which is z_p=167 and z_v=171 in this case. This also means essentially lower gear ratio according to Eq. 1, which amounts to i^{-1} = -41.75 for the above data. Some other attempts included a rectilinear tooth profile, where a tooth is defined by two lines enclosing an angle, by an inside or outside circle, by the root circle and fillet arcs [10]. The power is transmitted by the arc at the pinion tooth tip which slides over the linear tooth part of the gear ring. The problem is that such a gear composition does not follow the law of gearing. If a high-speed rotation is required, then such a gear arrangement can develop a high noise and torque fluctuation. Yet another tooth design is trapezoidal [11], where the contact of teeth is surface-like. However, the efficiency of such gear box may be poor, due to lack of rolling, amount of sliding and (non)conformity to the law of gearing.

A proposed solution is based on S-shaped tooth flank geometry for the meshing ring gear and planetary gears of the planocentric gear box. General ideas about S-gears have been described in several papers [12-14].

The S-gear configuration has several advantages, the most important being the following:

- convex-concave contact in the vicinity of the meshing start and meshing end point;
- a low amount of sliding during meshing which is due to the curved path of contact;
- an evenly distributed flank load, which is due to similar sizes of addendums and dedendums of both meshing gears.

The other features include better lubrication due to high relative velocities and amount of rolling. In the case of internal-external gear pair some features may become less pronounced. Additionally, the path of contact is less curved, which is on behalf of smaller dedendum and addendum heights. S-gear shape is ruled by two parameters, the height parameter α_p and the curve exponent *n*. By optimizing these two parameters one can shape this type of gearing in such a way to allow the gear and planet teeth number difference to be only one. So, it is possible to design gear boxes with small diameters and high reduction ratios. The S-gearing for planocentric gear trains is illustrated in *Fig. 4*. Some time ago it was difficult for any basically better gear tooth shape to compete with the involute gear shape, which was perfected during two centuries of development. Now it is essentially easier to produce S--tooth geometry, e.g. by machining with gear hobs based on the S-rack S-gear tooth profile. However, standard gear quality reports are based on E-geometry. So, CMM programs should be adapted.

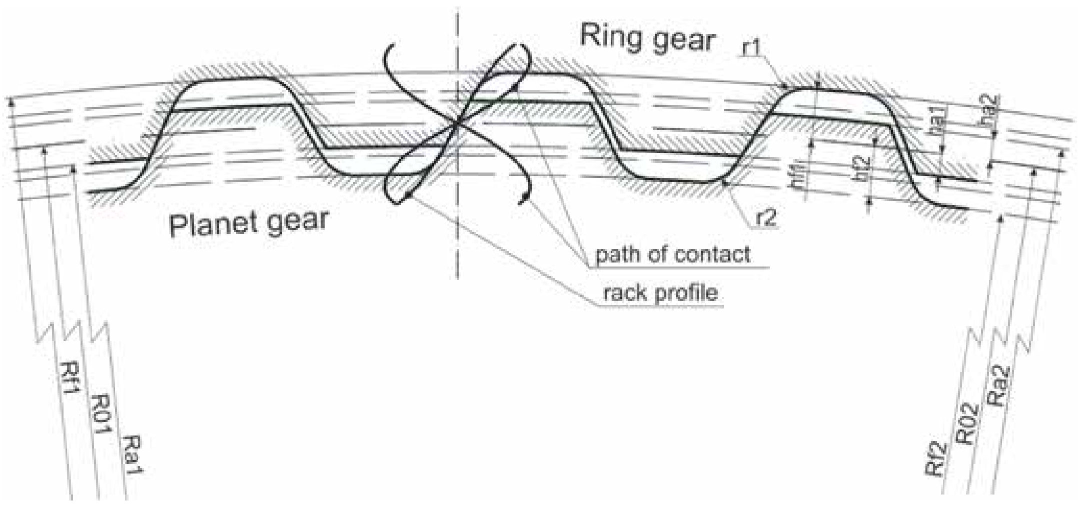

Figure 4 : *S-gear flank geometry of the planocentric gear box [15].*

4 Planocentric Gear Box Prototypes

Several prototypes were produced and assembled during the development period. These prototypes were used for testing important characteristics and to acquire knowledge in design of a succeeding gearbox. The gearbox is similar in function to those already mentioned. It contains an input shaft with eccentrics. As a motor rotates the eccentric shaft rotates two planetary gears which wobble on the ring gear. The planetary gears are positioned in such a way that they enclose 180° for the sake of symmetry. Arrangements with three or more planetary gears are possible, which would impose high manufacturing skills regarding eccentrics. A cage consists of a supporting ring and output ring (serving also as the output shaft) that are connected by pins in an interference fit. The cage is rotated by planetary gears, having appropriate holes in which connecting pins with bearings comply. The cage is fixed to the input shaft by bearings at the extremities and in a similar manner to the housing with the ring gear. In this way a compact low volume gearbox is achieved. Initial prototypes are devices with a reduction ratio of 80 (z_v=81 and z_p=80), an outer diameter of around

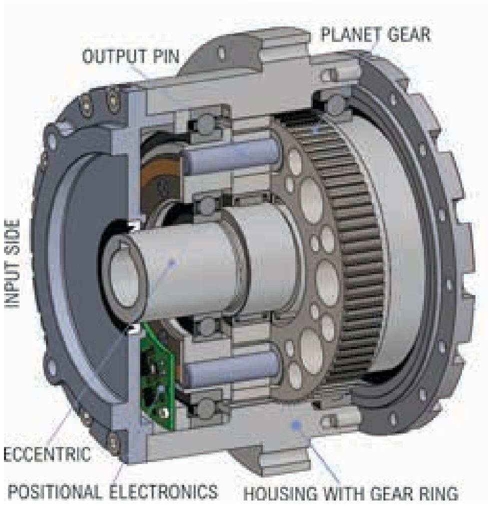

Figure 5 : *3D schematic of the planocentric gearbox (left); photo of the prototype (right).*


Figure 6 : *Ring gear (left) and a planet (right) of a specimen 01 after disassembly.*

Φ100 and having a module of 1 mm. The required maximal working torque is 120 Nm.

The device is presented in Fig. 5, 3D schematic (left) and photo (right). The gearbox from *Fig. 5* already includes an absolute output position encoder, which is also an innovative Slovenian product, namely the AksIM absolute rotary encoder made by RLS [16].

Individual components (the eccentric, the ring gear and the planets) are measured on a CMM (Computerized measurement machine) before assembly. Assembled devices were tested. The tests include backlash, hysteresis and stiffness, kinematic error, vibrations and noise, as well as durability tests. The devices were disassembled afterwards, and critical components were inspected on the CMM and optically.

Fig. 6 shows the ring gear and the planet of the specimen 01 after the conducted durability tests. The hole in the planet (which is adapted for the cage of pins with bearing bushings) is slightly worn in the circumferential direction according to the acting force on the output bearing bushings of the pins. The specimen was submitted to high torques and speeds. The planet gears were made of 42CrMo4 and the ring of 25CrMo4, all gears plasma nitrided to HV700. The gears were carefully examined by an optical microscope. The gear teeth did not show any wear or damages. Initial wear appeared in some planet teeth tips and at certain locations in teeth tips. The reason is in the meshing errors, which were discovered by measuring teeth of the planets and the ring gear with a CMM.

5 Conclusions

The paper presents a gradual development of a planocentric gearbox from starting concepts and based on S-gearing principles. The gearbox design enables high gear ratios, with the developed prototype having a reduction ratio of 80. Through gear shape optimization, design improvements, usage of CA tolerance analysis a substantial improvement of the planocentric gearbox mechanical performance was attained. The near zero backlash was accomplished, which enables usage of this product in robotic industry, i.e. robot arm joints. Severe durability tests showed no notable wear. The design is modular, so the gearbox can be purely mechanical, it can contain the absolute output encoder inside the gearbox body. It can also contain the torque flange, with electronics at the output side. It should be noted that both sensors should be provided if such a gearbox is intended for collaborative robots or adaptive control robots. The robotic companies do not sell such gearboxes, they are for internal use only. So, such a gearbox becomes even more interesting. Incorporation of a servomotor is also being considered.

Beside the gearbox with the reduction ratio 80, a smaller gearbox with a ratio of 48 and a bigger one with a ratio of 120 were designed. So, a gearbox family in a range of output torques from 40 Nm up to 400 Nm becomes available. Technological procedures for serial manufacturing are already being prepared and optimized. The project with the acronym SGU – S-Gearbox Ultra was therefore successfully brought to the end. Nevertheless, many tasks are still in progress, e.g. a setup of the serial mechanical and electronics production and serial assembly, automating calibration procedures, development of self-aware monitoring and many other.

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Razvoj inteligentnega planocentričnega prenosnika za robotsko industrijo

Razširjeni povzetek:

Za planocentrične zobniške prenosnike sta značilna visoka redukcija vhodne rotacijske hitrosti in veliko povečanje izhodnega navora v najmanjšem mogočem volumnu, zato so zanimivi za visokotehnološko industrijo. Predstavljena rešitev je zasnovana na S-obliki bokov zob in posebej fokusirana na robotsko industrijo. Zahtevane mehanske karakteristike prenosnika pomenijo striktne omejitve za sestavljeni proizvod, npr. največjo zračnost pod 1 kotno minuto oz. blizu nične zračnosti. Dodatna kvaliteta predstavljenega prenosnika pa je senzorika, ki je vgrajena opcijsko, na modularen način, zgolj z dodatnimi elementi. Tako je lahko prenosnik zgolj mehanski, lahko pa vsebuje absolutni rotacijski enkoder AksIM 2 firme RLS, ki posreduje točno izhodno pozicijo. Dodatno pa se lahko prigradi senzor izhodnega navora, ki je zasnovan kot posebna prirobnica z dovolj veliko torzijsko deformacijo, ki omogoča korekten signal. V ta namen je uporabljen ustrezen sistem merilnih lističev. Poznavanje lege in navora pa je podlaga za uporabo prenosnikov v sklepih rok v t. i. sodelujočih robotih ali pri adaptivnem krmiljenju.

Prenosnik iz tega članka je bil v osnovi zasnovan pred skoraj 30 leti z drugačnim ozobjem, t. i. paličnim ozobjem s cilindričnimi vdolbinami in ustreznimi izdolbinami. V tem prispevku je predstavljeno kinematsko delovanje bistveno izboljšanega prenosnika z drugačnim S-ozobjem in strukturo. Za namene testiranja zračnosti, histereze, kinematske napake, vibracij in obremenitvenih testov različnih prenosnikov – od tistih v razvoju do raznih na tržišču, npr. Spinea, Harmonic drive itd. – je bilo zgrajeno sofisticirano preizkuševališče. Rezultati testiranj so pripomogli k hitri konvergenci v razvoju. Vse bistvene komponente prototipov so bile izmerjene na CMM pred uporabo in po končanih trajnostnih preizkusih. Komponente so bile pregledane tudi na mikroskopu, kjer so se ugotavljale morebitne poškodbe.

Nujna pa je simulacija obnašanja prenosnika na osnovi toleranc. Sistem KissSoft je bil uporabljen za analizo vplivov toleranc, za kontaktno analizo in za vpliv toleranc ležajev in nosilcev. Ta analiza je pokazala na potrebo po parjenju zobnikov, tj. venca in planetnikov iz diametralnih tolerančnih mej – zgornja/spodnja ali spodnja/zgornja, po debelejših zobeh in povečani medosni razdalji. Kontaktna analiza razkriva potrebo po toplotno obdelanih legiranih jeklih. Vpliv kontaktnega tlaka na možno interferenco pa je zanemarljiv. Vpliv toleranc ležajev in ležišč ekscentra na skrajne lege planetnikov ob predvideni nominalni obremenitvi prenosnika pokaže na možne interference, kar vodi do konstrukcije zob z ustreznimi zaokrožitvami.

V tem primeru se je za KissSoftovo analizo uporabljala modificirana geometrija bokov zob, ki je omogočala računske postopke na osnovi korekcije evolventnih bokov v S-geometrijo. Zaradi omejene višine vrhov in vznožij je bila ta analiza dovolj natančna.

Ključne besede:

planocentrični prenosnik, zračnost, analiza toleranc, S-ozobje, senzor navora, senzor zaznavanja lege, preizkuševališče

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