Towards Intelligent Planocentric Gear Train for Robotic Industry – part 2

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

6 Testing Rig and Experimental Results

A special testing rig was manufactured for testing of this type and similar devices, *Fig. 7*. The tests comprise backlash and hysteresis measurement, kinematic error, vibration and noise and durability tests. Some tests are acquired in automatic and some in manual operation.

Since the gearbox is intended for precision industry and robotics, its backlash and stiffness characteristic become crucial. The characteristic should be symmetric, regardless of any initial position of the planets and rotation direction, and the backlash in very narrow limits, < 1 arcmin.

The stiffness characteristic of the specimen 04 exhibits the backlash of 9 arcmin and stiffness characteristic which becomes considerably stiffer with increasing load. Such results are unacceptable.

Doc. dr. Gorazd Hlebanja, univ. dipl. inž., University of Novo Mesto, FME, Slovenia and Podkrižnik, d. o. o., Nazarje, Slovenia;
Miha Erjavec, univ. dipl. inž., **dr. Luka Knez**, univ. dipl. inž., **dr. Simon Kulovec**, univ. dipl. inž., vsi Podkrižnik, d. o. o., Nazarje, Slovenia;
Prof. em. dr. Jože Hlebanja, univ. dipl. inž., University of Ljubljana, FME, Slovenia

However, this prototype was not optimized yet. So, keeping these results in mind, the gearbox was redesigned, preloaded bearings were used, gears designed with near zero backlash and crucial components with proper tolerances.

Stiffness of the specimen 04 in the zone 3–50 Nm was 10 Nm/arcmin, in the zone 50–100 Nm it was 30 Nm/arcmin and 16 Nm/arcmin in the zone 3–100 Nm. Actually, this gear train exhibits higher values of stiffness, since the working torque limit amounts to 120 Nm. A similar gear train of *Spinea* (Spinea TS110) with a gear ratio of 89 has a stiffness of 22 Nm/arcmin. Sumitomo Cyclogear drive A15 with a ratio of 89 is a bit stronger drive with stiffness values as follows: 15 Nm in 3–50 %, 28 Nm/arcmin in 50–100 %, and 20 Nm/arcmin in the segment 3–100 %. *Fig. 8* shows the stiffness characteristics of the specimen 04 and 05. The first characteristic was measured classically by invoking the torque by applying loads at a specified distance. The second one is measured automatically in a continuous manner, where the DeweSoft data acquisition system DeWe 43a was used. The specimens 05 and 06 were already redesigned as stated above, and the characteristics are improved substantially. So, the backlash (specimen 06) is below 1 arcmin and the stiffness curve shows an average value of 10 Nm/arcmin.

The kinematic error of a gear train is defined as a deviation of the actual angular position from the theoretical angular position:

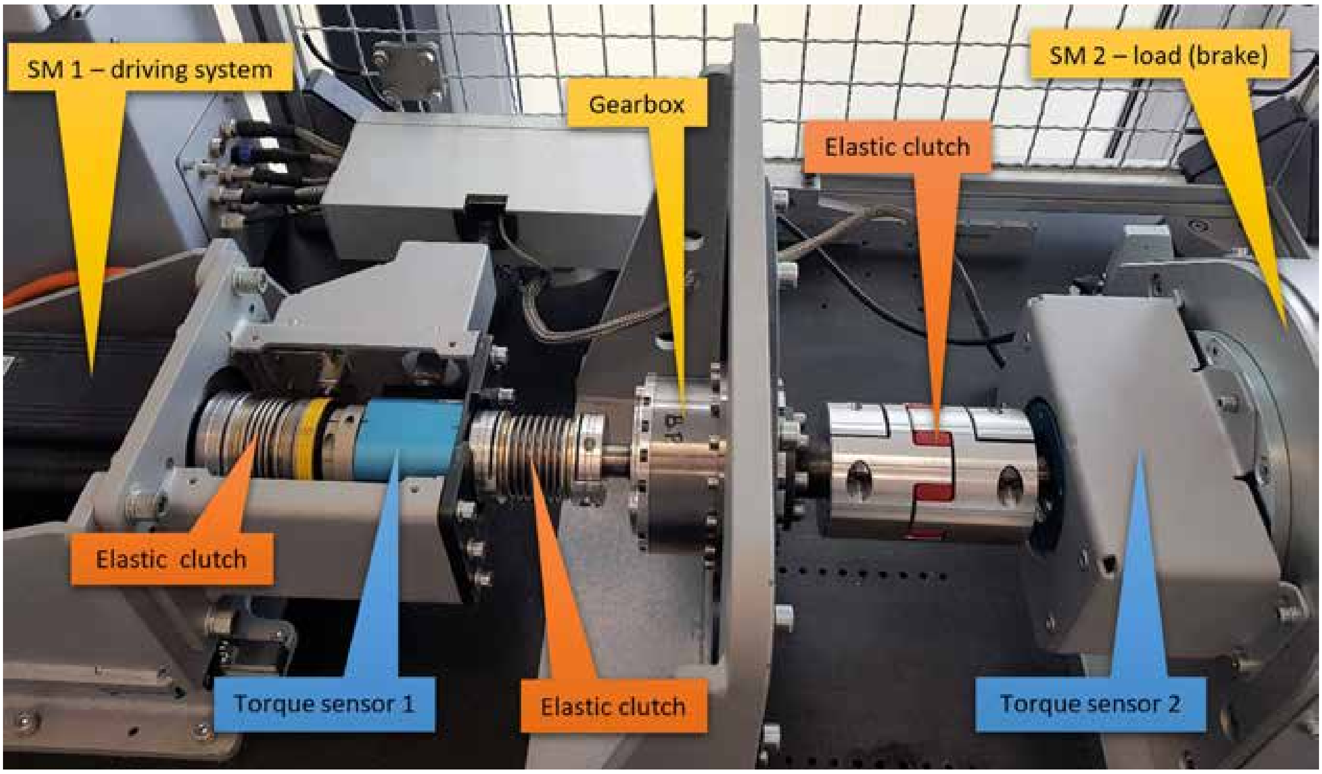

Figure 7 : *Testing rig for acquiring various characteristics of planetary gear trains (SM –servomotor).*

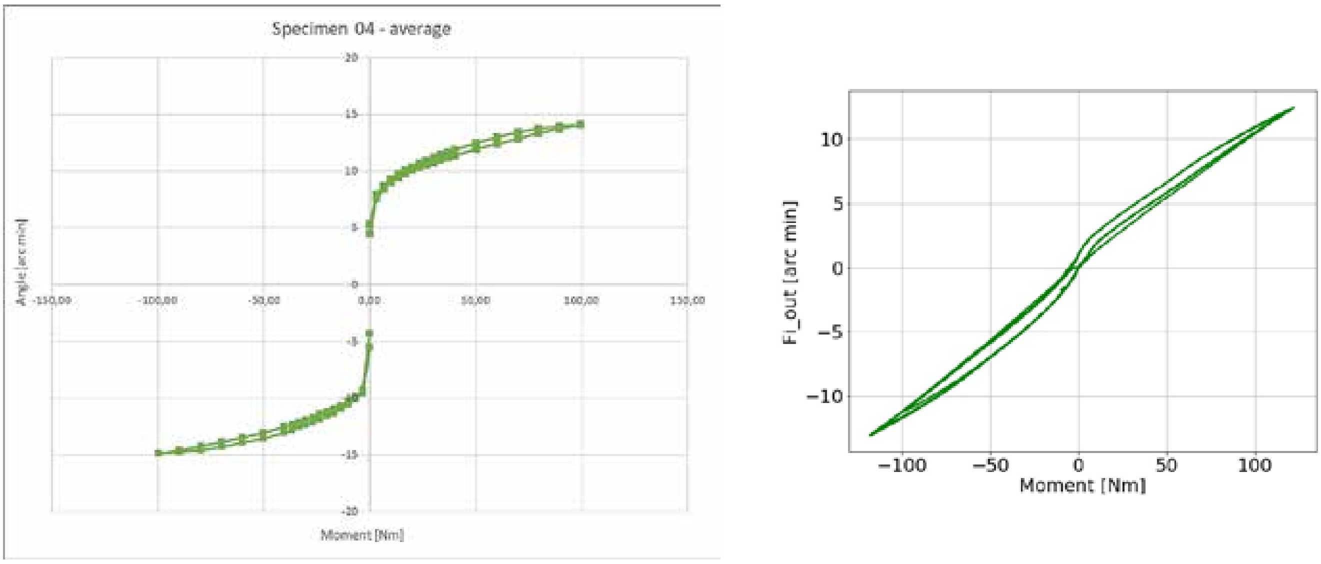

Figure 8 : *Average of backlash and stiffness measurement of specimens 04 (left) and 06 (right).*

$$\Delta\varphi = |i| \cdot \varphi_{inp} - \varphi_{out} \quad (7)$$

Rotation of the input shaft ($_{inp}$) is measured with a 16-bit incremental encoder, whereas the output shaft ($_{out}$) rotation is measured by a built in absolute optical encoder with a high resolution of 20-bits. The optical encoder and the reading head are mounted with some tolerance, which reflects in a sinusoid carrying the actual error signal. Several measurements of the kinematic error of the specimen 04, *Fig. 9*, yield a maximal error limit of 6 arcmin and about 2 arcmin for the specimen 06.

In general, one can conclude that the first prototypes did not meet all prescribed requirements yet. Thus it was necessary to improve the design and the quality of manufacturing in such a way that all measures are within prescribed tolerances. So, all components should be inspected by the CM machine. Regarding gearing, the nominal circumferential backlash should be as low as possible, i.e. less

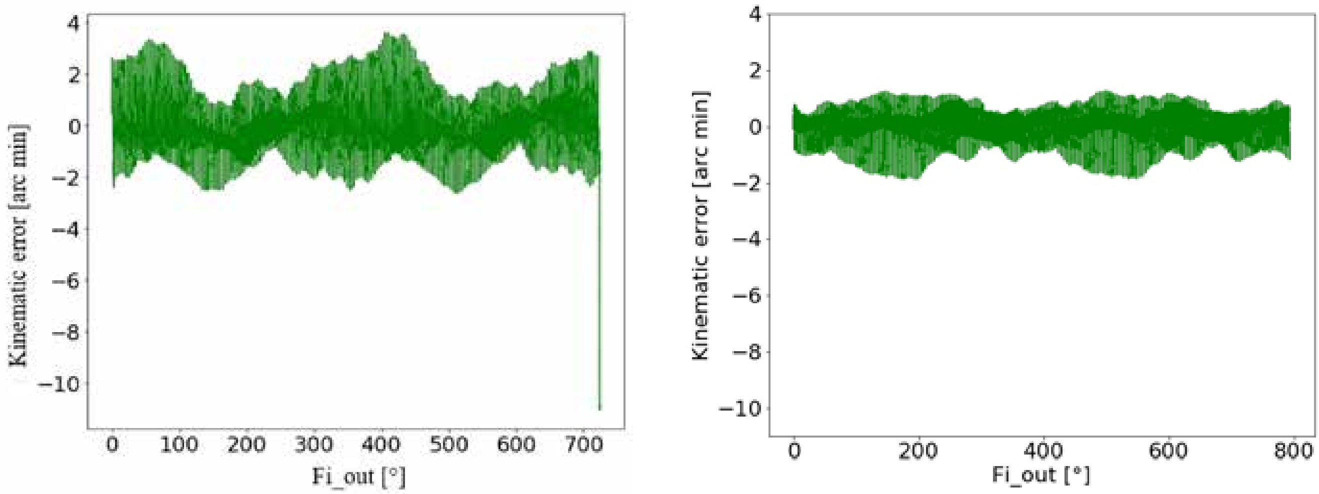

***Figure 9 :** Kinematic error of a specimens 04 (left) and 06 (right).*

than 10 μm, having a manufacturing tolerance in the range of 5 μm. The circumferential arc of 10 μm for the radius 40.5 mm gives an angle of 0.85 arcmin for geometrically precise circumstances.

7 Influences of Tolerances

This imposes the necessity to analyze influences of tolerances in varying circumstances. The aim of such an analysis is to disclose additional measures which can enable a convergence of the design towards requirements. As a primary tool for this task, the KissSoft system was employed. KissSoft is a software for an effective, high-quality tool for calculating machine elements, reviewing these calculations, determining component strength, and documenting safety factors and product life parameters, incorporating currently valid standards (DIN, ISO, AGMA).

However, KissSoft uses the prevailing involute gear geometry. So, the first aim is to adapt geometry in such a way to reflect S-gear geometry. The current KissSoft software edition includes the possibility of a progressive profile modification (User manual, p. 343 [17]), which can be used as a modification in the addendum and the dedendum of a gear tooth, and is defined as follows:

$$\Delta_{ad} = 2 \cdot C_{ad} \cdot \left(\frac{d-d_k}{d_t-d_k}\right)^{\frac{f_{ad}}{5}} \text{ and } \Delta_{dd} = 2 \cdot C_{dd} \cdot \left(\frac{d-d_k}{d_v-d_k}\right)^{\frac{f_{dd}}{5}} \tag{8}$$

Δ_{ad} and Δ_{dd} stand for a profile modification function in addendum and dedendum. C_{ad} and C_{dd} are modifying tip relief (or corresponding active dedendum flank modification) and f_{ad} and f_{dd} power coefficients. If a coefficient amounts to 5, the relief is linear. d_t, d_v, d_k, and d, are diameters of the tip circle, dedendum circle, kinematic circle and current circle respectively. One can adapt the involute flank addendum and dedendum to S-gear flank. Such a modification is justified since addendum and dedendum heights are rather small, between 0.2 and

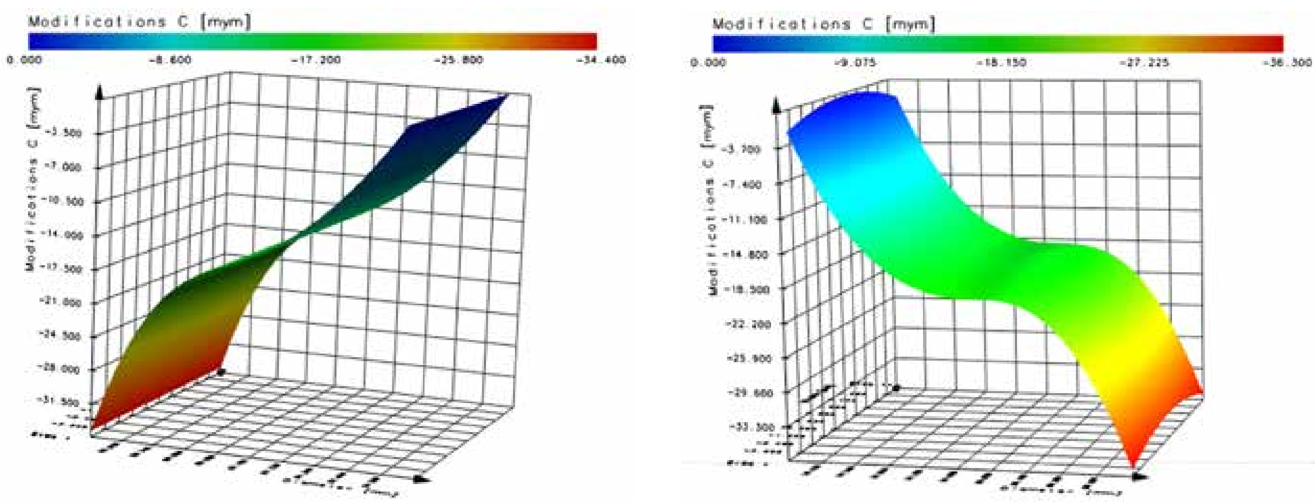

***Figure 10 :** S-gear data as they deviate from involute gearing (planet gear – left, ring gear – right).*

0.25 m. The modifying parameters C_{ad}, C_{dd}, f_{ad} and f_{dd} were computed for the given S-flanks (for the ring gear and the planet). The resulting deviations are represented in *Fig. 10*. Thus, the use of KissSoft for a non-involute geometry became possible.

The backlash should be kept much below 1 arcmin = 0.016o, to prevent the planned function of the gear box. The aim is also to create a method of manufacturing and assembly, which would ensure the required precision and at the same time the tooth thickness tolerance would not be lower than class h25, DIN3967 and the axis distance tolerance lesser than js6 (±0.003). The aspects, which are described in detail in [6], are as follows:

1. the analysis of tolerance influences,
2. the contact analysis, and
3. the influence of bearing tolerances and carriers on the position of the gear train.

Ad 1: A theoretical analysis of various tolerance combinations of the gearing and their effects was conducted. The analysis showed that the backlash for the eccentricity (axis distance) 0.500 js6 and the tooth thickness allowance DIN3967 h25 can be below the required limit only when both teeth are in the upper limit. So, the nominal axis distance was increased to 0.520 js6 and at the same time the tooth thickness tolerance was shifted for 0.020 mm towards thicker teeth for both, the planets and the ring gear.

If a planet gear tooth thickness is from the upper part and the ring gear from the lower part of the tolerance grade, the backlash is within the prescribed limit. The same is true for the reversed situation. A logical conclusion is that if both, the planet and the ring gear, tooth thicknesses are in the upper part of the tolerance zone, an advantageous result is expected. Results for both gears in the lower tolerance part are unsatisfactory. Therefore, gears should be paired according to their tolerance zone.

Ad 2: The load considered in contact analyses was a prescribed working torque of 120 Nm at the output shaft. Several simulations were carried out with varying the axis distance and tooth thickness deviations. In this context, a transmission error as a function of the rotation angle of the planet gear, a system stiffness in the contact zone and contact pressure were simulated. And finally, a simulation of meshing of a planet gear with the ring gear was provided. The aim of the contact analysis is to discover possible interferences due to changes in the axis distance and tooth thickness values. The transmission error is always in the range of less than 1 arcsec. The contact stiffness is in the range from 700 to 720 N/µm. The contact pressure is in the range between 1100 and 1170 N/mm^2, which implies usage of heat-treated alloy steels. Furthermore, the meshing scheme shows that around five teeth pairs are always in contact and no interferences resulted from these simulations.

Ad 3: The influence of tolerances in bearings and assembly of connected shafts was studied. These tolerances can have a negative influence on the function of the gear box during operation under load (120 Nm). So, a model clarifying deviations of a planet from its nominal position was developed in KissSoft. The nominal position in the model amounts to 0 mm, which is due to easier simulation. The situation illustrated in *Fig. 11* (above) shows the case where mean tolerance values are assumed for all bearing positions and clearances. The axis distance changes for 0.007 mm towards increasing the backlash. So, if the eccentric is produced on the lowest tolerance limit 0.517 mm, the resulting eccentric link radius becomes 0.510 mm, which increases the backlash. Assuming a possibility that bearing locations (shafts and housings bores) are made in such a way that these increase the deviation of the axis distance, as illustrated in Fig. 11 (below), the axis distance deviation amounts to 0.015 mm and the axis distance 0.505 mm. The increase of the backlash becomes even bigger.

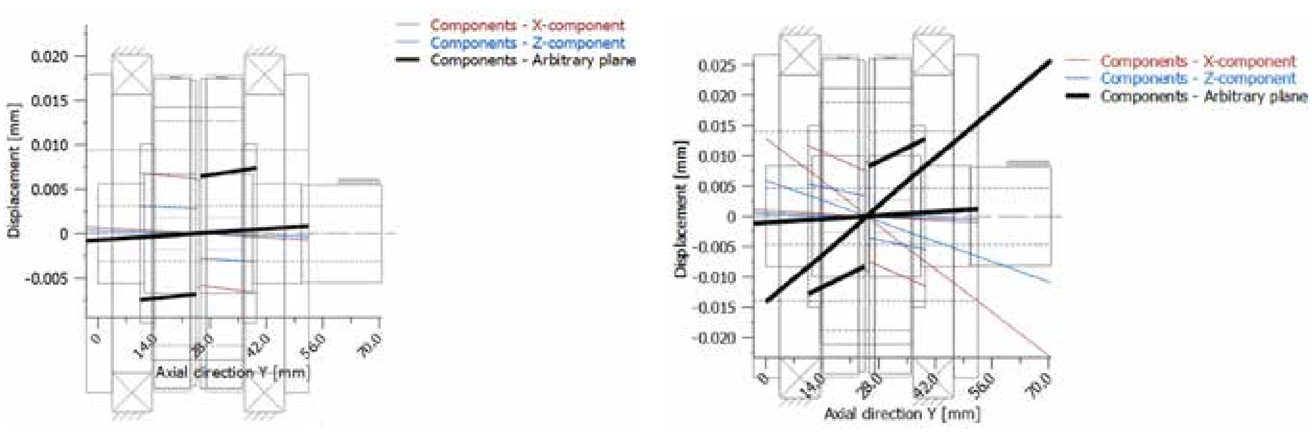

Figure 11 : *Radial deviation of the planet gear position with nominal (above) and increased (below) axis distance deviations.*

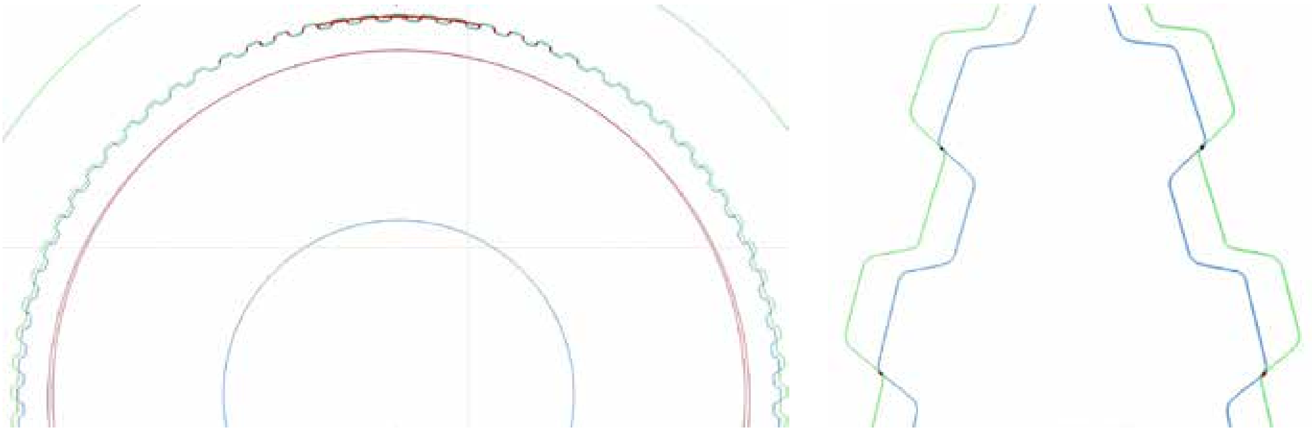

Figure 12 : *Meshing gear and planet (left) details showing collisions (right).*

Such deviations with an additional displacement of the prescribed eccentricity possible interference since the tolerances are narrow. So, the above analysis appears to be important in the context of functionality and detection of possible collisions between the ring gear and planet gears teeth tips. Fig. 12 (left) shows the ring gear and the planet with the already described S-gear geometry (z_p = 80, z_v = 81). However, *Fig. 12* (right) clearly shows collisions between the planet and ring gear teeth tips which are located in the zones around –70° and +70° from the pitch point. It is necessary to avoid such interferences, and since a near zero backlash can only be achieved with a bit bigger eccentricity (0.520 mm) and with very narrow tooth tolerances, the tooth tip rounding must be increased, which was accomplished with new prototypes.

8 Torque Sensor

AksIm 2 absolute rotary encoder was installed in the prototypes, whereas, the torque sensor was developed in cooperation with the Josef Stefan Institute. So, a special flange was designed and calculated by FEM to adjust proper deformation enabling sensorial output. The strain-gauge measurement principle was chosen with strain-gauges 1-XY-41-1.5/350, produced by HBM. The prototype flange was mounted on the gearbox. The gearbox input side is fixed and mounted in the testing rig to enable sensor calibration (*Fig. 13*). The sensor is equipped with provisional electronics. When prepared for a serial production, such electronics will be mounted inside the flange.

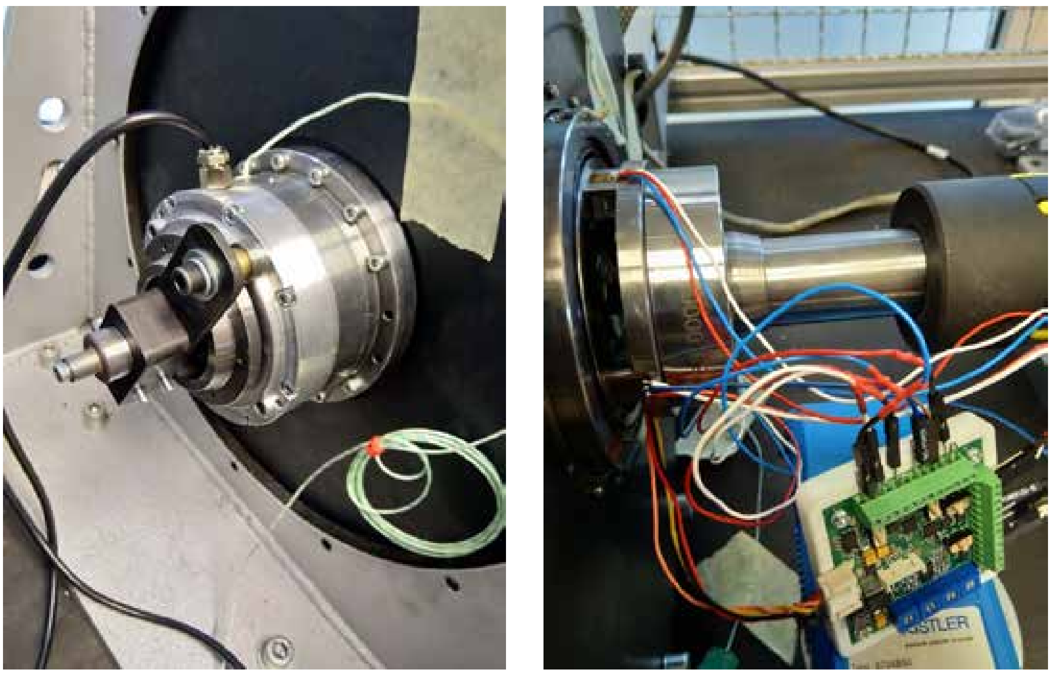

Figure 13 : *Fixed gearbox input side (left); output torque sensor with provisional electronics (right).*

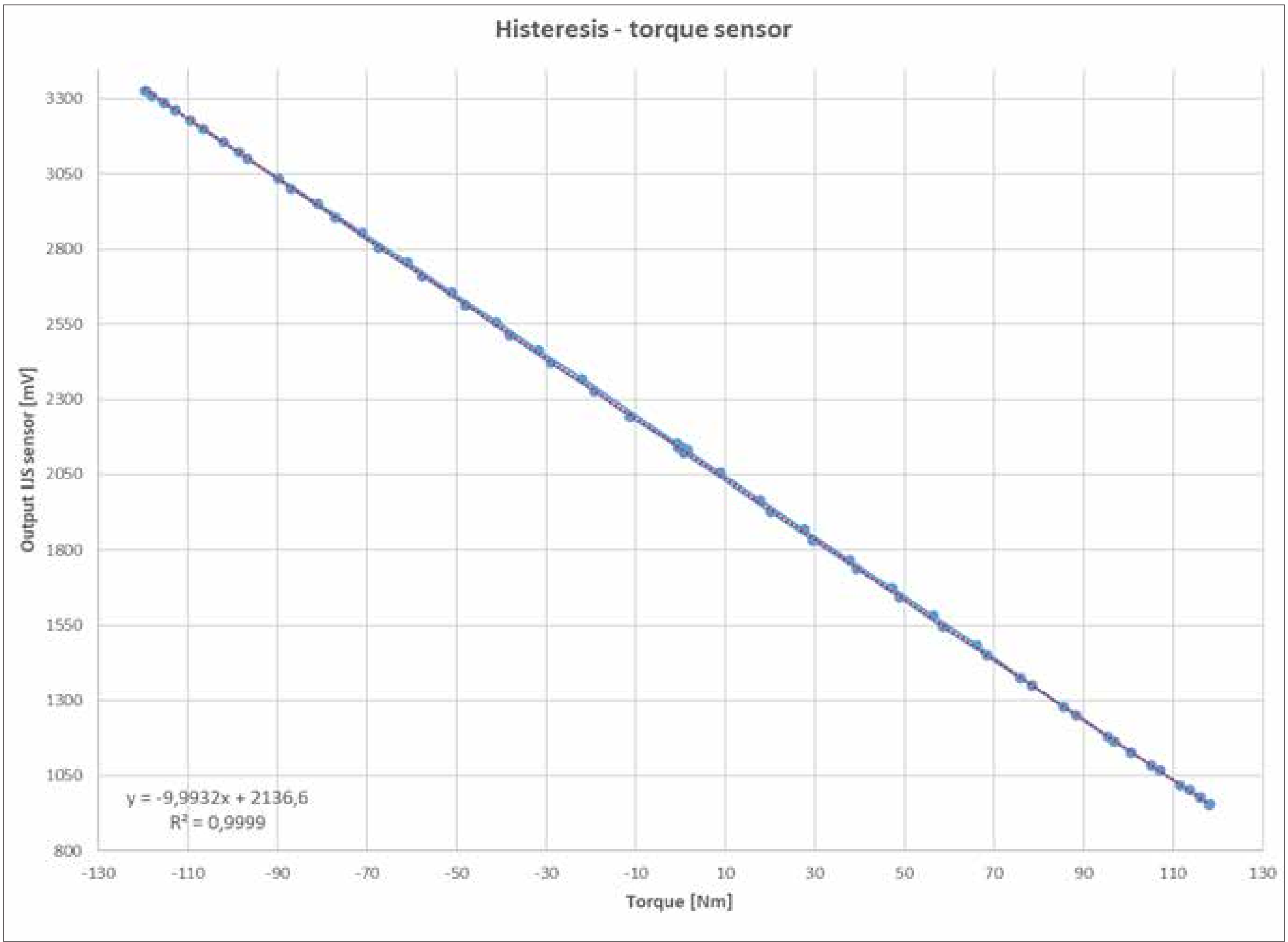

Figure 14 : *Hysteresis curve of the torque sensor.*

Since the torque sensor is a rotational part, the cables need to be fed to the stationary side by slip rings. The resulting torque – voltage characteristic is shown in *Fig. 14*.

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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