

# Development of a Planocentric Gear Box Based on S-Gear Geometry

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## Abstract

The planocentric gear boxes, devices for large transmission ratios, primarily applied in automation and mechatronic industry, decrease rotational movement and increase torque to the largest possible extent in the smallest available space. The paper discusses several planocentric gear boxes in common use in this context. Several solutions are patented. Due to industrial interest for such devices authors propose another solution which improves some characteristics of the existing devices.

## 1 Introduction

Planocentric gear boxes are in technical use for many decades due to their main characteristic, which is reduction of rotational speed and accordingly increased torque in the smallest available volume. This type of device is used in aircraft, marine, and robotics fields. The expected efficiency may be as high as 90 % or more and it can have ratio of even 120:1 in a single step configuration. Such arrangements are also described in references [1], [2]. The available industrial

solutions include Cyclo gear box [3], Spinea drives [4], newly developed gear boxes of Sumitomo and Jtekt Corporation [5], [6], [7], etc. The expected characteristics are low hysteresis for accurate positioning, low lost motion, compact design, high torsional stiffness, low inertia, high efficiency, overload capacity, ease of mounting and easy assembly, lifetime lubrication.

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 and the planet gears is small, preferably amounting one, Eq. (1). The planet gears are mounted on an eccentric shaft. Bearings separate the planet gears from the eccentric shaft. The planet gears wobble around the gear ring, that is, they reverse for one tooth each revolution of the eccentric. The wobbling movement is in accordance with a hypocycloid superimposed on the pitch circle of the ring gear, Eq. (2). And at the same time, the planetary gear pitch circle rolls on the inner side of the ring gear pitch circle, which is simultaneous with the rotation of the eccentric, illustrated by  $Z_P$  and  $Z_B$  in Fig. 1, right side. In this way the planetary gears develop rotation superimposed on the wobble. The rotation of the planetary gears is output rotation in reverse direction of the input shaft. The planetary gears transmit rotation to the output shaft by a pin coupling. The gear ring is fixed to the housing. The planocentric gear box arrangement is illustrated in Fig. 1.

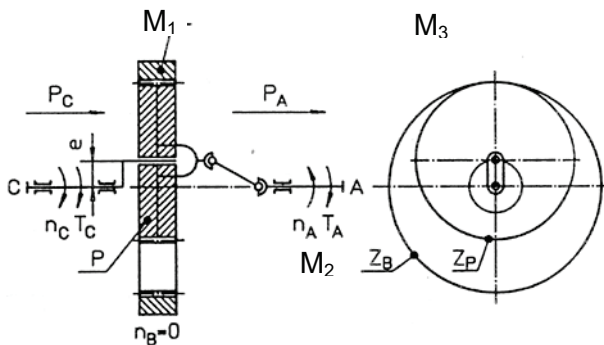


Fig. 1: Planocentric gear box arrangement

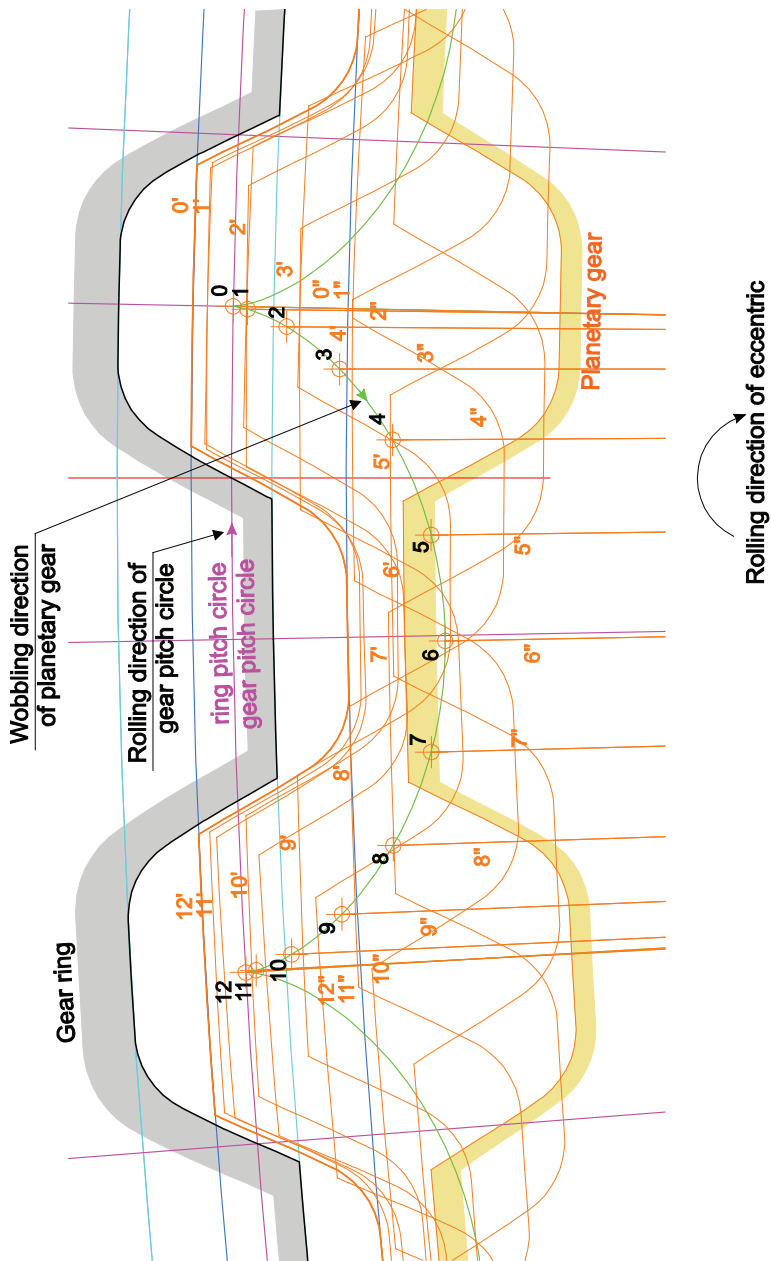


Fig. 2: Planetary gear movement in accordance with hypocycloid

The gear ratio equation is:

$$i = \frac{Z_P}{Z_B - Z_P} \quad (1)$$

The hypocycloid is defined by a 2D parametric Eq. (2)

$$x(t) = (r_r - e) \cos(t) + e \cos\left(\frac{t(r_r - e)}{e}\right) \quad (2)$$

$$y(t) = (r_r - e) \sin(t) + e \sin\left(\frac{t(r_r - e)}{e}\right)$$

where  $r_r/e \in \mathbb{Q}$  and  $t = [0, 4\pi e]$

Fig. 2 illustrates motion of the planetary gear as the eccentric rotates. It can be regarded as a two bar mechanism, where the first bar rotates and the tip of the other progresses along the hypocycloid. The points 0 to 12 on the hypocycloid correspond to the individual states of the mechanism, when the eccentric rotated for  $30^\circ$  in the clockwise direction each time. The corresponding marks  $x'$  designate the tooth tip at each state and the marks  $x''$  designate the corresponding linking bar position. The gearing contact point in each state can be tracked approximately at the eccentric rotation angle.

## 2 Early development

Potential advantages of planocentric gear boxes, namely a high speed reduction ratio combined with high output torque in a relatively small volume, were a driving force for a new development. So, the modified lantern gears were developed [8] and patented [9]. The internal gear pair during meshing is illustrated in Fig. 3, clearly indicating the value of eccentricity. The teeth of the gear ring are designed as semi-circular extremities, whereas the planetary gears are designed with corresponding semi-circular spaces, enlarged for a tolerance. Design aims were focused in automatic 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 various modules, down to  $m = 0,5$  mm.

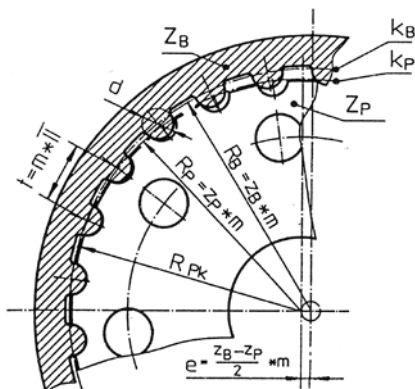


Fig. 3: Gear ring and planet gear during meshing

A variant of the assembly is illustrated in Fig. 4. Gear boxes of this type have been produced in various sizes and configurations to serve primarily wood cutting automation machines. Materials for gears included among others self-lubricating plastics for planet gears, despite through hardened steel with tensile strength of  $900 \text{ N/mm}^2$  prevailed. The specific load  $k_{lim}$  for continuous working conditions in the latter case amounted to  $125 \text{ N/mm}^2$ , which implied the output torque of 100 Nm for the illustrated gear box with the gear ratio 100 and the module 1 mm.

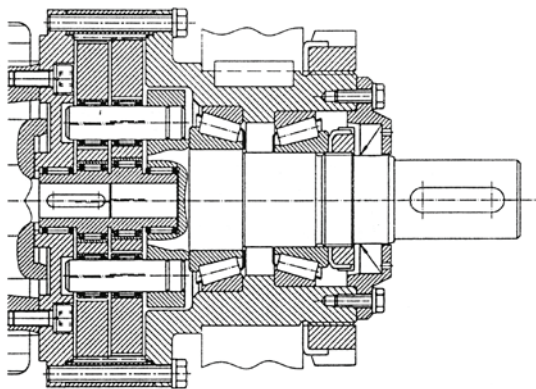


Fig. 4: Assembly of a robust planocentric lantern gear box

### 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 recent research was conducted [10] with the goal to produce a robotic gear box in order to replace an existing cycloidal planocentric gearbox. The key point of the new model was also regarded as to be economic to manufacture due to little influence of a manufacturing and assembly errors.

However, due to possible gear interference, such gears exhibit rather high pressure angle  $\alpha_p$  around  $30^\circ$ , and  $\Delta z$  cannot amount to less than 5, or 4 (if the numbers of teeth of pinion and ring gear are rather high – 167 and 171 in this case). This also means essentially lower gear ratio (Eq. 1),  $i=41,75$  in this case.

Another attempt was made with 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 [11]. 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 gear law. If there is a requirement towards high speed then such a gear arrangement can develop noise and torque fluctuation.

Yet another tooth design is trapezoidal [12], 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.

### 4 S-shaped gear geometry

A proposed solution is based on S-shaped tooth flank geometry for the ring gear and planetary gears of the planocentric gear box. The common characteristics of the S-gearings have been described in several papers, e.g. [13], [14], [15].

This gear box is similar in function to those already mentioned. It contains an input shaft with eccentrics. As a motor rotates the shaft the eccentric rotates two planetary gears which wobble on the ring gear.

The planetary gears are positioned in such a way that they enclose  $180^\circ$  for the sake of symmetry. Arrangements with three or more planetary gears are possible, which imposes a high manufacturing skill with regard to eccentrics. For each rotation of the eccentric the planetary gear proceeds for one tooth on the ring gear in the opposite direction. 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 gear box is achieved. The scheme is presented on Fig. 5.

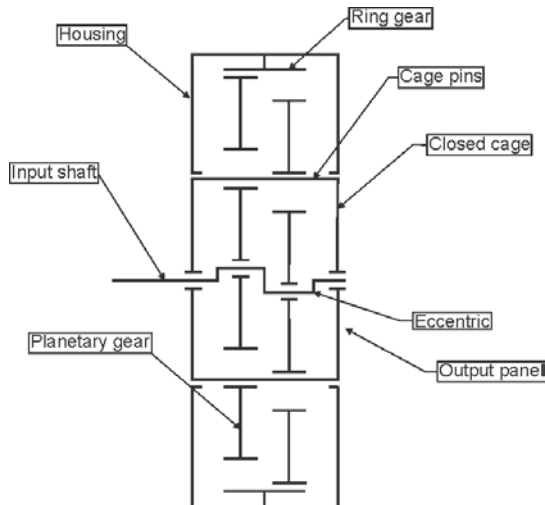


Fig. 5: Scheme of double cage, double bearing planocentric gear box

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

1. Convex-concave contact in the vicinity of the meshing start and meshing end point;
2. Low amount of sliding during meshing which is due to the curved path of contact;
3. 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. The derived geometry focused on gear ratio and volume, the requirement was to design a prototype gear box with less than  $\phi 100$  and ratio  $i=80$ .

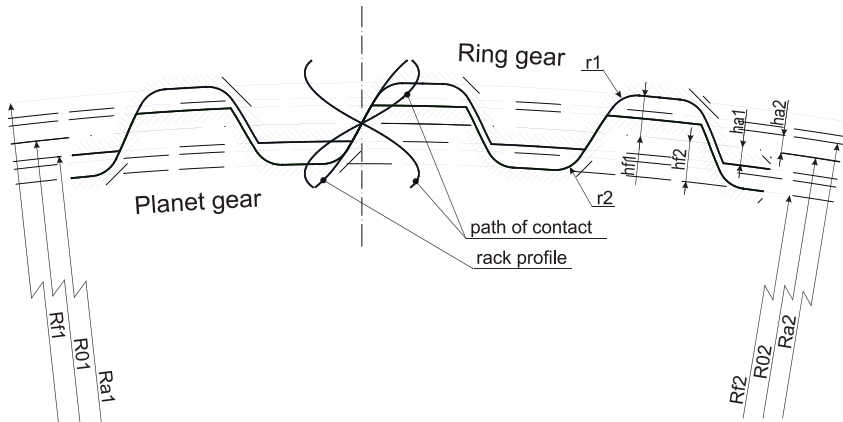


Fig. 6: S-geometry for the planocentric gear box

### 5 Prototypes

Several prototypes have been produced according to the prepared 3D models and technical documentation. Fig. 7 reveals the exploded view of the 3D assembly and Fig. 8 the prototype. The chosen gear materials are displayed in Table 1.

Tab. 1: Gear materials

Prototype	Planetary gear	Ring gear
1	Steel	Steel
2	PEEK	Steel
3	Steel	AMPCO bronze



The prototype planocentric S-gear boxes were submitted to experimental verification. The gear boxes have been run several times for a prescribed time in stepwise increased load levels. The input speed was controlled by a frequency inverter and the output torque was mechanically adjustable by a system of pulleys. The temperature was measured during testing as well. Temperature increased for 15,8 K after two hours of loaded operation. Since the level of vibrations was above admissible level, the gear boxes were disassembled after testing in order to measure components by a coordinate measuring machine and to observe gears by a microscope (Fig. 9). In this way necessary adjustments could be located.

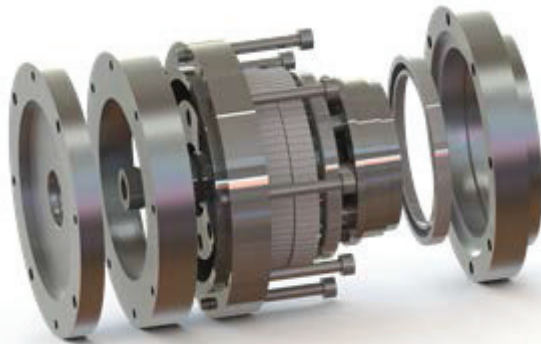


Fig. 7: 3D model of the planocentric gear box



Fig. 8: Prototype of the planocentric gear box

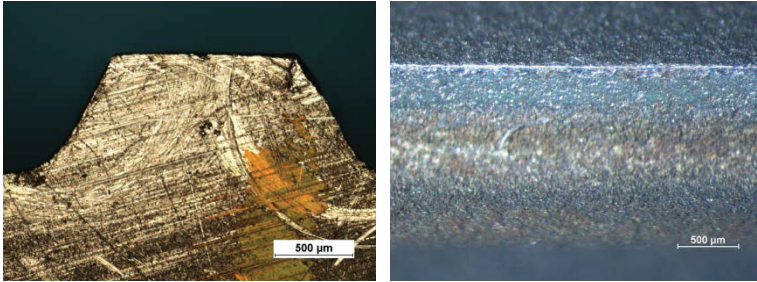


Fig. 9: Profile and flank view captured by an optical microscope

## 6 Conclusion

Many new proposals have been made for an optimal gear tooth shape of the planocentric gear box. One can also observe several new patents, which imply technical interest in such devices.

Authors of this paper suggest the S-gear tooth shape. Corresponding prototypes with several material combinations have been made of e.g. hardened steel, Peek (Polyether ether ketone) with carbon fibers and others. PEEK is material specially fit for gears, having increased endurance and temperature stability. Lifetime lubrication is also important, so there are investigated possibilities to use nanotubes of  $\text{MoS}_2$ .

The mentioned prototypes have proven functionality by preliminary testing. First load tests revealed that the developed gear box is capable of transmitting load. Since the gear box is intended for automation and robotic industry it is necessary to conform to rather strict rules. That is why measuring components after loading by a CMM and microscopic observation is necessary. In this way possible excessive wear could be recognized and the design can be optimized successfully. Since there is a need in the development stage and later in the production phase an advanced test stand capable of measuring more detailed data is also in preparation.

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