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TWO-STAGE GEAR WITH AN INNER MOVING WHEEL

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Abstract: *This paper describes a two-stage gear with several output shafts having different gear ratios and torque. The paper describes also their power drive possibilities*.
Key words: *reducer, two-stage speed, different outputs, power drive*

1. INTRODUCTION

Within a research activity in the field of the so- called high-accuracy gears carried out at the Department of Mechanics and Machine Parts of the Faculty of Manufacturing Technologies and the Department of Designing, Transport and Logistics of the Faculty of Mechanical Engineering of the TU in Košice, a design of a fundamental solution of a two-stage multi-output gears, which eliminates some unfavourable factors of the gears of this type, has been elaborated. The paper also presents the design of a two-stage multi-output gear with the kinematics description and gear ratios determination.

2. TWO-STAGE GEAR WITH ONE INNER MOVING WHEEL

The designed gear represents the solution of a two-stage gear with one inner moving wheel having two or more gear rims (fig. 1a,b).

A low gear is created by the rim gearing z_2 of the wheel *2* meshing with the inner gearing of the gearbox z_1.

The transmission of the torque moment to an output shaft by the alignment with an input shaft is then transmitted by the help of a higher gear which is also created by the gear with the inner and outer gearing having certain parameters, but with an equal eccentricity. This two-stage gear can be designed as a one-output or a multi-output gear, i.e. with several output shafts; all these gears, however, are two- stage gears. The number of output shafts is determined by the number of gear rims that are properly set on the inner eccentrically moving wheel. The number of outputs, or output shafts can be determined as follows:

$n_{vi} = n_{2i} - 1$,

where

n_{vi} - the number of output gears – shafts,

n_{vi} - the number of gear rims on the inner wheel *2*.

Each gear rim on the inner wheel meshes independently with a corresponding gearing of the output shaft. The number of the output shafts is limited by the need and the technical possibilities of the inner wheel manufacturing with the required number of gear rims with outer and inner gearing. Output shafts have different gear ratios in dependence on the teeth number (and modules) of individual gears. By a suitable selection of the teeth number of geared couples and taking into consideration the given eccentricity, based on this constructional design it is possible to design either a reducer or a multiplying gear, or by a suitable gearing combination, it is possible to design a gear, where one output shaft will have a reduction gear, while the other one will be provided with a speed-increasing gear. Thus, the gear mechanism can also represent an integrated solution of both a reducer and a multiplying gear.

Possibilities of Toothed Gear Utilization

This gear mechanism can be set:

a) with an involute gearing - with conventionally used modules ($m \geq 1$ mm),
 - with small modules ($m < 1$ mm),

b) with gearing of an arbitrary shape of the tooth flank (e.g. cycloidal).

c) as a friction gearing (i.e. without a gearing).

In fig. 1a,b there are two schemes of a possible two-stage gear with an inner or outer gearing. The same principle can be applied to the solution of a multi-output gear, when the wheel 2 is set by another inner and outer gearing. The meshing combination is with gear rims of the corresponding output shafts.

2.1. KINEMATIC RATIOS

Serving as an illustration, the determination of the gear ratio between the input shaft I and the output shaft II with the involute gearing (according to fig.1) is also described. Fig. 2 shows the separate kinematic scheme of both the low and the high gear based on fig. 1a,b, where the output shaft is set by the wheel with the inner gearing.

As the gear rims z_2 and z_3 are placed on one body of a geared wheel, it hold that : $\omega_2 = \omega_3$.

The eccentricity e equals the revolution radius of the axis of the wheel 2 (of the B point) round the „O“ point of the central axis of the gear. Thus, it is valid that :

$$e = r_e = |r_4 - r_3| = r_1 - r_2 \tag{1}$$

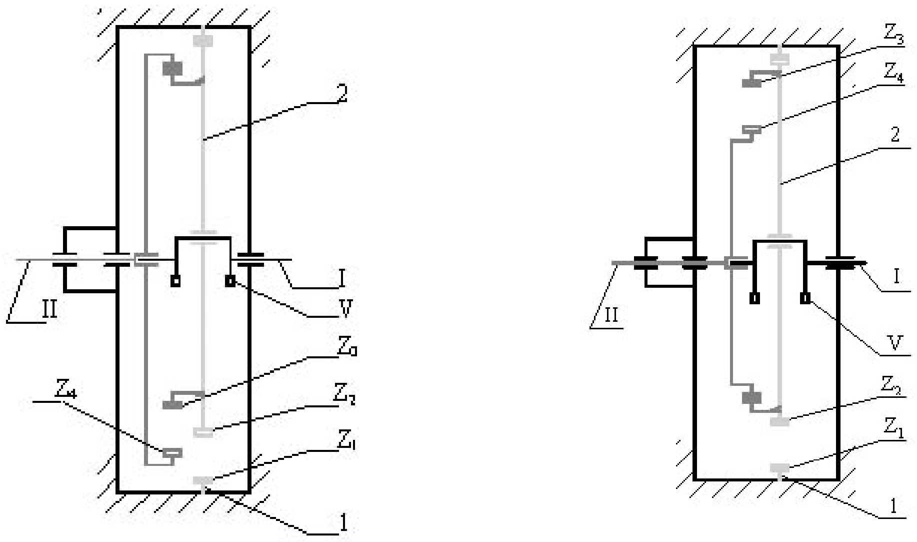
where r_1, r_2, r_3, r_4 are pitch diameters of the gearings z_1, z_2, z_3 and z_4.

Based on fig. 2, for the velocity of the B point it follows that: $\bar{v}_{B40} = \bar{v}_{B4E} + \bar{v}_{EO}$, (2)

where it holds:

$$\bar{v}_{B40} = \bar{v}_{B30}\text{, ďalej } v_{B40} = r_4\omega_4\text{, } v_{B4E} = r_3\omega_3\text{, } v_{EO} = r_e\omega_0\text{,} \tag{3}$$

where $\omega_1, \omega_2, \omega_3, \omega_4$ are angle velocities of the gear rims z_1, z_2, z_3 and z_4.



a) b)

Fig. 1 Two-stage gear
a) with an inner gearing of the output shaft,
b) with an outer gearing of the output shaft

Drawing symbols:

I – input shaft,
II– output shaft
z_1 – inner box gearing *1,*
z_2 – basic gearing of the wheel 2
z_3– gear rim of the wheel *2*,
z_4– output shaft gearing,

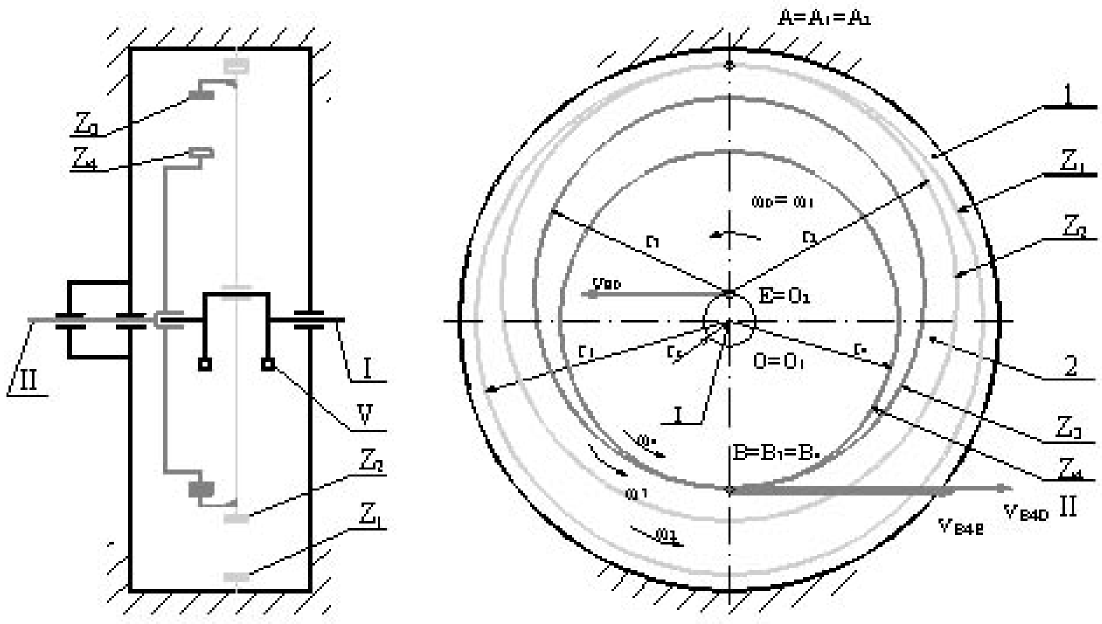


Fig. 2 Separate kinematic scheme of the low gear and the high gear
Output shaft **set** by the wheel with an outer gearing

After being substituted, considering the selected direction of a revolution, according to fig. 2 we obtain:

$$r_4\omega_4 = r_3\omega_3 + r_e\omega_O\,. \tag{4}$$

Based on the above mentioned, the relation between the angle velocity of the input shaft ω_1 and the angle velocity ω_2 of the wheel *2* can be derived. Fig. 2 shows that $\omega_0 = \omega_I$, where ω_O is the velocity of the *E* point of the shaft eccentric around the *O* point on the central axis of the input shaft I. Then for ω_2 regarding ω_O the following relation is valid:

$$\omega_2 = \omega_0 \frac{r_1 - r_2}{r_2} \tag{5}$$

After the above relations and equations (4) a (5) are substituted for a complete gear ratio u_{cII} between the shafts I a II we obtain:

$$u_{cII} = \frac{\omega_0}{\omega_4} = \frac{\omega_I}{\omega_{II}} = \frac{r_4 r_2}{r_2\left|r_4 - r_3\right| - r_3\left(r_1 - r_2\right)} \tag{6}$$

or

$$u_{cII} = \frac{r_4 r_2}{r_e\left(r_2 - r_3\right)}\,. \tag{7}$$

With the involute gearing , the radii r_1, r_2, r_3 and r_4 of the pitch circles can be expressed by means of a module and a number of teeth.

If we indicate:

m_1 – module of the meshing wheels z_1 and z_2,

m_2 – module of the meshing wheels z_3 and z_4,

then in this case for r_e it must be valid: $r_e = (r_1 - r_2) = (r_4 - r_3)$.

$$r_e = \frac{(m_1 z_1 - m_1 z_2)}{2} = \frac{|m_2 z_4 - m_2 z_3|}{2}. \tag{8}$$

After the equation (6) is substituted and modified, the gear ratio in dependence on the teeth number is as follows:

$$u_{cII} = \frac{z_4 z_2}{z_2 |z_4 - z_3| - z_3 (z_1 - z_2)}. \tag{9}$$

Identical relation is also obtained if $m_1 = m_2$.

The gear ratio determination if the output shift (II) is set by the wheel with the inner gearing (z_6).

A general relation used to express each gear ratio u_{cv} between an input and any output shaft when considering a required number of teeth in the gearing of a multi-output reducer can be as follows

$$u_{ci} = \frac{z_{2i} z_2}{z_2 (z_{2i} - z_{2i-1}) - z_{2i-1} (z_1 - z_2)}, \tag{10}$$

where

i – the number of a gear shafts (input shaft + output shafts),

z_1 – inner gearing of a central box wheel *1*,

z_2 – gearing of the inner wheel *2* meshing with z_1,

z_{2i-1} –gearing of the inner wheel *2* meshing with a corresponding output shaft (II, III,),

z_{2i} – gearing of the output shaft meshing inside the gearbox with the gearing z_i.

As mentioned at the beginning of the chapter, the outputs number, or output shafts n_{vj} depends on the number of the meshing gear rims n_{zi} set on the inner wheel *2,* and is expressed by the relation: $n_{vj} = n_{zi} - 1$. (11)

At the same time, it holds: $n_{vj} = i - 1 \Rightarrow n_{zi} = \text{i}$.

With the increasing wheel eccentricity, the entire gear ratio u_c can have the value smaller that *0*. Based on this it follows that, respecting its principle, this gear mechanism can be designed as a multiplying gear**,** which illustrates another important advantage of this solution.

A maximum gear ratio can be achieved if the difference of the teeth number between the gears of the 1st and the 2nd stage is minimum, i.e.: $z_1 - z_2 = z_{2i} - z_{2i-1} = 1$ tooth.

3. CONCLUSION

The aim of the paper was to present the design of the fundamentally new solution of the two-stage multi-output gears. The design is the result of the analysis process of the problems related to the so-called high-accuracy gears as well as an effort to improve some of the negative aspects regarding their loading and age.

The designed gears are capable of reaching high gear ratios ($u_{cv} \approx 1\,000$) in a considerably wide range. With a well-thought-out designing, such a multi-output two-stage gear can operate as a multiplying gear as well as a reducer. It can be assumed that (considering the specialities of involute gearings) even higher gear ratios could be obtained with a cycloid gearing , or with the gearing operating on the basis of a friction gear.

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