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UWZGLĘDNIENIE DEFORMACJI KOŁA PODATNEGO PODCZAS PROJEKTOWANIA

Streszczenie: Przekładnia zębata harmoniczna (falowa) jest w zasadzie przekładnią różnicową, w której zazębianie uzyskuje się poprzez elastyczne odkształcenie jednego z kół. Przekazywanie napędu odbywa się poprzez odkształcenie podatnego wieńca, który przenosi drgania występujące podczas kontaktu oraz zazębiania się koła elastycznego ze sztywnym.

Słowa kluczowe: przekładnia falowa, koło podatne, odkształcenie, zazębienie.

DEFORMATION OF SPRING WHEEL AND ITS IMPACT ON GEAR DESIGN

Summary: A harmonic toothed gear is basically a differential gear with frontal gearing where the meshing is achieved by a flexible deformation of one of the wheels. The deformation of the shape of the flexible wheel is the result of collision and interference, as well as the contact rate, occurring during the meshing of the flexible wheel with the rigid one.

Keywords: harmonic gear systems, flexible wheel, deformation, insertion the teeth.

1. Harmonic gearbox

Harmonic gearboxes are widely used due to the specific and unique features of their mechanism. This mechanism is based on the rolling of the teeth caused by the elliptical deformation of the flexible gear. It has excellent properties especially at steady state at constant speed and under suitable ambient conditions.

The first speciality rests in the fact that in gear and thus and the transmission at the same time a greater number of teeth involved. The greater the of load be will

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transmit flexible member team will grow and its deformation and therefore a greater number of teeth will huddle in toothed.

The second peculiarity the harmonic gearing rests in that, due to changes in shape of the elastic wheel from the load, or due to the choices an shape of the wave generator there is a change a very the small the relative movement between the teeth, the contained the with in toothed engagement.

The third particularity is also conditional on the design of the flexible wheel rests reduce of angles of pressure of kinematic pair of wave generator - of the flexible wheel, as reflected by the reduce friction of sides this pair in compared catch cam - satellite in the planet gear. The principle of harmonic gear (Fig. 1).

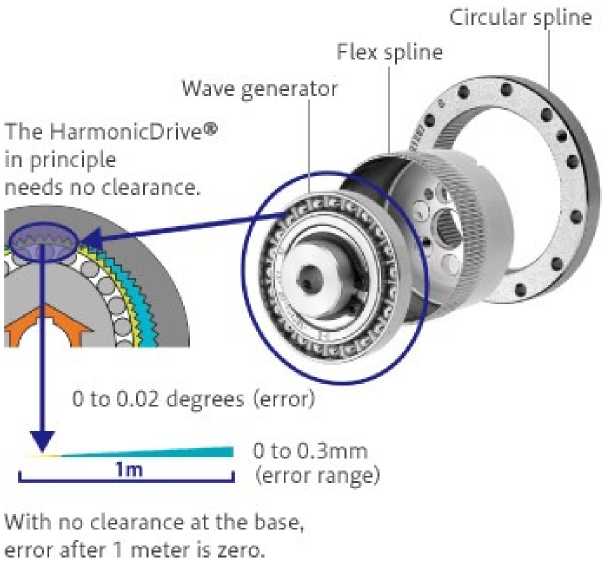


Figure 1. Harmonic drive

2. Construction of flexible wheels

Flexible wheel during operation straining very negative The following are the main stresses:

- Deformity Stress induced by the generator
- Stress induced/generated by the transmitted load
- Local stresses from the tooth flexion within the tooth gaps.

As a result of the adverse stress the flexible wheel is the limiting part of the load-bearing capacity of the harmonic gears.
The flexible wheel is usually made in the form of a thin-walled tube or a thin-walled container with a bottom. Some is in practice often use the which are shapes shown in Fig.2.

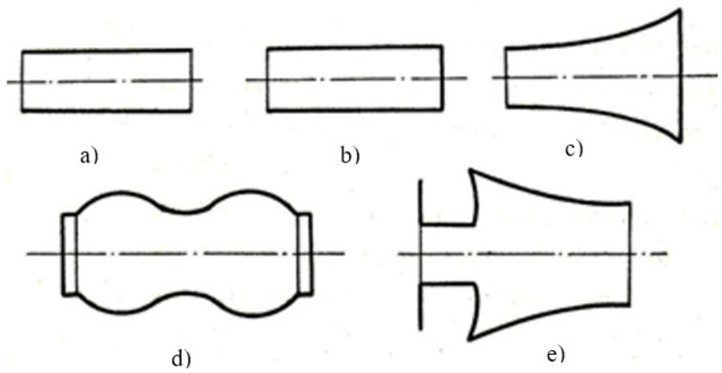


Figure 2. Various cylindrical shapes of flexible wheels

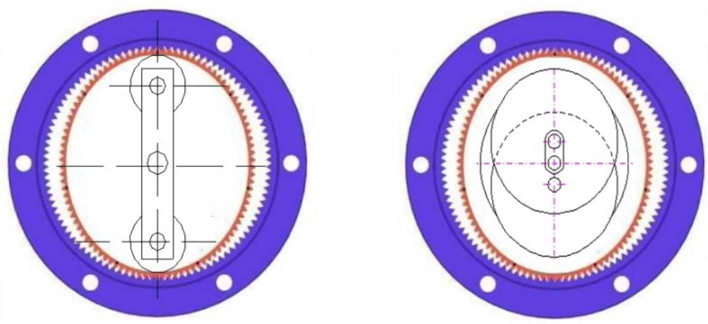
The flexible wheel in the form of a thin-walled container with a bottom is used most often (Fig. 2 b)).

Experimental results [3] had shown that the change in profile form and the tooth position is not significant in possible figures of the flexible wheel deformation and their real ratios between the average of the middle area (middle area is a geometrical area of points dividing the thickness of the flexible wheel wall into two) and of the thickness of the flexible wheel wall.

The flexible deformation of the harmonic wheel may be twofold:

a) The free deformation at which the harmonic wheel is deformed by means of rollers of a smaller cross-section (Fig. 3.a). Harmonic wheel is deformed to shape of ellipse.

b) Forced deformation, where the shape of the wheel is given by the shape of the generator harmonic deformation (Fig.3. b).



a) b)

Figure 3. Deformation of harmonic gear;
a) the loose deformation of harmonic wheel; b) forced deformation of harmonic wheel

2. Deformation of flexible wheel

Flexible wheel during (Fig. 4) operation straining very negative. It causes stresses from generator deformation, stresses from transmitted loads and local stresses from tooth bending in tooth gaps. The flexible wheel is under very disadvantageous stress during the operation. There is stress from the generator deformation, stress from transmitted load and local stress from the tooth flexion within the tooth gaps. The involute curves, which are advantageous in terms of production technology, ensure a suitable engagement of the teeth have the greatest application for the tooth flank profile. After determining the shape of the deformed tooth it is necessary to design an appropriate shape of the opposite profile so when meshing the flexible wheel with the rigid wheel of the harmonic gear it would not cause interference. Tooth flanks solid wheel must be enveloping curves of the tooth flanks of the flexible wheel. The internal gear is when the outer and the inner teeth mesh together. The harmonic gear is such a case where the outer teeth are provided by the flexible wheel and the inner by the rigid wheel. During meshing the associated teeth profiles are in point contact at all times.

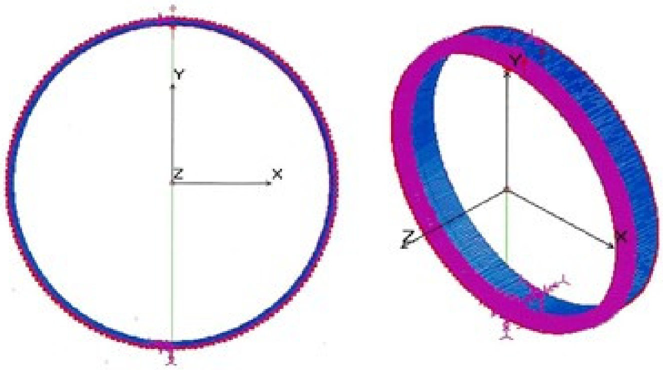
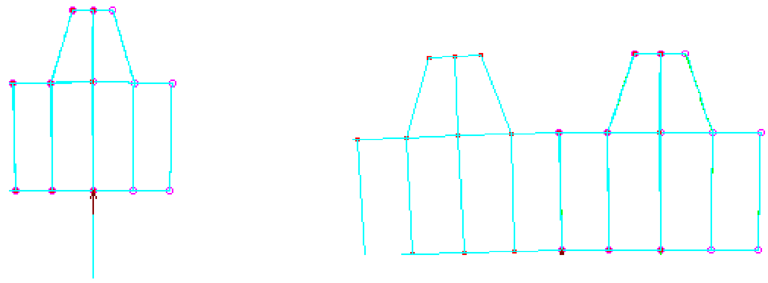


Figure 4. Geometric model of flexible wheel

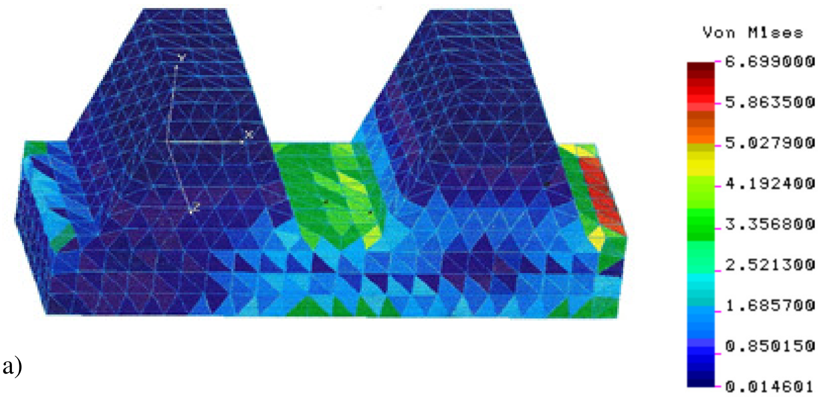
The amount of deformation of the elastic wheel is determined in two cases, namely:
Alternative 1 – if the force is emerging from the wave generator so it is applied in the centre of the tooth (Fig. 5 a).
Alternative 2 - if the force is emerging from the wave generator so it will operate in the middle of tooth gap (Fig. 5 b) .



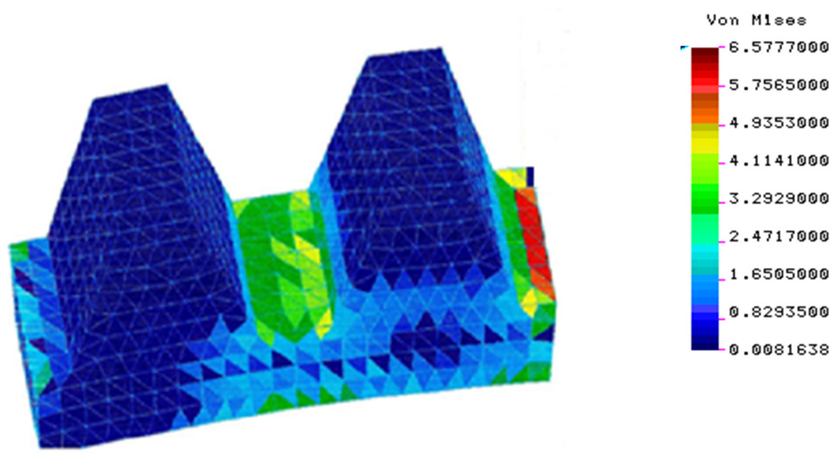
a) b)

Figure 5. Deposition the applied force; a) in the middle of the tooth, b) the force is applied in gap.

After performing a static calculation, we found the size of emerging tensions in the investigated models in alternative 1 (Fig. 6 a)) and the alternative 2 (Fig. 6 b)).



a)



b)

Figure 6. The size of emerging tensions a) in alternative 1, b) in alternative 2

When comparing the value of emerging tensions in the event that force is applied in the middle of the tooth or if force is applied in the tooth clear space find that if force is applied in the middle tooth voltage values are higher than in the case where the force is applied in the middle tooth gap.

In Table 1,2, the magnitude of displacements in the respective nodes in the X, Y direction are recorded

Table 1. Displacement sizes if force is applied in the center of the tooth

Nody	UX_1	UY_1	UZ_1
88	$4{,}142.10^{-3}$	$8{,}202.10^{-2}$	$1{,}262.10^{-6}$
62	$-1{,}157.10^{-6}$	$8{,}202.10^{-2}$	0
63	0	$8{,}153.10^{-2}$	0
64	$1{,}673.10^{-3}$	$8{,}102.10^{-2}$	0
253	$1{,}707.10^{-3}$	$7{,}916.10^{-2}$	0
209	$5{,}415.10^{-5}$	$7{,}860.10^{-2}$	0
2893	$9{,}049.10^{-5}$	$7{,}761.10^{-2}$	$-3{,}592.10^{-7}$
2849	$1{,}781.10^{-3}$	$7{,}717.10^{-2}$	$1{,}059.10^{-7}$
254	$1{,}884.10^{-3}$	$7{,}532.10^{-2}$	0
210	$2{,}566.10^{-4}$	$7{,}469.10^{-2}$	0
2894	$3{,}264.10^{-4}$	$7{,}371.10^{-2}$	$-3{,}751.10^{-7}$
2850	$2{,}03.10^{-3}$	$7{,}333.10^{-2}$	$2{,}292.10^{-7}$

Table 2. Displacement sizes if the force acts in the center of the tooth gap

Nody	UX_2	UY_2	UZ_2
88	$4.026.10^{-3}$	$8{,}014.10^{-2}$	$1{,}027.10^{-6}$
62	$-1{,}120.10^{-6}$	$8{,}014.10^{-2}$	0
63	0	$7{,}966.10^{-2}$	0
64	$1{,}626.10^{-3}$	$7{,}916.10^{-2}$	0
253	$1{,}659.10^{-3}$	$7{,}736.10^{-2}$	0
209	$5{,}183.10^{-5}$	$7{,}681.10^{-2}$	0
2893	$8{,}693.10^{-5}$	$7{,}585.10^{-2}$	$-3{,}382.10^{-7}$
2849	$1{,}730.10^{-3}$	$7{,}542.10^{-2}$	$3{,}212.10^{-8}$
254	$1{,}830.10^{-3}$	$7{,}762.10^{-2}$	0
210	$2{,}468.10^{-4}$	$7{,}301.10^{-2}$	0
2894	$8{,}693.10^{-5}$	$7{,}585.10^{-2}$	$-3{,}326.10^{-7}$
2850	$1{,}972.10^{-3}$	$7{,}168.10^{-2}$	$1{,}692.10^{-7}$

In the Table 3, the values by which the displacements in the respective directions and respective nodes are greater when the force is applied in the center of the tooth are recorded.

$$UX = UX_1 - UX_2$$
$$UY = UY_1 - UY_2$$
$$UZ = UZ_1 - UZ_2$$

Table 3. The difference in displacement values in the individual nodes

Nody	UX	UY	UZ
88	$1{,}16.10^{-4}$	$1{,}88.10^{-3}$	$2{,}35.10^{-7}$
62	$-3{,}7.10^{-8}$	$1{,}88.10^{-3}$	0
63	0	$1{,}87.10^{-3}$	0
64	$4{,}7.10^{-3}$	$1{,}83.10^{-3}$	0
253	$4{,}8.10^{-5}$	$1{,}8.10^{-3}$	0
209	$2{,}32.10^{-6}$	$1{,}79.10^{-3}$	0
2893	$3{,}56.10^{-6}$	$1{,}31.10^{-3}$	$-2{,}1.10^{-8}$
2849	$5{,}1.10^{-5}$	$1{,}75.10^{-2}$	$7{,}378.10^{-8}$
254	$5{,}4.10^{-5}$	$-2{,}3.10^{-3}$	0
210	$9{,}8.10^{-6}$	$1{,}68.10^{-3}$	0
2894	$2{,}3787.10^{-4}$	$-2{,}14.10^{-3}$	$-4{,}25.10^{-8}$
2850	$5{,}8.10^{-5}$	$1{,}65.10^{-3}$	6.10^{-8}

Using the values from Table 3, I find out on average what value is the displacement greater if the force acts in the center of the tooth in the direction X, Y, Z.

Arithmetic mean of the displacement in the X -Δ X

i - the number of nodes in which they were subtract displacement

$$\Delta X = \frac{\left|\sum UX\right|}{i} = \frac{5{,}280587.10^{-3}}{12} = 4{,}400489.10^{-4}\ mm$$

Arithmetic mean of the displacement in the Y - ΔY

$$\Delta Y = \frac{\left|\sum UY\right|}{i} = \frac{3{,}763.10^{-2}}{12} = 3{,}135833.10^{-3}\ mm$$

Arithmetic mean of the displacement in the Z - ΔZ

$$\Delta Z = \frac{\left|\sum UZ\right|}{i} = \frac{4{,}3228.10^{-7}}{12} = 3{,}60233.10^{-8}\ mm$$

4. Conclusion

After determining the shape of the deformed tooth it is necessary to design an appropriate shape of the opposite profile so when meshing the flexible wheel with the rigid wheel of the harmonic gear it would not cause interference. Tooth flanks solid wheel must be enveloping curves of the tooth flanks of the flexible wheel.

The internal gear is when the outer and the inner teeth mesh together. The harmonic gear is such a case where the outer teeth are provided by the flexible wheel and the inner by the rigid wheel. During meshing the associated teeth profiles are in point contact at all times. The most used direct construction is to design a profile like envelope circles but this is the goal of further work.

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