Planetary reducer with herringbone gear

Razumov M.S.

PhD in Technical Sciences, associate professor
Department of Machine-building technologies and equipment
Southwest State University, Kursk
E-mail: mika_1984_@mail.ru

Gridin D.S.

Magister of Department of Machine-building technologies and equipment
Southwest State University, Kursk
E-mail: swsu.st@inbox.ru

Abstract
In the paper, new design of planetary reducer with herringbone gear is suggested. This design will allow reducing dimensions of planetary mechanisms. Characteristics of the suggested reducer design are described, and it is compared with standard reducer.
Key words: PLANETARY REDUCER, HERRINGBONE GEAR, GEAR RATIO, ROTATION FREQUENCY

The planetary reducer becomes widespread in the modern world, ranging from transport mechanical engineering to armament industry. The main directions of the scientific researches devoted to reducers of this type are reduction of boundary dimensions of gear, increase of transmission power, possibility of operation at high rotational velocity [7, 9, 10].

The design of planetary reducer with herringbone gear has been developed at the department of Machine-building technologies and equipment of Southwest state university [1, 3].

The main difference of suggested planetary reducer from classical designs is usage of wheels with herringbone gear. When using planetary reducer during operation with high rotational velocity, it is necessary to consider that during rotation any mechanism produces vibrations which negatively affect operation of the mechanism in general. It is known that in course of operation helical wheel causes considerable axial loads, which are transmitted to the parts of gear.

Therefore, herringbone gear is applied for prevention of axial loads in the new scheme of planetary reducer. Due to opposite direction of tilt angle of teeth, the possibility of emergence of axial loads is eliminated, and also smoothness of mechanism operation increases because of bigger number of teeth which are in gear. For providing simplicity of rigging and service herringbone wheel is assembled by connection of two helical wheels (Figure 4) with the increased tilt angle of teeth; that will also allow transmitting rotation moments.

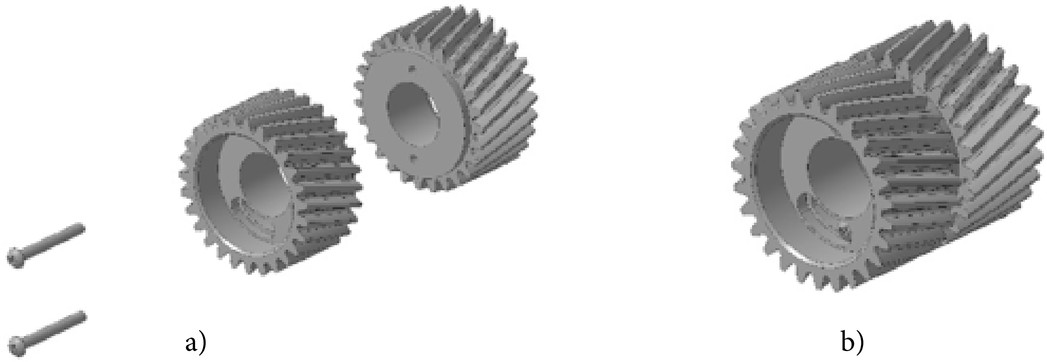

Figure 1. Design of assembled herringbone gear:
a) view with spaced parts, b) type of assembled wheel

As is seen from Figure 1, herringbone wheel possesses mechanical adjustment, which is achieved by cutting out of groove of necessary shape in the first wheel and drillings of threaded openings on the second one. For fixing of wheels relative to each other, any standard fastening element can act as fixing device; in our case, it is screw. This adjustment provides, first of all, simplicity of service, and also allows prevention of wheels positioning inaccuracy which grows because of instrument and equipment error when cutting out of wheel teeth at the place of production.

Before starting, the comparative analysis of all possible arrangements of planetary reducer elements was carried out, and also all the advantages and disadvantages of possible variants were revealed. After selection of the final scheme, the main objective was problem of determination of the main technical characteristics of such gear.

The 3D model of such reducer (Figure 2) has been created. As a result, dependences of rotation speed of mechanism function elements and gear ratio U_{red}=10.5 were determined by means of animation. These data are confirmed when determining gear ratio of the drive according to calculated value.

The gear ratio of planetary reducer is formation of gear ratios of each gear [6].

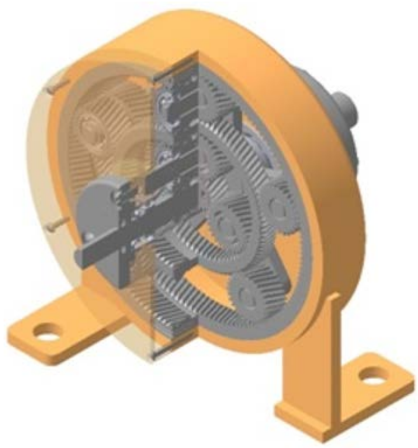

Figure 2. Model of planetary reducer with herringbone gear with the hole (the front cover is transparent for visual representation)

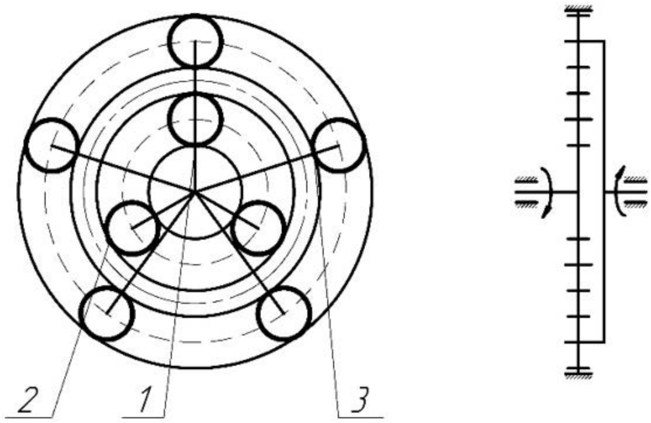

Figure 3. The kinematic scheme of planetary reducer with herringbone gear

Let us consider the selected scheme (Figure 3) where the gears participating in transmission are specified. Transmission from solar wheel to satellites of the first order $U_{12}^{(h)} = {z_2}/{z_1}$ is presented by position 1. The position 2 presents transmission from satellites of the first order to the combined wheel $U_{23}^{(h)} = {z_3}/{z_2}$. Transmission from the combined wheel to satellites of the second order $U_{4h}^{(6)} = 1 + {z_6}/{z_4}$ is signed by position 3.

It follows that

$$U_{red} = U_{12}^{(h)} \cdot U_{23}^{(h)} \cdot U_{4h}^{(6)}$$

$$U_{red} = {}^{z_2}/_{z_1} \cdot {}^{z_3}/_{z_2} \cdot (1 + {}^{z_6}/_{z_4})$$

$$U_{red} = {}^{z_3}/_{z_1} \cdot (1 + {}^{z_6}/_{z_4})$$

Having determined the number of teeth of each gear wheel, let us calculate the gear ratio of the total reducer:

$$U_{red} = {}^{27}/_{18} \cdot {}^{72}/_{27} \cdot (1 + {}^{142}/_{88}) = 10{,}5$$

For comparison with standard gear, let us select the number of teeth of classical gear wheels in such a way that not to exceed the maximum number of wheel teeth of the considered scheme. If it is not observed, conditions of coincidence of overall dimensions will be violated.

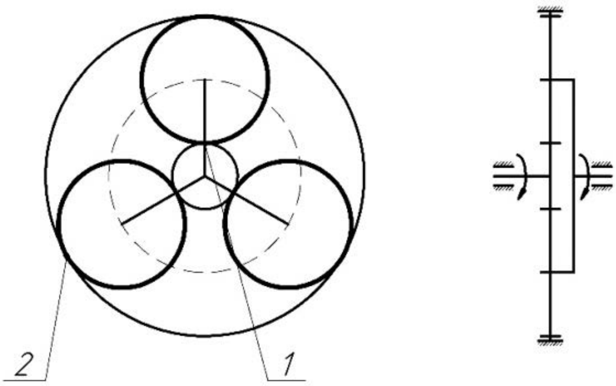

Figure 4. Kinematic scheme of standard planetary reducer

As the maximum number of teeth in the considered scheme is z_6 =142 , let us select the number of teeth which are possible for this type of gears z_1=18 by formula [8]. From the scheme of standard reducer (Figure 4), it is seen that the number of gears is less. The gear ratio of such reducer is determined by a formula:

$$U_{red} = 1 - U_{13}^{(h)} = 1 + {}^{z_3}/_{z_1}$$

$$U_{red} = 1 + {}^{142}/_{18} = 8{,}9$$

Proceeding from calculations, it is seen that the gear ratio of reducer according to the new scheme is higher, than of standard one; at the same time, overall dimensions are identical that confirms relevance of decision.

Let us compare both reducers by parameters of rotating moment. It is necessary to carry out previous calculations of rotation frequency *n*, angular speed "*ω*" and capacities of each step *P* [4]. Before determining the rotation frequency, it is necessary to select electric motor from Table 3 [5] by diameter of input shaft; in our case, it is AIR 71B4/1350 and we will use it for two options of reducers.

Let us find rotation frequency of each step. As rotation frequency of the electric motor is equal to 1350 rpm, let us designate it as speed of input shaft of reducer of n_{in}=1350. Further, calculations are conducted by the following formula:

$$n_i = n_{i-1}/u_i$$

where n_{i-1} - rotation frequency of previous *i-th* step; u_i– gear ratio of *i-th* gear.

It follows:

$$n_2 = {}^{n_{in}}\Big/_{u_{12}^{(h)}} = {}^{1350}/_{1,5} = 900 \text{ rpm}$$

$$n_3 = {}^{n_2}\Big/_{u_{23}^{(h)}} = {}^{900}/_{2.667} = 337.5 \text{ rpm}$$

$$n_{out} = {}^{n_3}\Big/_{u_{4h}^{(6)}} = {}^{337.5}/_{2.614} = 129.13 \text{ rpm}$$

Let us find the angular speed of each step by formula:

$$\omega_i = {}^{\pi \cdot n_i}/_{30}, ({}^{rad}/_{s})$$

where n_i– rotation frequency of *i-th* step; w_i– angular speed of *i-th* step.

$$\omega_{in} = {}^{\pi \cdot n_1}/_{30} = {}^{3,14 \cdot 1350}/_{30} = 141{,}372$$

$$\omega_2 = {}^{\pi \cdot n_2}/_{30} = {}^{3,14 \cdot 900}/_{30} = 94{,}248$$

$$\omega_3 = {}^{\pi \cdot n_3}/_{30} = {}^{3,14 \cdot 337,5}/_{30} = 35{,}343$$

$$\omega_{out} = {}^{\pi \cdot n_{out}}/_{30} = {}^{3,14 \cdot 129,13}/_{30} = 13{,}523$$

Let us determine capacity of each step. Electric motor power is known $P_{em} = 0{,}75\ kW$. From Table 1 [5], let us find efficiency of planetary transmission $\eta_{pl.tr.} = 0{,}97$. For determination of power of each step, it is necessary to multiply obtained step power by efficiency of planetary transmission [2] consistently as transmitting rotation. Let us carry out calculation by formula:

$$P_i = P_n \cdot \eta_{pl.pr.}, (kW)$$

where P_n – power of the considered gear; P_i – step power.

$$P_{in} = P_{em} \cdot \eta_{pl.pr.} = 0{,}75 \cdot 0{,}97 = 0{,}728$$

$$P_2 = P_{in} \cdot \eta_{pl.pr.} = 0{,}728 \cdot 0{,}97 = 0{,}706$$

$$P_3 = P_2 \cdot \eta_{pl.pr.} = 0{,}706 \cdot 0{,}97 = 0{,}685$$

$$P_{out} = P_3 \cdot \eta_{pl.pr.} = 0{,}685 \cdot 0{,}97 = 0{,}664$$

Knowing all the necessary data, let us calculate the rotating moment of each step by the following formula:

$$T_i = {(P_i \cdot 1000)}/{\omega_i}, (N \cdot m)$$

where P_i, ω_i – power and angular speed of *i-th* step respectively.

$$T_{in} = {(P_{in} \cdot 1000)}/{\omega_{in}} = {(0{,}728 \cdot 1000)}/{141{,}372} = 5{,}146$$

$$T_2 = {(P_2 \cdot 1000)}/{\omega_2} = {(0{,}706 \cdot 1000)}/{94{,}248} = 7{,}487$$

$$T_3 = {(P_3 \cdot 1000)}/{\omega_3} = {(0{,}685 \cdot 1000)}/{35{,}343} = 19{,}368$$

$$T_{out} = {(P_{out} \cdot 1000)}/{\omega_{out}} = {(0{,}664 \cdot 1000)}/{13{,}523} = 49{,}101$$

Let us carry out calculation of standard planetary reducer by determination of shaft rotation frequency, angular speed, power and rotating moment:

$$n_{in} = n_{em} = 1350\, rpm$$

$$n_{out} = {n_{in}}/{u_{red}} = {1350}/{8{,}9} = 151{,}7\, rpm$$

$$\omega_{in} = {\pi \cdot n_{in}}/{30} = {3{,}14 \cdot 1350}/{30} = 141{,}372\, ({rad}/{s})$$

$$\omega_{out} = {\pi \cdot n_{out}}/{30} = {3{,}14 \cdot 151{,}7}/{30} = 15{,}9\, ({rad}/{s})$$

$$P_{in} = P_{em} \cdot \eta_{pl.pr} = 0{,}75 \cdot 0{,}97 = 0{,}728\, (kW)$$

$$P_{out} = P_{in} \cdot \eta_{pl.pr.} = 0{,}75 \cdot 0{,}97 = 0{,}706\, (kW)$$

$$T_{in} = {(P_{in} \cdot 1000)}/{\omega_{in}} = {(0{,}728 \cdot 1000)}/{141{,}372} = 5{,}146\, (N \cdot m)$$

$$T_{out} = {(P_{in} \cdot 1000)}/{\omega_{out}} = {(0{,}706 \cdot 1000)}/{15{,}87} = 44{,}43 (N \cdot m)$$

The comparative analysis has shown that the planetary reducer is capable to compete with existing types of reducers, as at identical overall dimensions, parameters of the suggested design are higher.

Work is performed with financial support by contract No 6394GU/2015 on the subject “Development of Planetary Reducer with Herringbone Gear “.

References

1. Razumov M.S., Gladyshkin A.O., Kassikhin V.N., Pykhtin A.I., Skripkina E.V. (2014) Stress calculation of moment transmitting roll with profile on the base of Reuleaux triangle. *Metallurgical and Mining Industry*. No 3, p.p. 50-54
2. Ivakhnenko A.G., Kuts V.V. *Strukturno-parametricheskiy sintez tekhnologicheskikh system*. [Structural and parametric synthesis of technological systems]. Kursk, Kursk State Technical University, 2010. 153 p.
3. Razumov M.S., Gridin D.S., Dubovoy A.Yu. (2015) Increase of constructive capacity of planetary transmissions. *Sovremennye instrumental’nye materialy, informatsionnye tekhnologii i innovatsii*. Kursk, 2015, p.p. 362-366.
4. Uchaev P.N. *Mekhanicheskiy privod s reduktorami, motor-reduktorami i korobkami skorostey*. [The mechanical drive with reducers, the motor reducers and transmission gears]. Staryy Oskol, TNT, 2010
5. Uchaev P.N. *Zubchatye peredachi s zadachami i primerami raschetov*. [Tooth gearings with tasks and examples of calculations]. Staryy Oskol, TNT, 2007
6. Frolov K.V. *Teoriya mekhanizmov i mekhanika mashin*. [Theory of mechanisms and mechanic of machines]. Moscow, Vysshaya shkola, 2003. 496 p.
7. Amirova Yu.D. *Tekhnologichnost’ konstruktsii izdeliya*. [Technological effectiveness of product design]. Moscow, Mashinostroenie, 2002. 768 p.
8. Dunaev P.F. *Konstruirovanie uzlov i detaley mashin*. [Desogning of units and details of machines]. Moscow, Vysshaya shkola, 2006. 447 p.
9. Anur’yev V.I. *Spravochnik konstruktora mashi-*

nostroitelya. [Reference manual of mechanician]. Moscow. Mashinostroenie, 2001. Vol. 1, 920 p.
10. Markov N.N. *Normirovanie tochnosti v mashinostroenii*. [Rating of accuracy in mechanical engineering]. Moscow, Vysshaya shkola, 2001. 335 p.

Analysis of the design versions of auxiliary verifier for weight-free verification of large-load platform railroad scales

Boryak K. F.

D.Sc. in engineering
Odessa State Academy of Technical Regulation and Quality,
Odessa, Ukraine

Abstract
Two design versions of auxiliary verifier VNKU-60 for weight-free verification of large-load platform railroad scales are compared. Both versions are new in their designs and were elaborated with the participation of the author in the course of performing the research work. Individual verification schemes of simultaneously loading the standard weight measuring sensors embedded in the VNKU-60 design and the duty sensors of load carrier with ballast weight via a weight-transmitting device using hydraulic jacks for both designs of VNKU-60 are proposed, respectively. On the basis of analysis of the advantages and disadvantages of each version of VNKU-60 design the recommendations on its further use in present-day conditions were given.
Keywords: NON-AUTOMATIC WEIGHING CALIBRATION DEVICE, AUXILIARY VERIFIER, LOAD CARRIER, WEIGHT-TRANSMITTING DEVICE, RAILROAD SCALES, WEIGHT-FREE VERIFICATION, CALIBRATION OF LARGE-LOAD SCALES, STANDARD WEIGHT MEASURING INSTRUMENT

Large-load platform scales are the most common means of instrumentation when performing commercial transportation by railroad. Technical and metrological characteristics of the scales shall meet the requirements [1, 2]. Verifying (calibrating) the scales shall be performed in accordance with the requirements [3]. According to these requirements the basic method of verifying of large-load stationary platform railroad scales is directly loading the scale platform with M_1 class standard weights 2 tons in mass. The to-