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Decrease in vibroactivity of gear drives

O S Dyundik, I P Zgonnik

Omsk State Technical University, 11, Mira ave., Omsk, 644050, Russia

E-mail: tmm@omgtu.ru

Abstract. The analysis of technical and technological solutions to reduce the influence of dynamic intermating process to vibroactivity of a gear drive is carried out. The problem of a complete elimination has no solution due to a discrete way of power flow transformation by a gear drive. It is shown that a reduction of mesh stiffness is a controlled parameter in vibroactivity decrease. We have proposed to reduce the stiffness by means of a gear rim radial cut that provides small initial and stepwise variable stiffness of gearing. The technical solutions resulting in a smooth, dependent on a power flow running of stiffness by means of built-in control chain and usage of elastomeric materials are given. As a result, it is noted that the reduction of mesh stiffness at the same time leads to the increase in static deformation δ , and the dynamic load under intermating decreases. Elastomeric parameters' variation t can provide the achievement of calculated mesh stiffness, depending on a variable power flow.

1. Introduction

Gear drives have some advantages such as converting the motion between shafts irrespectively of location in space, compactness, high bearing capacity and efficiency, reliability, manufacturability. These advantages determine a wide spread of these types of transmissions in mechanical engineering objects. However, along with the advantages, gear drives objectively have disadvantages, such as increased vibroactivity caused by the dynamic tooth changeover, under inherent pitch error, explained by primary and power errors, as well as by variable mesh stiffness both in one-pair and in multipair toothing. The problem of vibroactivity decrease of gear drives is urgent and is widely considered in foreign literature, such as in [1, 2].

2. Problem statement

Let us set and solve the problem of analysis for technical and technological solutions to reduce the influence of dynamic intermating process to vibroactivity of a gear drive, as the problem of a complete elimination of this phenomenon has no solution due to the discrete way of power flow transformation by the gear drive.

3. Experiment methodology

It is known that specific dynamic loading P_{dyn} causing vibrations under intermating is defined as follows:

$$P_{dyn} = Av\sqrt{m_{ef}(\delta+\Delta p)\cdot c}, \quad (1)$$

where A is a proportionality factor (its value is not important for a quality process estimate);

v is a peripheral velocity in initial cylinders of transmission wheels (m/s);

m_{ef} is a gear drive mass reduced to a polar point;

δ is a static flexural deformation of a tooth under loading (a power error of a base pitch);

Δp is a primary error of the base pitch;

c is the mesh stiffness.



Analyzing (1) let us note that the mesh stiffness c is the only controlled parameter. To decrease stiffness is necessary to reduce dynamic loading P_{dyn} which generates a forced vibration with intermating frequency

$$f=nz,$$

where n is a revolution number of a gear (or a wheel), z is a number of teeth in the wheel. This recommendation is realized in many technical solutions of gear drives. Thus, gear drives with increased tooth height are used in high-speed airborne gearboxes. There are well-known methods of reducing teeth stiffness by specific technological cuts, by drilling in rims, etc. However, one should keep in mind that the reduction of mesh stiffness at the same time leads to the increase in static deformation δ .

Let the tooth be a cantilever beam (figure 1), static deflection δ under the force P will be the following:

$$\delta = \frac{P \cdot l^3}{3EJ}, \quad (2)$$

where P is the force operating in gearing;

l is a tooth height;

E is a material elasticity modulus;

J is an inertia moment of a root section.

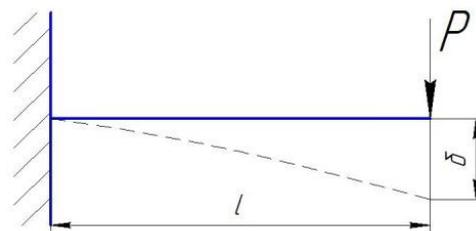


Figure 1. Defining of a tooth deflection as a cantilever beam.

4. Experimental results

As a result we can see that the reduction of mesh stiffness leads simultaneously to the increase in static deformation δ , and the dynamic load under intermating decreases.

Thus following (2) if to increase twice L console length of the tooth with a cut, the static deflection δ will increase eightfold, and stiffness $c = \frac{P}{\delta}$, with the same P will also decrease eightfold. According to (1) this will result in threefold reduction of specific dynamic loading R_{dyn} .

5. Results and discussion

There is a very progressive technical solution [3], which provides the small initial mesh stiffness under a simultaneous deformation restriction of a bearing teeth pair. It is achieved due to teeth hollows in a wheel body are made with rectilinear radial grooves of substantial length l . To limit gearing deformation, the grooves enter hollows via arcform cuts with (0,2÷0,4) mm width.

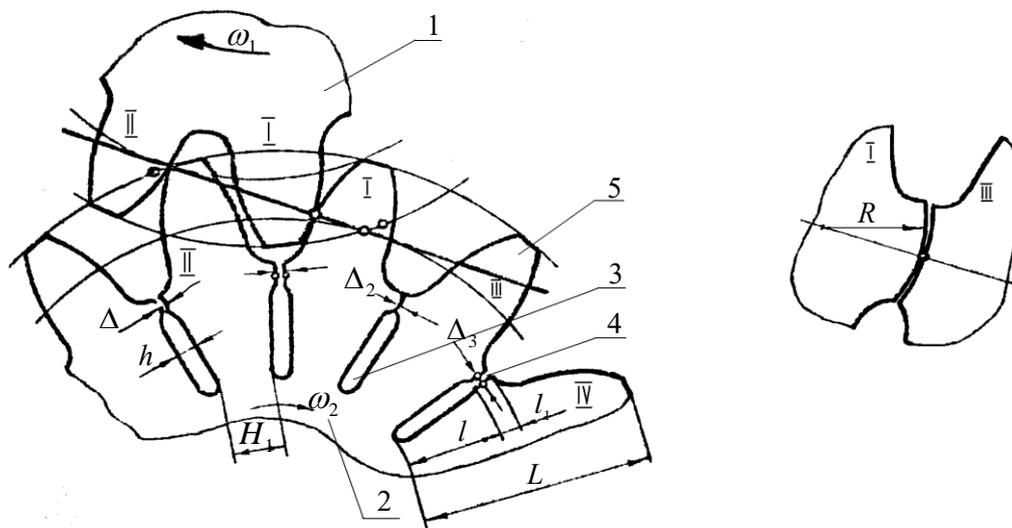


Figure 2. A gear wheel with small initial stepwise variable mesh stiffness [3].

The technical solution [4] (figure. 3) obtained in the development of [3] is more universal. This solution provides an adjustable control of stiffness depending on the level of power flow transferred by the transmission by changing of an active part of radial cut. The gear rim is movably installed on a shaft by means of bearings, thus there is no rim axial displacement towards the shaft, but a relative angular motion is allowed. In wide parts of radial cuts in the proposed gear drive there are special sliders 2 which are connected with a hub 4 by means of flexible rockers 3 made as flat curved springs hingedly fixed on the hub. It should be noted that the hub is rigidly fixed on a shaft. Flexible elastic elements 5 between the shaft and the gear rim are installed to obtain a desired angular shaft displacement to the gear rim. The proposed elastic elements 5 can be designed as an elastomeric ring or a metal spring of a particular form.

The tooth height regulation is provided with the built-in control chain including the elastic element 5 which deformation depends on power flow parameters and this deformation is used for the motion of special sliders 2 in radial cuts. The position of the sliders determines the cut length and, therefore, the mesh stiffness.

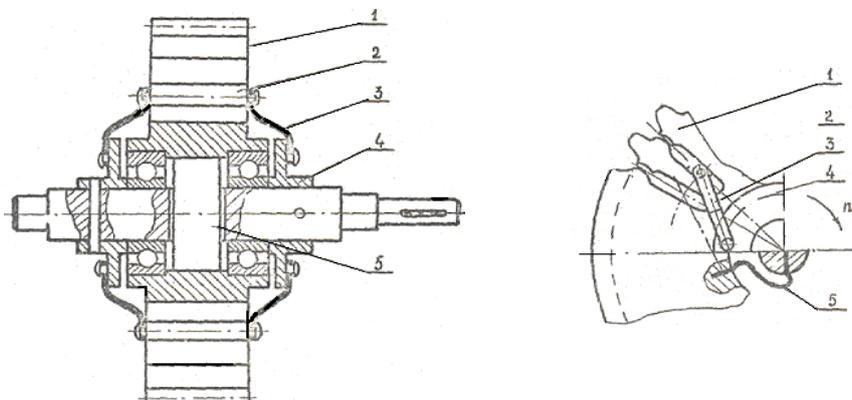


Figure 3. A gear wheel with self-guided mesh stiffness [4].

However, it should be noted that the technical solution [4] is technologically complicated and its implementation is hardly possible for universal mechanical gearboxes. It is much simpler to execute the mesh stiffness control by a gear drive with automatically controlled stiffness on the technical solution [5] where complicated in production sliders with flexible rockers are excluded from the control chain and are replaced with technologically more available elastomeric inserts located in a radial cut in a gear wheel hollow of a modified design according to figure. 4, a, which can also be realized by elastic inserts (figure. 4, b).

6. Conclusion

Since the deformation of such inserts is non-linear and essentially depends on the elastomer shape and its fixing method in a cut, it is possible to provide the achievement of calculated mesh stiffness depending on a variable power flow by the variation of these parameters. This calculation is mandatory, and using of elastomeric inserts is prospective for the decrease in vibroactivity of gear drives. The implementation of technical solution [5] has no restrictions and is applicable in universal gearboxes.

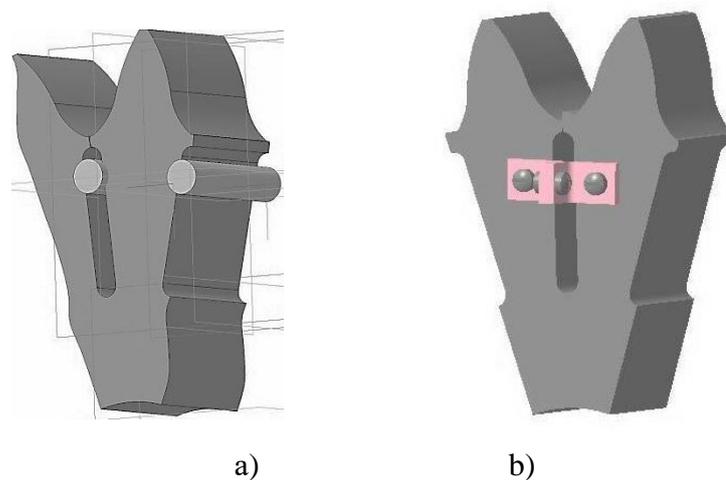


Figure 4. A part of gear wheel with variable flexural mesh stiffness [5].

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