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VIBRATION DAMPING OF BEVEL GEAR

Summary. The article points out to causes effecting dissipation of mechanical energy of toothed wheels of toothed gear. The results of experimental investigations which aim was to determine the influence of load and change of mesh rigidity on damping of vibrations of wheels of toothed bevel gear has been shown. The analysis of effects has revealed that the damping varies together with variation of mesh rigidity, the relative value of change amplitude remains constant. The damping increases together with load increasing.

The vibration damping is one of manifestation of energy dissipation which is associated with a movement of mechanical systems. In wide range of problems of investigation of dynamic phenomena occuring while the toothed gear operates, the investigations associated with damping of vibration movement of toothed wheels belong to a group of less advanced problems. The processess causing vibration damping of toothed gear are very complex and a konwledge in this scope is insufficient. The results of investigations published in [1, 2, 3, 4, 5] determining the damping values differ considerable between each other. Therefore the problems associated with this topic and especially referred to the toothed gear are still open. The simplest form of damping is asumed in numerical calculation [6, 7, 8]. It is to be assumed a linear relationship between the damping force and the velocity of vibration, whereas the damping factor is to be chosen in such a way, so thaf at least within a resonacce states the effects of experimental investigations coincide with the effects of numerical calculation.

There are various causes effecting dissipation of mechanical energy of vibration of toothed wheels. In technical approache to the problem the main causes may be the following:

- internal friction associated with the structural design of material of which the toothed wheels and other parts of toothed gear are performed,
- static friction occuring on the contact surfaces of parts connected motionless,

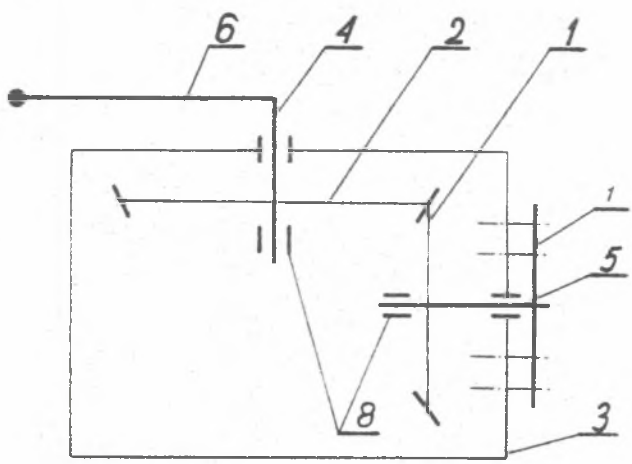
- friction between moving parts occuring at distinct relative movement of parts of toothed gear,
- presence of an oil film between matching teeth.

The aim of this elaboration is, on the basis of experimental tests to define the influence of load and rigidity of mesh on the dissipation of mechanical energy caused by internal friction, static friction and friction between moving parts.

OBJECT OF INVESTIGATION

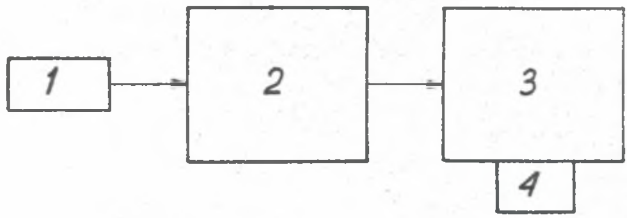
The object of investigation were the spiral toothed bevel gears made with "Oerlikon" method having the following parameters: transmission ratio - $u = z_2 : z_1 = 54:41$; face module - $m = 4,5$ mm; apparent pressure angle at the pitch diameter - $\alpha_{on} = 0,349$ rad; face width of gear - $b = 12$ mm; engle of inclination of spiral line in central section of toothed rim - $\beta_m = 0,384$ rad; angle between rotation axes of wheels - $\sum = \Pi/2$ rad. The toothed wheels are carried out of 40 HM quenched and tempered steel in 7 th grade of tolerance according to GOST.

The investigation stand (Drg. No. 1) is designed so thad the disc (7) seated on the shaft (5) on which the wheel (1) is seated may be fastened to the housing of toothed gear in any location. The wheel (2) seated on the shaft (4) might freely perform (limited) roational motion. The lever (6) of determined arm length served for application the load to the toothed wheel. The geometrical dimensions of the lewer (6) and the disc (7) were so selected, thad their rigidities were about twenty five time



Drg. No.1. Investigation stand diagram
1,2 - toothed wheels, 3 - case of toothed gear, 4,5 - shafts on which the toothed, 6 - lever arm, 7 - setting disc, 8 - rolling cone bearing

greater than that of other parts of toothed gear. Both shafts were supported by rolling bearings (8). The load of toothed gear was realized by increasing the weight applied to the arm of lever. The piezoelectric sensor serving as a vibration meter were fastened to the disc of wheel (2). The signals from the sensor (Drg. No. 2) were led to the accelaration, velocity, and displacement of vibration motion and then to the oscilloscope input. The course of vibration motion might be observed on the oscilloscope monitor or photographed with a camera specially adapted for this purpose.



Drg. No. 2. Block diagram of vibrations measuring and recording aparatus set

1 - piezoelectric sensor, 2 - vibration meter, 3 - cathode oscilloscope

COURSE AND RESULTS OF INVESTIGATION

The investigations were carried out at various loads of wheels. The diagrams were registered on the cine-film at relative position of wheels when:

- one pair of teeth was in mesh,
- two pairs of teeth were in mesh.

The torque was a load measure. According to change of torque the load factors Q amounted to: 0,85; 1,18; 1,62; 2,05; 2,5; 3 and 3,5 MPa. These values were calculated from the formula:

$$Q = \frac{2M}{bd_m^2} \tag{1}$$

where:

M - prescribed given torque,

b - face width of gear,

d_m - diametr of gear measured in the middle of face width of gear.

There were registered vibrations at, least for twelve various associations for one pair in mesh and for ten various associations for two pairs in mesh for each load. Attention was paid that the relative position of teeth was the same at each association.

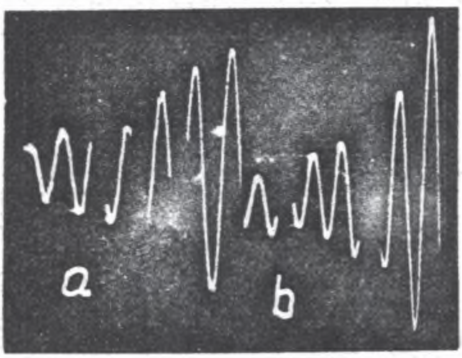
As a damping measure the logarithmic damping decrement of free vibrations which value was calculated from the following formula:

$$\delta = \frac{1}{m} \ln \frac{a_n}{a_{n+m}} \tag{2}$$

where:

a_n - deflection of vibrations amplitude,

a_{n+m} - deflection of vibrations amplitude measured after m half-periods.



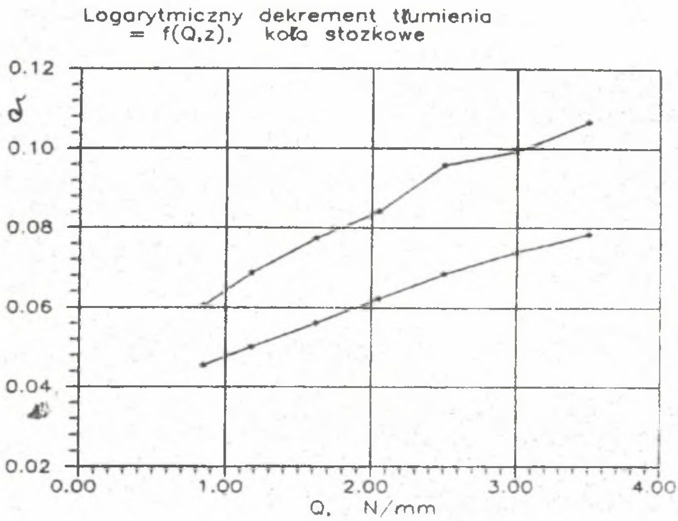
Drg. No.3. Courses of vibration at bipair meshing
a) Q = 2,5 MPa, b) Q = 3,5 MPa

The action of exciting force has been broken after forcing the vibration of determined amplitude and then the diagrams of free vibrations was registered on the cine-film. Two diagrams registered for various loads when two pairs of teeth were in mesh are shown in drg. No. 3. When exciting the vibrations efforts have been made that the exciting amplitudes were the same independign on the load. The obtained results were worked out statistically determining the logarithmic damping decrements by one pair in mesh (δ_1) and two pairs in mesh (δ_2) for each load. These results are graphically shown in drg. No. 4. There is visible clear dependence of logarithmic damping decrement bot on load and rigidity of mesh.

In order to determine the influence of variation of rigidity of mesh on dissipation of mechanical energy of vibrations the difference of logarithmic damping decrement expressed in percentages has been determined for each load using the following formula:

$$\Delta\delta = \frac{\delta_1 - \delta_2}{\delta_1} 100\% \tag{3}$$

The logarithmic damping decrements (δ_1, δ_2) and the differences between them expressed in percentages for various loads (Q) are laid down in the table 1. Comparing the loss of mechanical energy of vibrations of wheels determined for one pair and two pair mesh it is seen (table 1) that the



Drg. No. 4. Vibration decrements values as a function of load factor
1 - unipair meshing, 2 - bipair meshing

Table 1

Lp.	Q [MPa]	δ_1	δ_2	$\Delta\delta = \frac{\delta_2 - \delta_1}{\delta_1}$ 100%
1	0,85	0,0455	0,0605	33
2	1,118	0,0501	0,0687	37
3	1,620	0,0561	0,0774	38
4	2,050	0,0623	0,0841	35
5	2,500	0,0689	0,0958	39
6	3,000	0,0740	0,0992	34
7	3,500	0,0784	0,1067	36

loss for two pair mesh are in average greater by 36%. It should be remembered, that the determined values of logarithmic damping decrements take into account the loss of mechanical energy of vibration of wheels effected by internal friction, static friction and friction between moving parts which take place also in real objects. The loss of energy caused by viscous resistance in the oil film lead between matching surfaces of teeth during operation of the toothed gear has not been taken into consideration.

FINAL CONCLUSIONS

The loss of mechanical energy of vibration of toothed gear caused by internal friction, static friction and friction between moving parts depends on load of toothed gear and on rigidity of mesh.

Independing on number of pairs of teeth being in mesh the damping of vibrations of toothed wheels of toothed gear increases together with increasing of load.

The value of damping changes together with change of rigidity of mesh during operation of the toothed gear. The relative value of change amplitude remains constant.

REFERENCES

[1] Brauer J.: Rheonome Schwingungserscheinungen in evolventenverzahnten Stirnradgetrieben. Dissertation, TU Berlin, 1969. "Rheonomic vibration phenomena in involute spur gearrs".

[2] Friedrich G.: Experimentelle Untersuchung der inneren dynamischen Zusatzkräfte bei geradverzahnten Stirnrädern. Dissertation, TU Dresden 1970. "Experimental investigations of internal dynamic additional forces by spur gears".

[3] Kubo A.: Untersuchungen über das dynamische Verhalten von Hochgeschwindigkeitgetrieben. Dissertation, Universität Kyoto 1971. "Investigations of dynamic behaviour of high-speed gears with spiral toothing".

[4] Pagel J.: Innere dynamische Kräfte von einstufigen Stirnradgetrieben mit Schrägverzahnung. Dissertation, TU Dresden 1972. "Internal dynamic forces of single-stage spur gears with spiral toothing".

[5] Rettig H.: Zahnkräfte und Schwingungen in Stirnradgetrieben. Konstruktion 17,41-53, 1965. "Teeth forces and vibrations of spur gears".

[6] Müller L.: Przekładnie zębate - dynamika. WNT, Warszawa 1986. "Dynamic of toothed gears".

[7] Wilk A.: Wpływ parametrów technologicznych i konstrukcyjnych na dynamikę przekładni o zębach prostych. Politechnika Śląska. Zeszyt Naukowy - Mechanika 72, 1981. "Influence of process and design parameters on dynamic of spur toothed gears".

[8] Kücükay F.: Über das dynamische Verhalten von einstufigen Zahnradgetrieben. Dissertation, TU München 1981. "About dynamic behaviour of single stage toothed gears".

[9] Osiński Z.: Tłumienie drgań mechanicznych. PWN, Warszawa 1986. "Damping of mechanical vibrations".

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TŁUMIENIE DRGAŃ PRZEKŁADNI STOŻKOWEJ

S t r e s z c z e n i e

W artykule wskazano na przyczyny powodujące rozproszenie energii mechanicznej drgań kół przekładni zębatej. Przedstawiono wyniki badań doświadczalnych, których celem było określenie wpływu obciążenia oraz zmiany sztywności zazębienia na tłumienie drgań kół przekładni stożkowej. Przeprowadzona analiza wyników wykazała, że tłumienie zmienia się wraz ze zmianą sztywności zazębienia, wartość względna amplitudy zmian jest stała. Tłumienie rośnie wraz ze wzrostem obciążenia.

ДЕМПФИРОВАНИЕ КОЛЕБАНИЙ УГЛОВОЙ ПЕРЕДАЧИ

Р е з ю м е

В статье показаны причины вызывающие рассеяние механической энергии колебаний колес зубчатого редуктора. Представлено результаты опытных испытаний, целью которых было определение влияния нагрузки и изменения жесткости зацепления на глушение колебаний колес конусного редуктора. Проведенный анализ результатов показал, что глушение изменяется вместе с изменением жесткости зацепления, относительная величина амплитуды изменений - константа. Глушение растет вместе с ростом нагрузки.