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ANALYTICAL SUGGESTION OF STRESS ANALYSIS ON FATIGUE IN CONTACT OF THE CYCLOIDAL - VASCULAR GEARING SYSTEM

Summary. This paper describes the proposal of design strength calculations of the cycloidal gear on fatigue in contact. It forms the first part of the full suggestion of stress analysis cycloidal gear. Another part will include a calculation for fatigue in bend and calculation for single-shot load in contact and bending. The second part will be published separately.

Keywords. Cycloidal gears, fatigue in contact, load of gears.

ANALITYCZNY PROJEKT OBLICZENIA WYTRZYMAŁOŚCI ZMĘCZENIOWEJ W STYKU PRZEKŁADNI CYKOLOIDALNEJ

Streszczenie. Niniejszy artykuł opisuje projekt obliczenia wytrzymałości przekładani cykloidalnej na zmęczenie w miejscu styku. Stanowi pierwszą część kompleksowego projektowanego obliczenia wytrzymałości przekładni cykloidalnej. Kolejna część będzie zawierała obliczenie wytrzymałości zmęczeniowej w miejscu zginania oraz obliczenie dotyczące jednostkowego obciążenia w miejscu styku i zginania. Druga część zostanie opublikowana oddzielnie.

Słowa kluczowe. Przekładnie cykloidalne, zmęczenie stykowe, obciążenia przekładni.

1. INTRODUCTION

The content of the paper is to design strength calculation for fatigue in contact of the cycloidal - vascular gearing system (C - E gearing system), or C - E gearing. This contribution is part of solving the issue of special mechanisms of gear drives of the manufacturing technique.

2. SUGGESTION OF STRESS ANALYSIS

Stress analysis of C - E gearing system is made on the base of the analysis of forces presented in [3]. Similarly to involute gearing system, also in this case it is necessary to

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perform the calculation or gearing system check on fatigue in contact (Hertz pressure).

No binding standard STN, EN, ISO has so far been available for stress analysis or gearing system check for this purpose.

3. STRESS ANALYSIS IN CONTACT

The basic calculation of *C - E* gearing system in contact is given by the Hertz contact stress in the points of engagement – contact of the pin or the cylinder of the pin wheel and the tooth of cycloid wheel. The size of the Hertz stress (pressure) σ_H in contact is

$$\sigma_H = \sqrt{\frac{\dfrac{K_H F_N}{b_v}\left(\dfrac{1}{\rho_1}+\dfrac{1}{\rho_2}\right)}{\pi\left(\dfrac{1-\mu_1^2}{E_1}+\dfrac{1-\mu_2^2}{E_2}\right)}}, \tag{1}$$

where

K_H – coefficient of dynamic load in contact. For its determination it is suitable to use the procedure in accordance with [60],
F_N – normal force in the point *N* of the contact (Fig. 1.) [N],
b_v – tooth system width [m],
ρ_1 – radius of pin, cylinder curvature [m],
ρ_2 – radius of cycloid wheel tooth curvature in the point *N* of engagement,
μ_1, μ_2 – Poisson material constants of pin and cycloid wheels,
E_1, E_2 – modules of flexibility of wheel material types.

Hertz stress (pressure) is checked mainly in the points of greatest load caused by normal force F_N between engaging teeth and in the point of the smallest reduced radius of curvature according to Hertz.

Maximum normal (pressure) force between the teeth is, as it has been presented for the calculation in bend, in the last point of engagement on addendum circle ka1 of pin wheel. In this point N there is also the smallest reduced radius of curvature ρ_H according to Hertz:

$$\rho_H = \frac{1}{\rho_1}+\frac{1}{\rho_2} = \frac{\rho_2+\rho_1}{\rho_1\rho_2} \tag{2}$$

where ρ_1 – radius of curvature of pin, cylinder and it is therefore equal to: $\rho_1 = \dfrac{d_c}{2} = m$ (3)

For the determination of the stress size σ_H it is necessary to determine the size of the radius of curvature ρ_2 of cycloid wheel tooth in the point of engagement. It is possible to determine the size of the radius of curvature mathematically. As it has been presented in [2], the side tooth curve profile is given parametrically by two equations:

$$x_e = x + r_e \sin\alpha, \qquad y_e = y - r_e \cos\alpha.$$

If the tooth curve profile is given parametrically, then it is known from mathematics that the radius of curvature ρ_2 in given point of engagement N can be described by the following expression:

$$\rho_{2N} = \frac{\left(x'^2 + y'^2\right)^{\frac{3}{2}}}{\left|y''x' - y'x''\right|}, \tag{4}$$

where x', x'', y', y'' – are the first- and second-order derivatives of parametrical equations (3) in the point N.

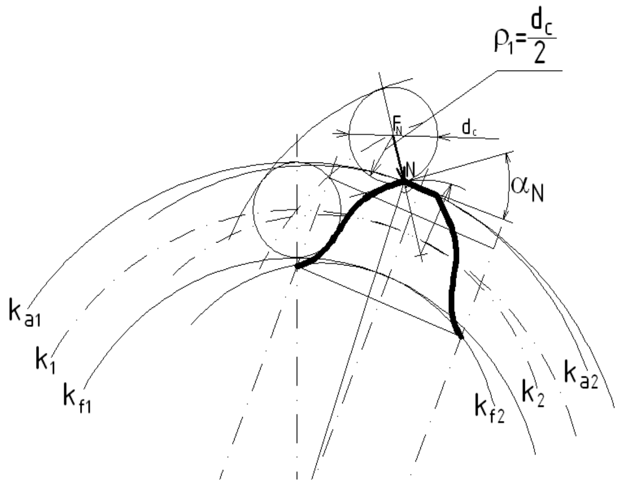

Fig. 1. Scheme of cycloid wheel load in contact
Rys. 1. Schemat cykloidalnego obciążenia zęba

Considering relative complexity of the expressions after the derivative of the equations, it is suitable to use information technologies with relevant software for the calculation of the radius of curvature ρ_2.

The process of the calculation and check of σ_H on spacing circles and in the area of tooth heel should be analogical. The check of the gear system in contact in the point N will then be performed according to the relation:

$$\sigma_H = \sqrt{\frac{\frac{K_H F_N}{b_v}\left(\frac{2}{d_c} + \frac{1}{\rho_{2N}}\right)}{\pi\left(\frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2}\right)}} \le \sigma_{HD}. \tag{5}$$

The expression in denominator under the root expresses mechanical characteristics of used material in gear system. If material coefficient z_M equals:

$$z_M = \sqrt{\frac{1}{\pi\left(\frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2}\right)}}, \quad \text{then} \quad \sigma_H = z_M \sqrt{\frac{K_H F_N}{b_v}\left(\frac{2}{d_c} + \frac{1}{\rho_{2N}}\right)} \le \sigma_{HD} \tag{6}$$

The expression $\frac{2}{d_c}$ in the above given relation can be, considering (7), replaced by the expression $\frac{1}{m}$. Then: $\sigma_H = z_M \sqrt{\frac{K_H F_N}{b_v}\left(\frac{1}{m} + \frac{1}{\rho_{2N}}\right)} \le \sigma_{HD}$. (7)

Permitted stress in contact σ_{HD} can be calculated from fatigue limit in contact divided by the safety coefficient in contact for selected material of C - E train of gears.

$$\sigma_{HD} = \frac{\sigma_{H\,lim}}{S_{H\,lim}}, \tag{8}$$

where

σ_{HD} – is permitted stress on fatigue in contact,

$\sigma_{H\,lim}$ – fatigue limit of selected material in contact after determined number of load cycles [60],

$S_{H\,lim}$ – safety limit in contact. For the detailed analysis of dynamic character of operation and other influences it is possible to select $S_{H\,lim} = 1{,}1 \div 1{,}2$. For common design it is $S_{H\,\mathrm{lim}} \approx 1{,}3$

4. CONCLUSION

This proposal of strength calculation of cycloidal gearing will be verified by further research - development experimental methods in the Department of Technical Devices Design. We assume that the proposal of strength calculation will be have positive contribution of solution for special gearing on based of cycloidal gearing. This paper was prepared and published with the support of grant KEGA no 058TUKE – 4/2012.

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