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A Study of New Cycloid Swing Link Speed Reducer by using Algorithmic Design

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Abstract: ***The aim of this article is to present a design algorithm for a new modern and high mechanical efficiency cycloid swing link speed reducer. Estimate that the new cycloid swing link speed reducer has and others advantages in comparison with the existing push road reducer such as: strong capacity of anti impact and over load, accessible process technology. We propose an algorithm for kinematics, dynamics and resistance (dimensional) calculus for specific main portent elements. Dimensional design of common elements as: shafts, bearings, carcasses, etc. not offer in the paper.***

Keywords: design, algorithm, cycloid swing link speed reducer, kinematics, dynamics and dimensional calculus**.**

1. Introduction

This document presents an algorithmic design for a cycloid swing link speed Reducer. According with our past research, authors propose the formulae for design the mains elements of the swing link cycloid pin-toothed speed reducer [1-8].

The cycloidal style of speed reducer is generally used in numerous industries for the purpose of power transmission applications. This type of mechanism, known for its high torque density and extreme shock load capacity, incorporates a unique reduction mechanism, which is different from that of the more commonly understood involutes gearing [9 and references therein].

To make out the technical payback of the cycloidal reduction mechanism, one needs to understand the forces, load distribution and contact stresses associated with the reduction components within the mechanism. This type of study is also essential in design optimization processes to improve the overall performance of the reducer [9-16].

2. The Structure and Driving Principle

For the structure and driving principle, let us consider one swing link speed reducer. An important scheme of the new type of swing link speed reducer has been shown in Figure 1 (a-b).

In this figure (Fig. 1 (a-b)), the part 1(a) represented to the scheme of longitudinal section and the part 1(b) is for the scheme of transversal section.

We have assigned some numerical numbers (e.g. 1, 2, 3…..) at appropriate place in Fig.1 (a & b) for better understanding of its mechanics. And their details are such as, the input shaft is united with the surge wheel, materialized as an eccentric bearing (1), swing link (2), driven ring (3), fixed on the output shaft (3'), annulus cycloid internal gear (motionless) wheel (4), outside roller (5) and inside roller (6).

To achieve output reach static balance, two identical shape sets of surge wheel and annulus are employed in the speed Reducer; make an angle of 180^0 each other.

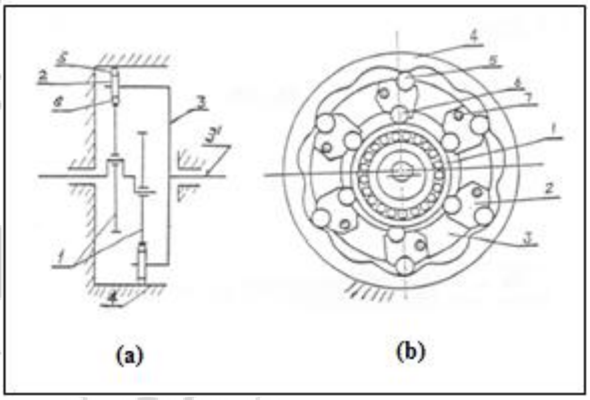

Figure 1 (a-b): The scheme of the swing link speed reducer

The transmitting of rotation movement process for the swing link speed reducer may be presented as: when surge wheel (1) have an angular speed 1, the swing link (2) swings back and forth around the pillar pin (7).

The pillar pin drives driven ring (3) rotating in the same time, because the inside roller (6) is driven by surge wheel and outside roller (5) of swing link meshes whit the inner teeth of annulus (4).

If the number of swing elements is n_2 and we notice z_4 as the number of tooth of annulus, for single surge speed reducer, general functional condition is:

$$n_2 = (z_4 \pm 1) / k, \text{ where } k = 1, 2, 3 \ldots \quad (1)$$

Usually, the value of "k" i.e., (k = 2) has been taken into account as a convenience solution (see Fig. 1 (a-b)).
When the rotational direction of the driven ring (3) is the same as surge wheel (1) takes positive sign; otherwise the formula above takes negative sign.

3. The Transmitting ratio

For single surge wheel, when annulus cycloid internal gear wheel (4) motionless, input shaft have an angular speed 1 and output shaft an angular speed 3, the general expression of transmitting ratio are:

$$i_{34}^{4} = \frac{\omega_1}{\omega_3} = \frac{z_4}{k \cdot n_2 - z_4} = \pm z_4 \qquad (2)$$

4. Calculus of tangential forces in the swing link speed reducer

Notice P_1 [kW] power speed and $\omega_1[s^{-1}]$ angular speed of the surge wheel (1), then input torque is:

$$M_{r1} = \frac{10^6 \cdot c_d \cdot P_1}{\omega_1} \; [N \cdot mm] \qquad (3)$$

where: the numerical value for dynamical coefficient, c_d we take from [1] Table 7.2. at page 342, between $c_d = 1.3 \div 5.0$.

The output torque actuate on the driven ring (3) is:

$$M_{t2} = M_{t1} \cdot i_{13} \cdot \eta \; [N \cdot mm] \qquad (4)$$

where: η = 0.80 ÷ 0.95 is mechanical meshing efficiency.
The tangent force actuate on one pillar pin (7) is:

$$F_{t7} = \frac{M_{t2}}{r_{b7} \cdot n_2} \; [N] \qquad (5)$$

where: r_{b7} [mm] is the circle radius of the laying center pillar pin (7); η_2 means the number of the pillar pin of the swing link identical with the number of swing elements, the number outside roller (5) and the number of inside roller (6).
The tangent force actuate on one outside roller (5) is:

$$F_{t5} = \frac{M_{t2}}{r_{b5} \cdot n_2} \; [N] \qquad (6)$$

where: r_{b5} [mm] is the circle radius of the laying center outside roller (5).
The tangent force actuate on one inside roller (6) is:

$$F_{t6} = \frac{M_{t2}}{r_{b6} \cdot n_2} \; [N] \qquad (7)$$

where: r_{b6} [mm] is the circle radius of the laying center outside roller (6).
The tangent force actuate on one tooth of the annulus wheel (4) is:

$$F_{t4} = \frac{M_{t2}}{r_4 \cdot z_4} \; [N] \qquad (8)$$

where: r_4 [mm] is the medium circle radius of the tooth of the annulus wheel (4); z_4 - the number of tooth of annulus.

5. The Hertzian stresses between support rolling and their ways of the eccentric bearing

The total radial force on the eccentric bearing (1) (as we see Fig.1) is:

$$F_r = \frac{7 \cdot 5 \cdot 10^5 \cdot c_d \cdot \eta \cdot P_1 \cdot \left(i_{13}^{4} + 1\right)}{e \cdot z_4 \cdot \omega_1} \; [N] \qquad (9)$$

where: η = 0.80 ÷ 0.98 represent the mechanical efficiency of gearing; "e" means the eccentricity of the surge wheel (1).
The eccentricity of surge wheel (1) may be established with expression:

$$e = \lambda \cdot r_4 / z_4 \; [mm] \qquad (10)$$

where λ = 0.50 ÷ 0.85 is the eccentric coefficient.

The radial force for one roller or pillar pin is:

$$F_{r1} = F_r / n_2 = \frac{7 \cdot 5 \cdot 10^5 \cdot c_d \cdot \eta \cdot P_1 \cdot \left(i_{13}^{4} + 1\right)}{e \cdot z_4 \cdot \omega_1 \cdot n_2} [N] \qquad (11)$$

where: n2 is the number of swing link (2), equal with the number of inside roller (6) and the number of outside roller (5).
The general relation to verify hertzian stresses between cylindrical support rolling and their rolling ways is:

$$\sigma_{Hr} = 1.672 \left[\frac{F_{r1} \cdot E_1 \cdot E_2}{l_r \cdot d_r \cdot \left(1 \pm \frac{d_r}{D_r \mp d_r}\right) \cdot (E_1 + E_2)} \right]^{\frac{1}{2}} \le \sigma_{HP} [MPa] \qquad (12)$$

where E_1, E_2 [MPa] are the Young module of the roller and their rolling ways material; l_r[mm], d_r[mm] are the length, the diameter of roller, D_r[mm] is the diameter laying circle of axis cylindrical roller. σ_{HP} =550 ÷ 650 [MPa] is the admitted resistance to hertzian stresses for usual steel.

If the roller and their rolling ways material is same $E_1 = E_2 = E$, the relation (11) becomes simple as:

$$\sigma_{Hr} = 1.672 \left[\frac{F_{r1} \cdot E}{2 l_r \cdot d_r \cdot \left(1 \pm \frac{d_r}{D_r \mp d_r}\right)} \right]^{\frac{1}{2}} \le \sigma_{HP} [MPa] \qquad (13)$$

6. The Hertzian stresses between eccentric bearing and the inside roller

The total radial force on the eccentric bearing (1) transmit to the inside roller (6) and produce a hertzian stresses, then the relation to verify these stresses is

$$\sigma_{Hr} = 0.836 \left[\frac{F_{r1} \cdot E_1 \cdot E_2 (d_{eb} + d_{ir})}{b_{eb} \cdot d_{eb} \cdot d_{ir} \cdot (E_1 + E_2)} \right]^{\frac{1}{2}} \le \sigma_{HP} [MPa] \qquad (14)$$

where d_{eb} and d_{ir} [mm] are the outside diameter for eccentric bearing (1) and of the inside roller (6) respectively.

If the eccentric bearing and the inside roller material is same $E_1 = E_2 = E$, the relation (13) becomes simple as:

$$\sigma_{Hr} = 0.836 \left[\frac{F_{r1} \cdot E \cdot (d_{eb} + d_{ir})}{2 \cdot b_{eb} \cdot d_{eb} \cdot d_{ir}} \right]^{\frac{1}{2}} \le \sigma_{HP} [MPa] \qquad (15)$$

7. The Hertzian stresses between the outside roller and tooth profile of the annulus

One outside roller in contact with a tooth cycloidal profile of the annulus transmits a normal forces "F_n" as a resultant of the radial F_{r1} and the tangential force F_{t5} , and

$$F_n = \sqrt{F_{r1}^2 + F_{t5}^2} \quad [N] \tag{16}$$

where F_{r1} has (11) formula and

$$F_{t5} = \frac{10^6 \cdot c_d \cdot P \cdot i_{13}^4 \cdot \eta}{\omega_1 \cdot r_{b5} \cdot n_2} \quad [N] \tag{17}$$

The relation to verify the hertzian stresses between the outside roller and tooth profile of the annulus is:

$$\sigma_{Hr} = 0.836 \left[\frac{F_n \cdot E_1 \cdot E_2 (d_{er} + 2\rho_{t4})}{b_{er} \cdot d_{er} \cdot \rho_{t4} \cdot (E_1 + E_2)} \right]^{\frac{1}{2}} \leq \sigma_{HP} [MPa] \tag{18}$$

where: b_{er}, d_{er}[mm] are the length and the diameter of de outside roller (5); ρ_{t4} is curvature radius of the profile annulus tooth in the contact point. If $E_1 = E_2 = E$, then formula (18) becomes:

$$\sigma_{Hr} = 0.836 \left[\frac{F_n \cdot E \, (d_{er} + 2\rho_{t4})}{2 \cdot b_{er} \cdot d_{er} \cdot \rho_{t4}} \right]^{\frac{1}{2}} \leq \sigma_{HP} [MPa] \tag{19}$$

8. Some Considerations on Resistance Calculus of Main Portent Elements

For design of main portent elements the previous relations to verify the hertzian stresses may be covert in dimensional expressions use the last inequality in each formula.

For example:-

(a) The Diameter of laying circle of axis cylindrical roller D_r [mm] for eccentric bearing (1) may be use formula (12) or (13):

$$D_r = d_r \cdot \left[1 + \frac{2 \cdot l_r \cdot d_r \cdot \sigma_{HP}^2}{2.8 \cdot F_n \cdot E - 2 \cdot l_r \cdot d_r \cdot \sigma_{HP}^2} \right] \tag{20}$$

(b) The Diameter d_{eb}[mm] for eccentric bearing (1), or diameter of inside roller d_{ir}[mm], using formula (14) or (15):

$$d_{eb} = \frac{0.35 \cdot F_{r1} \cdot E \cdot d_{ir}}{b_{eb} \cdot d_{ir} \cdot \sigma_{HP}^2 - 0.35 \cdot F_{r1} \cdot E} \quad [mm] \tag{21}$$

(c) The Diameter or the length of de outside roller (5) d_{er} [mm], b_{er} [mm] using formula (19):

$$d_{er} = \frac{0.7 \cdot F_n \cdot E \cdot \rho_{t4}}{b_{er} \cdot \rho_{t4} \cdot \sigma_{HP}^2 - 0.35 \cdot F_n \cdot E} \quad [mm] \tag{22}$$

The designer may use the standard dimensions of diameters, and of lengths of rollers, see [1], Table 7.4, pg. 343. Than the precedents verifying formulae are used in showing forms in the present research work.

9. Results and Discussions

In detailed calculus we defined an algorithm of the mains element of the new type swing link speed reducer.

10. Conclusions and Final Remarks

The author's researches verify the Hertzian stresses between the mains elements of a new type swing link cycloid pin-tooth speed reducer. This partial algorithm calculus can be successfully application for a new modern speed reducer and offered to designers and specialists in industrial domains.

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