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OPTIMIZATION OF PARAMETERS OF HELICON DRIVES ON THE BASIS OF THE TRIBOLOGY CONDITIONS

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Abstract

The relationships being derived by the kinematic method make possible the digital simulation of the meshing conditions, the exact determination of the curvature of the meshing tooth surface during the engagement, the analysis of the conditions being necessary to avoid the undercut of the tooth surface on the face-typed gear, and the numerical analysis of the effect of the geometrical parameters. The positions of the line of contact change to a significant extent due to the position of the worm. The angle enclosed by the tangent of the pitch line in the point of contact with the vector of the relative velocity served for the qualitative estimation of the position of the pitch lines.

Since the magnitude of the angle enclosed by the contact line and the relative velocity vector is suitable only for an approximate estimation of the helicon drives. Complex kinematic and geometrical indicatrixes including the velocity and curvature, and characterising the hydrodynamical conditions served for the detailed analysis of the toothing data and for optimization of the position of the worm.

Keywords: Gear, drives, tribology condition, load carring capacity.

Introductions

Since Azor Robins publication in 1952, several researchers have been concerned with helicon drive consisting of a cylindrical worm and a face-type gear. However, their investigations covered primarily only cylindrical worms cut with straight sided V-shaped tools. Because of the complicated connecting conditions in space their analysis were generally obtained from approximated connections with the plotting-calculating procedures.

Considering that the worm of a new, modern drive is made almost exclusively with grinding and the most wide-spread, simplest technology of the grinding of the worm is working with a conical grinding-wheel, it was necessary to investigate the helicon drives having a worm ground with a conical grinding-wheel (*Fig. 1*)

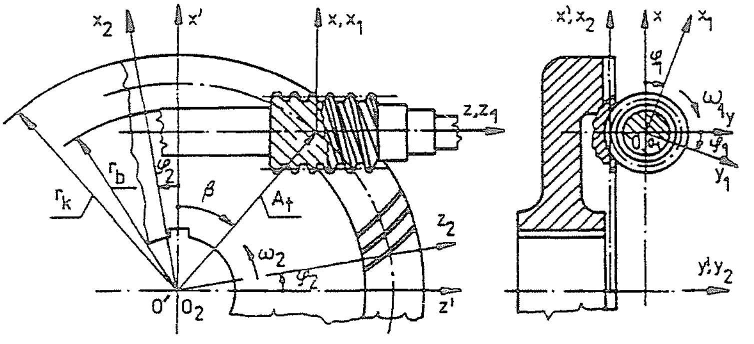

Fig. 1. Helicon drive

A Model for Investigation of the Helicon Drives

According to the HD lubrication theory the complex kinematic-geometrical indicatriexes including the velocity and curvature conditions, characterizing the hydrodynamical state can be deduced. because of approximations applying during demonstration these figures are not suitable for numerical determinations of load-carrying capacity of the helicon drives, but they can be well-used for analysis of the effect of geometrical parameters on the contact-conditions conclusions concerning the teeth, as well as for optimization of the toothing parameters.

Along the contact-line with the length of ds of the helicon drives the normally directed forces belonging to the $h = h_0$ minimal thickness of oil-film can be given by the formula developed by Niemann and Weber from HD theory:

$$dF_{HD} = 2.45 \cdot \eta \cdot \frac{1}{h_0} \rho_{\text{red}} \cdot |\underline{v}^{(12)}| \cdot \sin \delta \cdot ds \tag{1}$$

In order to compare the face load-carrying capacity, Hertz-type equations of contact stress can be accepted taking notice that besides the Hertz-stress the face load-carrying capacity is influenced by other factors such as the directions of relative motion, the velocity of it, the pressure-distributions of the lubricant, and so an. Using the concept of Stribeck type k_H factor the ds elementary part of the contact line is loaded with the limit of the face-load carrying capacity of

$$dF_H = 2 \cdot k_H \cdot \rho_{\text{red}} \cdot ds \tag{2}$$

where dF_H is normally directed force.

On the axis of the face-type gear the useful stress load, which corresponds to elementary loads determined by criteria of single load carrying capacity, is given by

$$dM_2 = (j' \cdot e' \cdot r') \cdot dF \tag{3}$$

relationship. Integrating this expression along the parts being in the contact field of the contact lines, summarizing the momentum values calculated for stages in simultaneous contact, for the given moment limit loading of drive the momentum can be given in the following formula:

$$M_2 = \sum_{i=1}^{n} \int_{s_1}^{s_2} ((j' \cdot e' \cdot r') \cdot dF \tag{4}$$

Replacing the geometry relationships for helicon drive into (1) - (4) equations, and introducing the surface parameter instead of the arch-length parameter, according to the criteria of single load-carrying capacity the output momentum expressing the load-carrying capacity can be determined by the formulas:

$$M_{2HD} = 2.45 \cdot \eta \cdot \frac{1}{h_0} \sum_{i=1}^{n} \int_{\xi^{(1)}}^{\xi^{(2)}} \rho_{\text{red}} \cdot |\underline{v}^{(12)}| \cdot \sin\delta \cdot T \cdot S \cdot d\xi \tag{5}$$

$$M_{2H} = 2 \cdot k_H \cdot \sum_{i=1}^{n} \int_{\xi^{(1)}}^{\xi^{(2)}} \rho_{\text{red}} \cdot T \cdot S \cdot d\xi \tag{6}$$

where T and S are the factor depending on the geometry parameters of the drive.

Using the characteristic length of the helicon drive, that is the A_t technology axis distance, from the (5) - (6) relationships the dimensionless kinematic-geometrical figures depending on the kinematic-geometrical characteristics of the drive can be deduced by the formulas:

$$Q_{HD} = \frac{h_0}{\eta \cdot \omega^{(1)} \cdot A_t^4} \cdot M_{2HD} \tag{7}$$

$$Q_H = \frac{1}{k_H \cdot A_t^3} \cdot M_{2H} \tag{8}$$

Investigations of the Contact-lines

Investigating the effect of the change of the worm position on the position of the contact lines it can be ascertained that

- the character, situations and number of the contact lines taking shape at the helicon drive are changing dependently on the quality of the change of β angle, as well as on the direction of the change compared to the middle-positions at $\beta = 0^0$.
- in the wider sense of the modification – in the case of the full change compater to the tooth-surfaces of the cylindrical worm gear drives – in the environment of y, z middle plain of the worm the localized field of contact can be form on each tooth-side of the drive (*Fig. 2*).

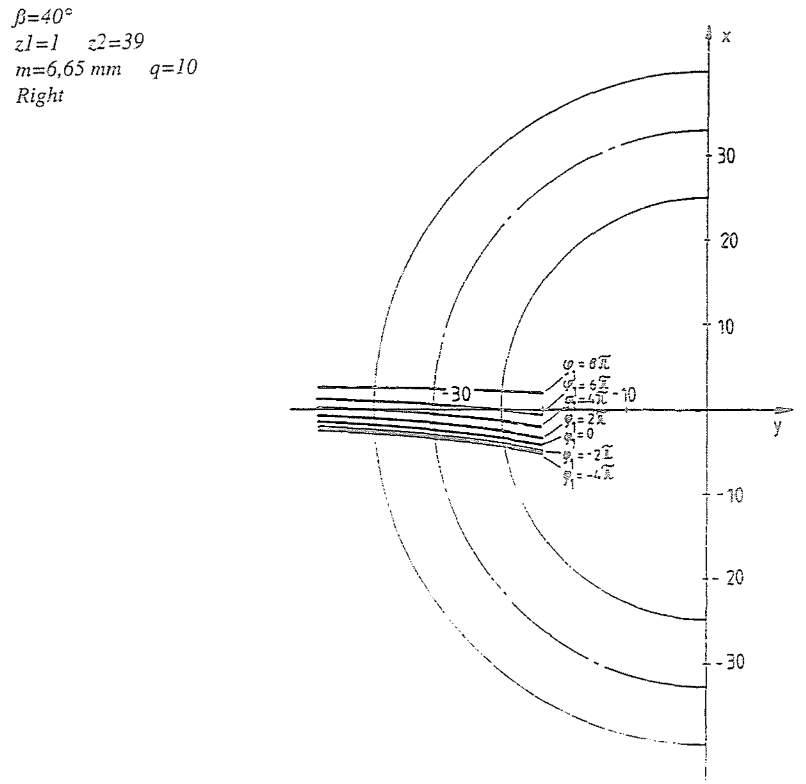

Fig. 2. Localized field of contact

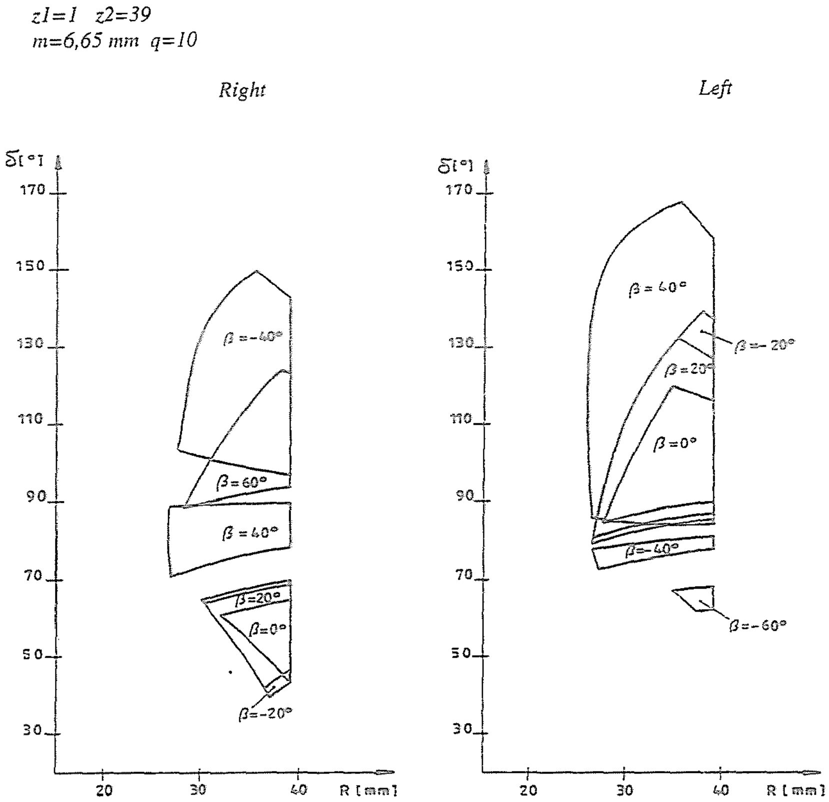

Fig. 3. The value of σ angle inside the field of contact

- in the case of helicon drives – with the suitable choice of the β angle – much more gear – tooth takes part in connecting, than in the case of the cylindrical worm gear drives.

The Optimization of Parameters of the Helicon Drive

To estimate unambiguously the effect of the position of the worm exerted on the load-carring capacity, the investigations have to be spreaded to two factors which influences essentially the load-carrying capacity. These factors are the σ angle enclosed by the tangent of the pitch line in the point of contact with the vector of the relative velocity, and the reduced curva-

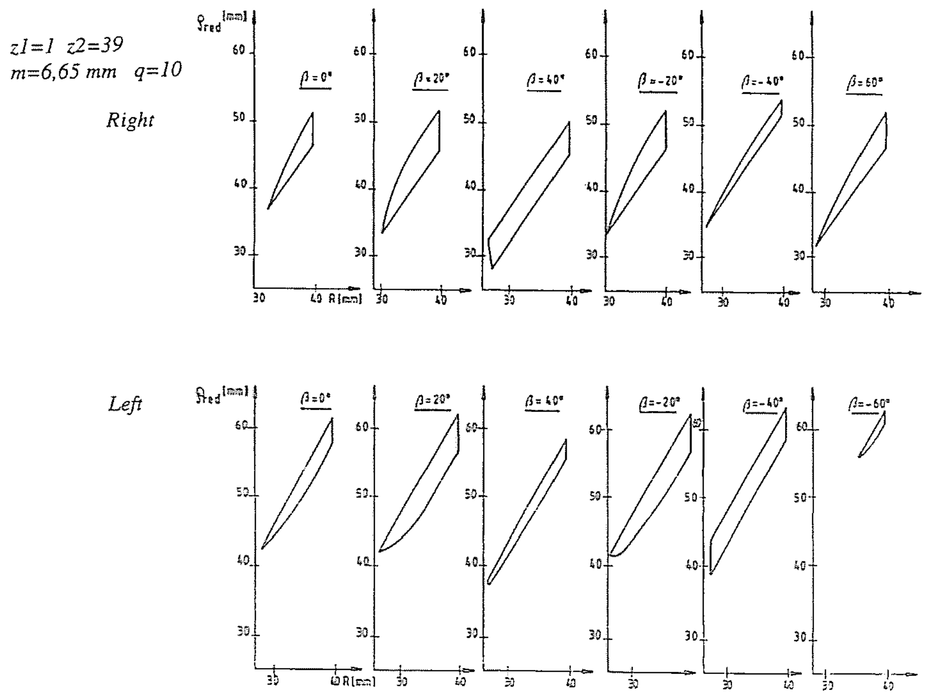

Fig. 4. The value of the reduced curvature radius inside the field of contact

ture radius, which is perpendicular to the contact characteristics. These investigations showed that, when the magnitude of the σ angle, as well as the sizes of the pitch line-length and of the field of contact are changing dependently on the position of the worm (*Fig. 3*), the change of the position of the worm results just small change in the magnitude of the reduced curvature radius (*Fig. 4*).

Furthermore it can be proved, that with the suitable choice of the position of the worm the favourable position of the pitch lines and of the field of contact from the point of view of hydrodynamics can be ensured.

The results of comparing investigation based on the dimensionless kinematic-geometrical indicatrixes deduced on the basis of the HD theory give possibilities to have conclusions concerning to the structural forming and the load-carrying capacity, to compare the same type drives from the aspect of toothing geometry, as well as to set up optimal criteria, because these indicatrixes do not depend on the factors influencing the load-carrying capacity such as revolution, oilviskosity, size of drive and so on, and they are constant quantities as those in the case of geometrically strictly similar drives.

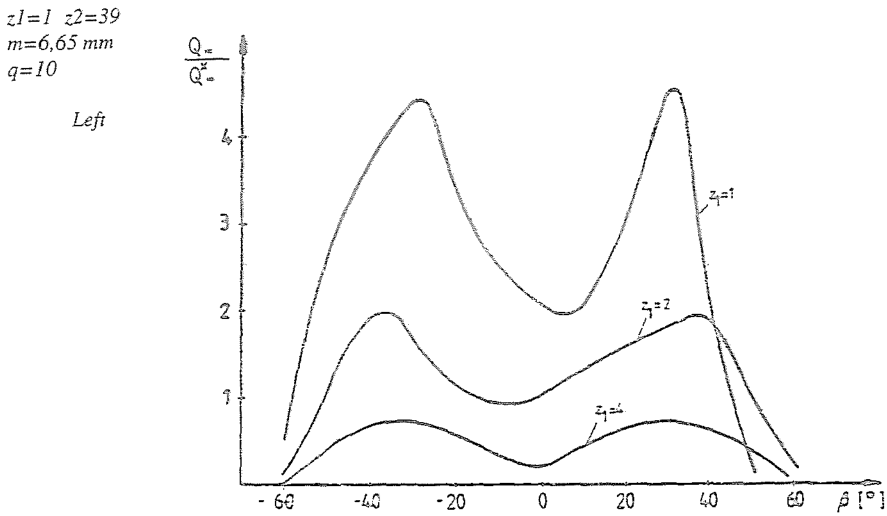

Fig. 5. The curve of the load-carrying capacity

The results of the camparing investigations the aim of which is the determinations of the optimal position of the worm, showed that from the point of view of load-carrying capacity the toothing parameters of the helicon drive can be optimalized, and the position of the optimum depends on the number of starts or threads of the worm (*Fig. 5*).

In the environment of $\beta = 0^0$ the hydrodynamic load-carrying capacity gets the minimal value, that means that the symmetrical building of the helicon drives results essential decrease in load-carrying capacity. It also has been proved that in the case of the helicon drives with similar basic and deduced geometrical parameters the load-carrying capacity-equality of the right and left tooth-side can be ensured just only in the positive range of β angle. From the point of view of the avoidance of the interference phenomena and to ensure the transmission of the performance in both senses of rotation, the optimal worm position is obtained between $35° - 45°$.

References

1. NIEMANN, G – WEBER, C.: Schneckentriebe mit flüssiger Reibung. VDI-Forschungsheft 412. Berlin, 1942.
2. BERCSEY, T.: Toroid hajtások elmélete. Kandidátusi értekezés. Budapest 1977.
3. LITVIN, F.L.: A fogaskerékkapcsolódás elmélete. Műszaki Könyvkiadó. Budapest, 1978.
4. HOANG, N.H.: Nem vonalfelületű helicon hajtások vizsgálata és optimalizálsa a hidrodinamikai teherbírás szempontjából. Doktori értekezés. Budapest, 1988.
5. HOANG, N.H.: Investigation of Contact Characteristics of Helicon Gears. V.Tribology conference Budapest, 1991.
6. VINH, N.D. - HOANG, N.H. - BERCSEY,T.: Testing Meshing Condition of Involute Helical Spur Gearand Globoidal Worm from the Point of View of Hydrodynamical Load Carraing Capacity. V.Tribology conference. Budapest, 1991.