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Assessment of Worm Gearing for Helicopter Transmissions

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ASSESSMENT OF WORM GEARING FOR HELICOPTER TRANSMISSIONS

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SUMMARY

This report assesses a high-efficiency hydrostatic worm gear drive for helicopter transmissions. The example given is for a large cargo helicopter with three 4000-kW engines and a transmission reduction ratio of 110. The report also contains an efficiency calculation, a description of the test stand for evaluating the feasibility of worm gear hydrostatic mesh, a weight calculation, and a comparison with conventional helicopter transmissions of the same power and transmission reduction ratio.

INTRODUCTION

Gear trains are extensively used in mechanical systems, but in no other transport vehicle is the relative weight of the transmission as great as in the helicopter. The transmission system is the helicopter's most complex and expensive assembly and requires the most labor-intensive maintenance. In addition, the weight of the transmission is comparable to that of the fuselage. Any substantial progress in the quality of helicopters will most likely come through improvements in their transmission systems. Worm gearing is very attractive for use in helicopter transmissions because it can transmit torque from the horizontal engine shaft to the vertical rotor shaft, reducing the revolutions by any ratio necessary in one stage. Transmission reduction ratios for existing helicopters average from 60 to 110. Using this otherwise advantageous worm gearing in helicopter gearboxes has long been prevented by a serious problem: low efficiency due to sliding friction between worm gear teeth and worm coils. Creation of a worm-gearing helicopter transmission is contingent upon reducing this friction. There are four ways to eliminate friction between two surfaces:

(1) By using ball or roller bearings
(2) By creating an aerostatic clearance between the planes
(3) By creating an electromagnetic clearance between the planes
(4) By creating a hydrostatic clearance between the planes

The highest load-carrying worm gear is the double-enveloping worm gear (globoidal worm gear or "cone drive" worm gearing, ref. 1). This study considers only the hydrostatic double-enveloping worm gear transmission. In this transmission, friction is decreased by pumping the oil through meshes between worm gear teeth and worm coils under high pressure. This high oil pressure creates a lift force that separates the surfaces of the worm gear teeth from the worm coils. The sum of these lift forces from several meshes creates a torque on the worm gear and an axial force on the worm. In order to react to this axial force, the worm needs a counteracting hydrostatic bearing (fig. 1). The efficiency of the hydrostatic worm gear transmission is determined by the

friction losses of all these hydrostatic meshes and the power losses of the oil pumping systems.

SYMBOLS

A	hydrostatic mesh area
A_l	land area
A_p	pocket area
B	slot width
F	load force per worm gear tooth
F_f	friction force in hydrostatic mesh
h	slot height (clearance)
K	volume of part
N	total power loss
N_f	friction power losses in hydrostatic mesh
N_p	pump power
n	shaft speed, rpm
Q	oil flow
P_p	oil pressure at pump
P_x	pressure needed to create lift force of 57 706 N on one tooth
R	worm gear radius
S	slot length
T	torque
V	velocity in hydrostatic mesh
W	weight
α_x	helix angle of worm
η	oil viscosity
ρ	pitch of worm

Subscripts:

c worm coil

d disk

g worm gear

opt optimum

t tooth

tot total

w worm

EXAMPLE: LARGE CARGO HELICOPTER TRANSMISSION

As an example let us assume a large cargo helicopter has three gas turbine engines (4000 kW each), and the engine shafts rotate at 14 280 rpm. The main shaft of the helicopter combines the power of these three gas turbine engines by using three input modules and three worm meshes (fig. 2). If the main rotor speed is 130 rpm, the transmission reduction ratio is 14 280/130, or 110. The main rotor torque per worm is

$$T = \frac{N}{n} = 288\ 531\ \text{N·m}$$

If the worm gear radius R is 1 m, the total force for one worm is

$$\sum F = \frac{T}{R} = 288\ 531\ \text{N}$$

Each worm gear mesh has five teeth in engagement; therefore, the load force per tooth F equals 57 706 N. The working surface of the worm gear tooth looks like a horseshoe; part of it is a hydrostatic pocket and the rest is land. The oil pressure is at its maximum in the pocket and gradually drops to zero from the pocket edge to the perimeter of the horseshoe (fig. 3).

Determination of Optimum Clearance, Efficiency, and Oil Viscosity

The optimum clearance, efficiency, and oil viscosity were calculated for the conditions that take place on a hydrostatic bearing test stand (see refs. 2 and 3):

Oil flow, Q = 10.5 gal/min (39 liters/min)

Oil pressure at pump, P_p = 5000 psi (350 bar)

Hydrostatic mesh area, $A = 87\ \text{cm}^2$

Pocket area (fig. 3), $A_p = 23\ \text{cm}^2$

Land area (fig. 3), $A_l = 64\ \text{cm}^2$

Slot width, $B = 1.5$ cm

Slot length, $S = 23.3$ cm

Slot height (minimum boundary film), $h = 0.25$ mm (0.025 cm)

Oil viscosity, $\eta = 50$ cP ($5\times10^{-6}\ \text{N}\cdot\text{s/cm}^2$)

Velocity in centerline of hydrostatic mesh, $V = 5800$ cm/s

The oil flow through a hydrostatic worm gear mesh is constant.

$$Q = \frac{P_p h^3 B}{12\eta S} = \text{Constant}$$

Hence

$$P_p h^3 = \text{Constant}$$

Pump power is given by

$$N_p = \frac{QP_p}{0.8}$$

where 0.8 is the efficiency of the pump.

$$N_p = \frac{P_p^2 h^3 B}{9.6\eta S}$$

Power loss due to friction in a hydrostatic gear mesh is

$$N_f = F_f V$$

where F_f force of friction in the hydrostatic mesh

$$F_f = \frac{\eta V A}{h}$$

or

$$N_f = \frac{\eta V^2 A}{h} = 4.15\ \text{kW}$$

One worm has five hydrostatic gear meshes and one hydrostatic counterforce bearing. If the friction losses in the counterforce hydrostatic bearing are three times greater than that of one worm gear mesh, the friction power losses of the worm will be

$$N_f = 4.15 \times 5 + 4.15 \times 3 = 33.2\ \text{kW}$$

The total power loss in a hydrostatic gear mesh equals the sum of the friction power losses and the pump power losses:

$$N = \sum N_p + \sum N_f$$

$$N = \frac{P_p^2 h^3 B}{9.6 \eta S} + \frac{\eta V^2 A_1}{h}$$

The geometrical sizes B, S, and A_1, the pressure P_p and the velocity are constant, and therefore

$$N = \text{Constant}\ \frac{h^3}{\eta} + \text{Constant}\ \frac{\eta}{h}$$

Clearance. - The optimum clearance (h_{opt}) can be found from a derivative of the functional dependence of the power losses on the clearance:

$$\frac{dN}{dh} = 0$$

$$\frac{P_p^2 B 3 h_{opt}^2}{9.6 \eta S} - \frac{\eta V^2 A_1}{h_{opt}^2} = 0$$

then

$$h_{opt} = \sqrt[4]{\frac{\eta^2 V^2 A_1 S\ 3.2}{P_p^2 B}} = \sqrt[4]{\frac{\left(5\times10^{-6}\right)^2 \times 5800^2 \times 64 \times 23.3 \times 3.2}{3500^2 \times 1.5}}$$

$$= 0.0216\ \text{cm}\ (0.22\ \text{mm})$$

Hence, for the given application the optimum mesh clearance is 0.22 mm (fig. 4).

Efficiency. - The worm gear mesh efficiency may now be calculated. The pump power is

$$N_p = \frac{P_x^2 h^3 B}{9.6 \eta S}$$

The pressure P_x needed to create a lift force of 57 706 N on one tooth of the worm gear can be found as

$$P_x A_p + \frac{P_x A_1}{2} = 57\ 706\ \text{N}$$

or

$$P_x = \frac{57\ 706}{23 + 32} = 1049\ \text{N/cm}^2$$

The pumping power required is

$$N_p = \frac{1049^2 \times 0.022^3 \times 1.5}{9.6 \times 5\mathrm{x}10^{-6} \times 22.3} = 16\ 419\ \mathrm{N\cdot cm/s}\ (0.16\ \mathrm{kW})$$

The engagement of a worm gear transmission involves five teeth and one counterforce bearing. This counterforce bearing uses three times more oil than one tooth of the hydrostatic worm gear mesh, and therefore pump power losses of the worm gear mesh can be calculated as

$$\sum N_p = (5 + 3) \times 0.16 = 1.28\ \mathrm{kW}$$

The total power losses on one worm, including friction and pump power losses, are

$$N = \sum N_f + \sum N_p = 33.2 + 1.28 = 34.48\ \mathrm{kW}$$

Engine power is 5350 hp, or 4000 kW. The efficiency of the hydrostatic worm gear transmission is

$$100 - \frac{34.48 \times 100}{4000} = 99.14\ \mathrm{percent}$$

Oil viscosity. - The optimum oil viscosity was determined as follows (ref. 2): The total power losses in the hydrostatic gear mesh were found as

$$N = \frac{P_p^2 h^3 B}{9.6 \eta S} + \frac{\eta V^2 A_1}{h}$$

The optimum oil viscosity can be found from a derivative of the functional dependence of the worm gear power losses on the oil viscosity at maximum worm gear efficiency $dN/d\eta = 0$ and then

$$- \frac{P_p^2 h^3 B}{\eta_{opt}^2 9.6S} + \frac{V^2 - A_1}{h} = 0$$

or

$$P_p^2 h^4 B = V^2 A_1 \eta_{opt}^2 9.6S$$

$$\eta_{opt} = \left(\frac{P_p^2 h^4 B}{V^2 A_1 S 9.6}\right)^{1/2} = 3\mathrm{x}10^{-6}\ \mathrm{N\cdot s/cm^2}$$

HYDROSTATIC WORM GEAR MESH TEST

The goals of a hydrostatic worm gear mesh test are:

(1) Determination of the optimum clearance h_{opt} corresponding to the highest efficiency of the hydrostatic worm gear mesh for tilt and parallel surfaces of a worm gear tooth

(2) Determination of the optimum position of a pocket on the surface of a worm gear tooth and the optimum shapes of the pocket and the "horseshoe"

(3) Determination of the optimum oil viscosity

All these parameters can be experimentally determined on a hydrostatic bearing test stand. The surface of the worm coils is a globoidal helix, where the helix angle gradually changes from coil to coil. The helix angle for any position of engagement (ref. 4) is

$$\alpha_x = \arctan \frac{\rho}{\pi d_x}$$

where ρ is the pitch of the worm. At the best efficiency the surfaces of the worm gear teeth should be parallel to the surfaces of coils -1 and 1 (fig. 1). In this case worm gear teeth have a minimum tilt angle with neighboring coils 0, -2, and 2 (fig. 1).

The efficiency test is conducted on a model representing a worm gear tooth (horseshoe). Two of the horseshoes are installed in a frame (caliper) facing the disk with clearance h (fig. 5). The high-speed rotating disk imitates a coil of the worm. High-pressure oil flow is pumped through the clearance between the disk and the horseshoes (fig. 5). The oil pressure of the hydraulic system and the oil flow will determine the pump power. The load force is measured by a strain gage on the caliper. The friction forces of the hydrostatic meshes are measured by a friction force gage. The horseshoes can be installed with different clearances and at different tilt angles to the rotating disk. Also, horseshoes of different shapes and with different pocket positions will be tested. The results of these tests will show the experimental dependence of the hydrostatic worm gear transmission efficiency on these factors.

Optimum viscosity tests must also be conducted. The theoretical viscosity was calculated as 3.00×10^{-6} N·s/cm^2. In a real situation the oil in the worm gear mesh will be heated as the power losses are converted to heat. This heat will be distributed among the coils of the worm, the worm gear teeth, and the oil film in the engagement. The oil film temperature will increase and the real viscosity will decrease. The test for experimental determination of the optimum oil viscosity should be conducted with constant clearance and oil flow but variable oil viscosity.

CALCULATION OF HYDROSTATIC WORM GEAR TRANSMISSION WEIGHT

The weight of a hydrostatic worm gear transmission strongly depends on the pressure of the hydraulic system. In some industrial hydraulic systems, oil pressure can be 30 000 psi (2000 bar). The following calculation of the worm gear transmission weight is made for a 15 000-psi (1000-bar) hydraulic pressure. Such high pressure occurs in the short line connecting the hydraulic cylinder and a worm gear tooth for a short time during the full-load takeoff of the helicopter. For the three-engine helicopter the power of one engine is 4000 kW and main rotor speed is 130 rpm. The worm gear torque is

$$T = \frac{4 \times 10^6 \times 60}{2\pi \times 130} = 288\ 218\ \text{N·m}$$

If the worm gear radius R is 0.75 m, the force from one worm is

$$\sum F = \frac{288\ 218}{0.75} = 384\ 290\ \mathrm{N}$$

This force is distributed among five worm gear teeth so that the force on one tooth is

$$F = \frac{384\ 290}{5} = 76\ 858\ \mathrm{N} = 7834\ \mathrm{kg}$$

Next we will calculate the area of a pad (horseshoe) when the hydraulic pressure is 15 000 psi (1000 bar) and the pocket area is 10 percent of the pad area A - the remaining 90 percent of the pad area being land. The load force created by hydrostatic pressure in the pocket is A_pP. The load force created by hydrostatic pressure in the land is $A_lP/3$. The load force for the entire pad is

$$A_pP + \frac{A_lP}{3} = 7834\ \mathrm{kg}$$

$$A_p = 0.1\ A;\ A_l = 0.9\ A;\ P = 1000\ \mathrm{kg/cm^2}$$

$$0.1\ A \times 1000 + \frac{0.9\ A \times 1000}{3} = 7834$$

$$A = 20\ \mathrm{cm^2}$$

The tooth volume (fig. 6) is given by

$$K_t = \frac{2\pi(R + r)}{2 \times 4}\ \frac{(3.5 + 0.5) \times 3}{2} = 42.4\ \mathrm{cm^3}$$

Hence, the volume of the 110 teeth is 4665 $\mathrm{cm^3}$. The volume of the rim will be three times more (14 000 $\mathrm{cm^3}$). The total volume of the worm gear thus is 18 665 $\mathrm{cm^3}$. If the worm gear is made of titanium (density, 4.5 $\mathrm{g/cm^3}$), the weight of the worm gear is W_g = 18 665 × 4.5 = 84 000 g (84 kg, or 185 lb)

The disk volume (fig. 7) is

$$K_d = 29\ 450\ \mathrm{cm^3}$$

The weight of the titanium disk is

$$W_d = 29\ 450 \times 4.5 = 132\ 000\ \mathrm{g}\ (132\ \mathrm{kg, or}\ 290\ \mathrm{lb})$$

The volume of a worm coil is four times greater than the volume of a worm gear tooth:

$$K_c = 42.4 \times 4 = 170\ \mathrm{cm^3}$$

The volume of a counterforce hydrostatic bearing is twice as great as the volume of one coil. A worm comprises five coils and one counterforce hydrostatic bearing. The total volume equals

$$K_w = 5K_c + 2K_c = 7 \times 170 = 1190 \text{ cm}^3$$

The worm shaft and bearings double the volume, and the weight of the steel worm (density, 7.85 g/cm^3) is given by

$$W_w = 2.1190 \times 7.85 = 18.683 \text{ g (18.7 kg, or 41 lb)}$$

Hydrostatic worm gearing comprises a worm gear wheel, a disk, and three worms. Its weight W_t equals:

$$W_t = 84 + 132 + 3 \times 18.7 = 272 \text{ kg}$$

The weight of the high-pressure hydraulic system cannot be estimated until the optimal design of the hydrostatic worm gear mesh is found. Assuming that the weight of the high-pressure hydraulic system doubles the weight of the transmission, we calculate that this transmission weighs 500 kg (1100 lb). Input modules of existing helicopters have transmission reduction ratios from 2.5 to 3.5. The input module ratio of the hydrostatic worm gear transmission equals 1, and hence it will weigh two-thirds as much as existing input modules. The following table compares the weights of various parts and units of the worm gear transmission with those of a representative baseline planetary transmission having the same power and reduction ratio. It is expected that the worm gear transmission weight will only be about 41 percent that of the baseline weight.

	Baseline planetary transmission weight, lb	Hydrostatic worm gear transmission
Housing and rear cover	1451	500
Planetary	2146	----
Worm gear and worms	----	1100
Bearings	292	255
Gimbal mount	103	----
Accessory gearbox	22	22
Rear cover accessory	150	150
Sumps and pumps	140	60
Input (right and left modules)	430	300
Input aft	532	150
Rotor drive	853	----
Scissors and standpipe	66	----
Total	6185	2537
Savings, percent	0	41

DISCUSSION

Almost all helicopter gearboxes have three or four stages of gear trains with one or two bevel gear stages and one or two spur gear (usually planetary) stages (i.e., many gears, bearings, shafts, and support parts). Gear trains are extensively used in mechanical systems, but in no other transport vehicle is the relative weight of the transmission as great as in the helicopter. In recent years there has been progress in improving the quality of helicopter gearboxes as manufacturing was improved and new technology and materials were developed. However, any new steps in this progress will be more and more difficult. Obviously, new energy transfer trains for improving the drive from the gas turbine to the main rotor should be investigated. A hydrostatic worm gear

transmission could be one such train, since it weighs less than a conventional gear train transmission. Also, a hydrostatic contact could have a longer service life and make less noise as well as provides an opportunity to use light metals such as titanium, aluminum, and composites. The case where one of the three engines is inoperable must be considered for a hydrostatic worm gear transmission -- in this case the worm gear must run the worm (i.e., the transmission must be reversible). Reversibility requires that the friction angle be less than the minimum helix angle of the worm. For the realization of this condition the worm gear tooth must have one more hydrostatic pad on the back side. This hydrostatic pad could be very small. The lift force of this pad need be only enough to run the worm, the shaft, and part of the overrunning clutch.

The hydraulic system of a hydrostatic worm gear transmission must use high oil pressure; a plunger oil pump (fig. 8) is best suited for this purpose. The hydraulic system of a hydrostatic worm gear transmission consists of one plunger per tooth mounted in the body of the worm gear. As the worm gear rotates, the plungers overrun a cam, which is an eccentric stationary mounted in the sector of the gear mesh. The plunger thus displaces oil in the gear mesh and creates hydrostatic clearance between the worm gear teeth and the worm coils.

CONCLUSIONS

The results of this analysis of a hydrostatic worm gear transmission demonstrate:

1. That it could have the same high efficiency as a conventional gear transmission

2. That its benefits include lighter weight, less noise, greater compactness, longer service life, and low manufacturing and maintenance costs.

Future research and development of this transmission could open new perspectives for improving power transfer trains. For example, using high oil pressure (to 30 000 psi) would allow a weight decrease or an efficiency increase in the hydrostatic worm gear transmission.

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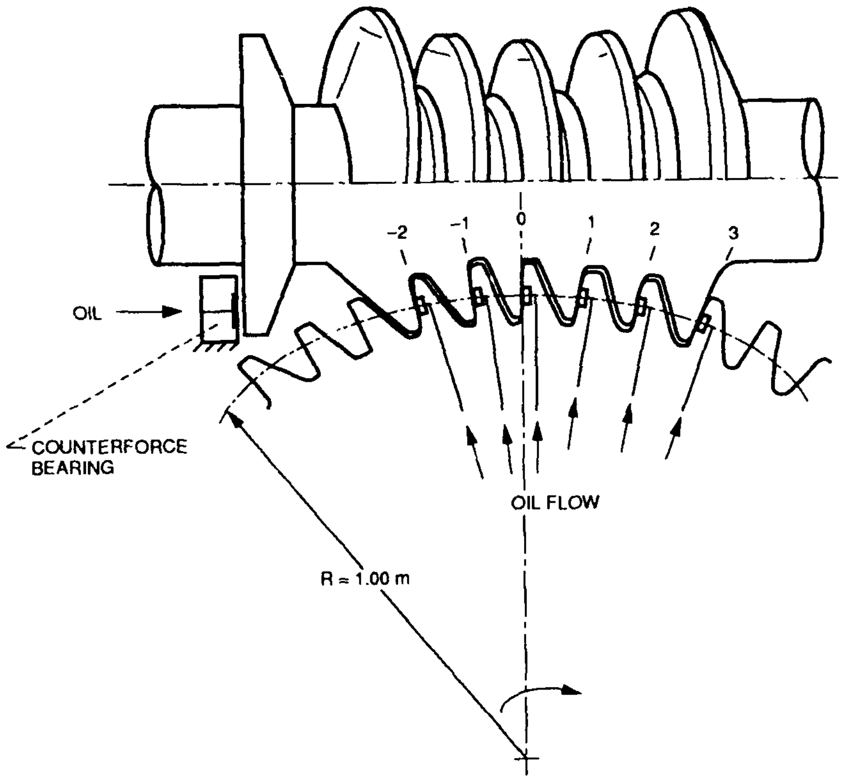

Figure 1. - Hydrostatic worm gearing engagement.

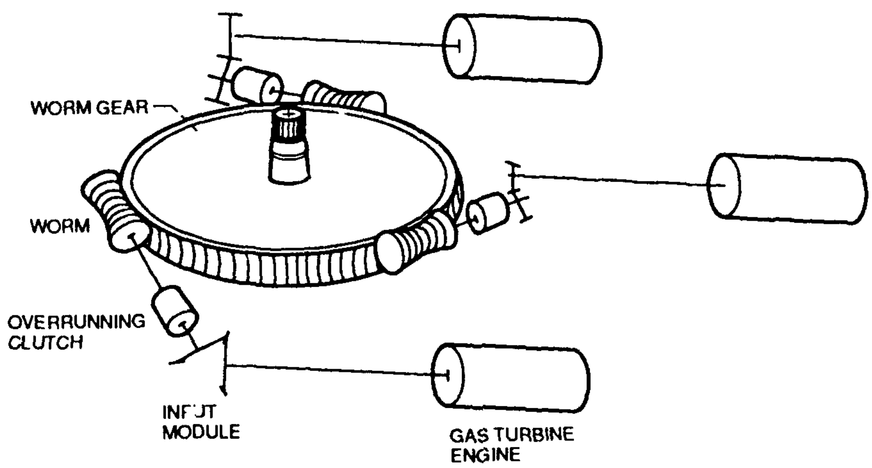

Figure 2. - Three engines and hydrostatic worm gearing helicopter transmission.

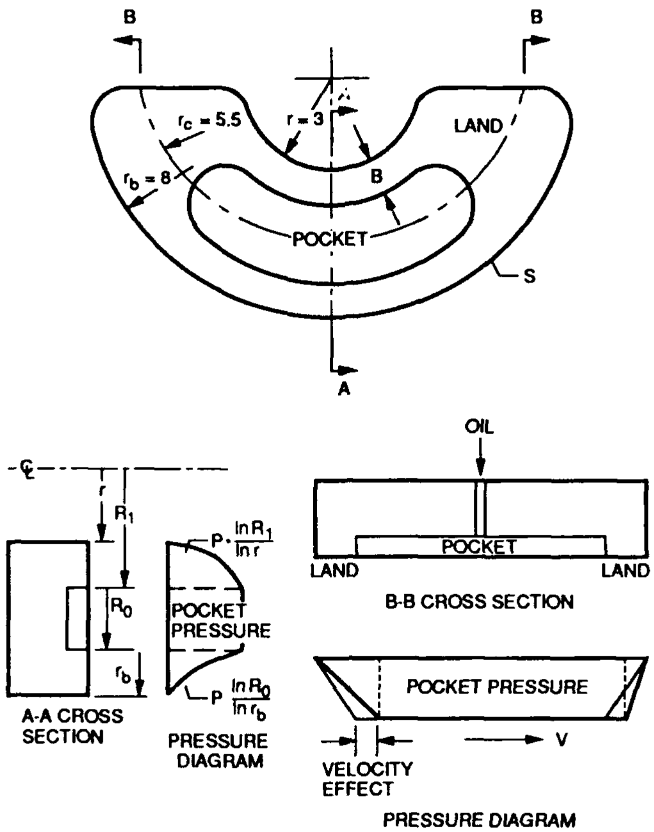

Figure 3. - Surface of worm gear tooth (horseshoe) and pressure diagrams.

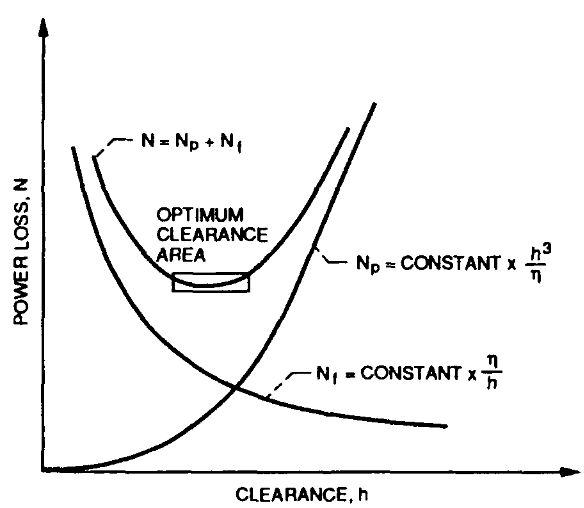

Figure 4. - Dependence of power losses on hydrostatic clearance.

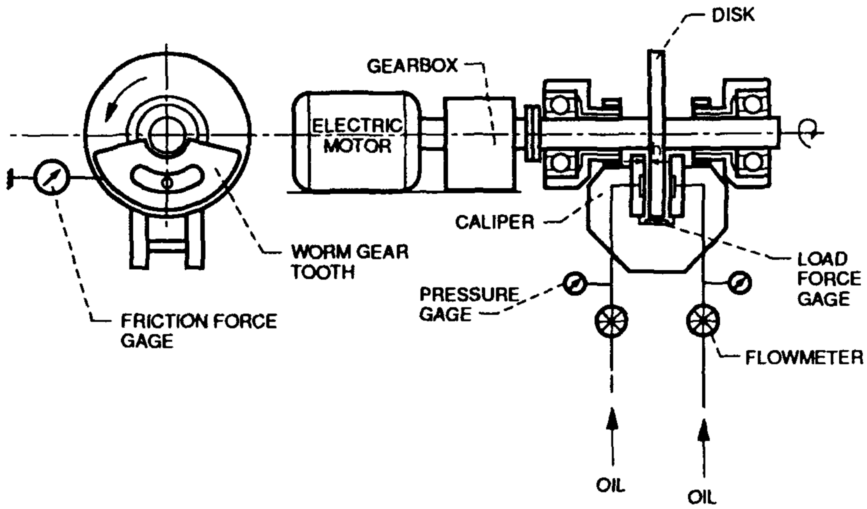

Figure 5. - Hydrostatic bearing test stand.

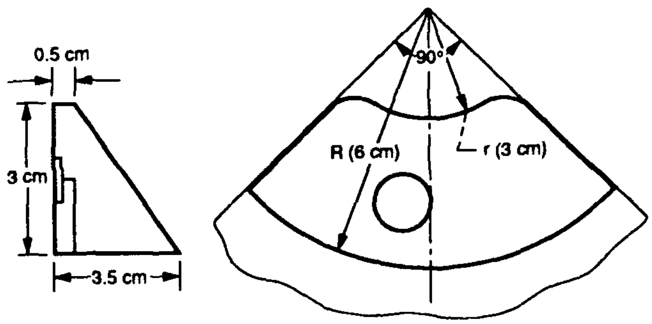

Figure 6. - Shape and cross section of worm gear tooth.

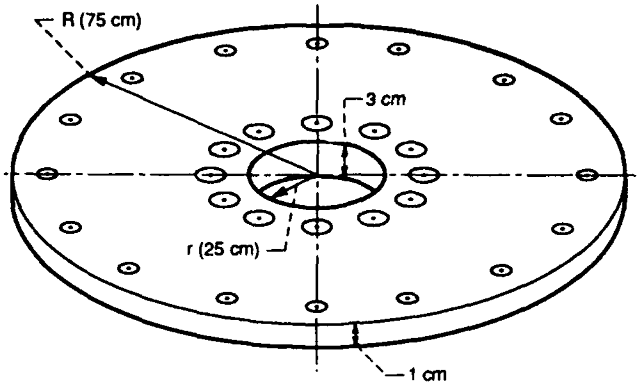

Figure 7. - Worm gear disk.

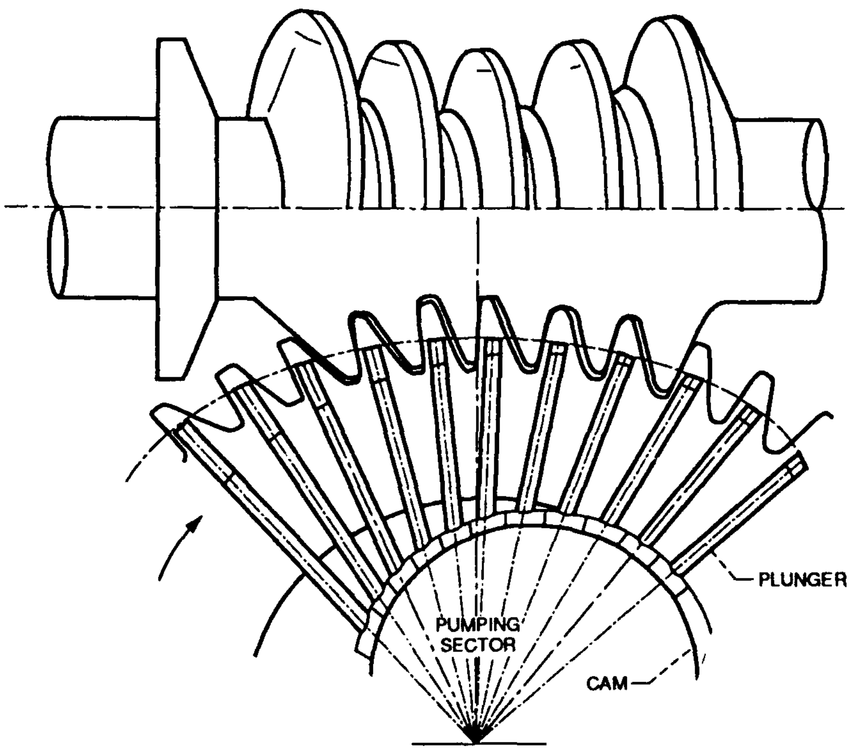

Figure 8. - Hydraulic system of hydrostatic worm gear transmission.

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15. Supplementary Notes

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