Design and Manufacture of Eccentric Planetary Gear Drive with Hydraulic Motor
(Power Transmission Efficiency of Trial Gear Reducer)

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Abstract
An eccentric planetary gear drive with hydraulic motor was developed, specifications of which were determined based on its service conditions as required for a relatively small-sized crawler-type conveyer for agriculture-use. The gear train was originally designed by arranging gears, including an eccentric planetary gear, and manufactured. Measured efficiency of power transmission of the gear drive by means of the torque balance method under quasi-steady state was approx. 0.92 at the highest, lubricated with hypoid gear oil, promising availability of the newly developed reduction gear drive in practical use. Also shown in operating test was a relatively high efficiency of 0.87 in speed increasing conditions.

Keywords: design, manufacture, planetary gear drive, eccentric gear, efficiency, power transmission

1 Introduction
Various types of gear drive are employed to increase or decrease speed for various machinery and transportation such as robot and automobiles.

In this study, an eccentric planetary gear drive with hydraulic motor was developed for irrigation, drainage and reclamation, featuring a simple and robust device with fewer components. For the requirement, a gear arrangement based on planetary gear with an eccentric gear is adopted and a detailed design is carried out. A prototype gear drive is tested to certify its performance.

2 Design concepts of the gear drive
The gear drive to be developed in this study is for a commercial crawler-type conveyer of relatively small-size used for agriculture-use. In addition to the usual requirements of high efficiency with high power for its light weight and small size, easy operation and maintenance is necessary as such a machinery tends to be handled roughly in the field regardless of weather. Also from a view point of mass production, less material and manufacturing costs, than used in a standard manufacturing environment and fewer components, and easy assembling are always essential.

3 Mechanism and principal features
Given that the gear drive developed here is planned to be attached to a crawler-type conveyer, it is preferable that this be an integral structure of a hydraulic motor and a speed decreasing device more than a power transmission device with a gear train driven by an engine. The hydraulic motor adopted here, a piston-type with swash plate, requires reduction of speed by 1/20, which is in an application range of planetary gears but relatively large for a single reduction.

Figure 1 (a) and (b) illustrate the outlook and the schematic drawing of the proposed eccentric planetary gear drive, which consists of an external gear assembled by two planetary gears G1 and G3 and internal gears of two independent gears [1], a type of differential gears. By replacing the eccentric planetary gear drive with a hydraulic engine for the conventional engine and reduction gear system, the number of parts can be reduced by 30%.

The revolution of the input shaft, an eccentric shaft,
4.2 Lubrication

The performance of the trial eccentric planetary gear drive was measured with five types of lubricant in bath lubrication: they are Oil A (Spindle oil, [10]), Oil B (Cylindrical gear oil, [63]), Oil C (Hypoid gear oil, [199]), Oil D (HW46, [48]), Oil E (SN/CF 5W-40, [94]), where the number in the square brackets indicates the dynamic viscosity in the unit of mm²/s at 40°C of each oil. The latter two oils were expected to be used in a practical application.

4.3 Experimental results and discussion

Figure 2 shows the test results in a speed decreasing condition. The gear drive gives an almost constant efficiency in the region of input torque greater than 1Nm. The efficiencies are almost the same with the two oils, Oil D and E, expected for practical use in wide operating conditions of input torque. Also in the figure, efficiencies are compared when other types of Oil A, B and C are used, clearly showing the importance of appropriate selection of oil for high performance. With Oil C, hypoid gear oil including various additives, the efficiency 0.92 is the highest improved by 3 percent compared to those with Oil D and E.

Shown in Fig. 3 is the efficiency of the EPGD used in a speed increasing condition by exchanging the input and output operations. The efficiency was acceptably high over a wide range of input torque and the highest efficiency of 0.87 was almost the same as that of the speed reduction obtained when Oil C is used. The results suggest EPGD developed in this study is also available in speed increase condition.

5 Theoretical efficiency of EPGD

It is still difficult to estimate the theoretical efficiency of a gear drive in power transmission with sufficient accuracy. This difficulty mainly arises from difficulty in precise estimation of magnitude of friction coefficient μ between teeth working under constantly changing normal load, contact stress and the sliding speed in meshing besides the deformation of teeth under
load is ignored.

In order to solve this problem provisionally, the friction coefficient $\mu_e$ at properly designated points on the tooth face is adopted as the representative value of $\mu$. For the sake of simple estimation, the deformation of gear teeth is to be ignored.

EPGD developed by the authors works under the power input to Gear 1 in Fig. 1 and the power output from Gear 4 through the drive shaft; its reduction mechanism consists of two stages. Here, in a usual way to derive the theoretical efficiency of this kind of multi-stage mechanism [3], the mechanism of this gear drive is divided into two stages and the efficiency in each stage of meshing between the internal and external gears derived individually as $\eta_{1st}$ and $\eta_{2nd}$. The overall efficiency $\eta_{CB}$ is estimated by combining the two efficiencies. Figure 4 illustrates the schematic diagram of reduction mechanism of the eccentric planetary gear drive for the derivation of theoretical efficiency below.

5.1 Theoretical efficiency in single gearing

The efficiencies in power transmission in meshing between the pair of external and internal gears in Figs. 5 and 6 is estimated as bellows [5] in approach contact and recess contact, $\eta_A$ and $\eta_R$ respectively:

$$\eta_A = 1 - \frac{\mu_e \lambda_{mean,2}(1/R_1 - 1/R_2)}{\cos \alpha + \mu_e (\sin \alpha - \lambda_{mean,2} / R_2)} \quad \ldots \ldots \ldots (3)$$

$$\eta_R = 1 - \frac{\mu_e \lambda_{mean,1}(1/R_1 - 1/R_2)}{\cos \alpha + \mu_e (\sin \alpha + \lambda_{mean,1} / R_2)} \quad \ldots \ldots \ldots (4)$$

where $\lambda_{mean}$ is the characteristic length between the pitch point and the contact point determined the next formulae.

$$\lambda_{mean,1} = (R_1 \sin \alpha - R_2 \sin \alpha) / 2 \quad \ldots \ldots \ldots (4)$$

and $\mu_e$ is the coefficient of friction at the point $\lambda_{mean}$ distant from the pitch point.

The efficiency between the external gear and the internal gear is finally determined by the next formula.

$$\eta_{1st} = \frac{\varepsilon_A \eta_A + \varepsilon_R \eta_R}{\varepsilon_A + \varepsilon_R} \quad \ldots \ldots \ldots \ldots (5)$$

where $\varepsilon_A$ and $\varepsilon_R$ are the meshing ratio in approach and recess contact conditions, respectively.

$$\varepsilon_A = \frac{R_1 \sin \alpha - \sqrt{R_1^2 - (R_1 \sin \alpha)^2}}{t \cos \alpha} \quad \ldots \ldots \ldots (6)$$

$$\varepsilon_R = \frac{R_1 \sin \alpha - \sqrt{R_1^2 - (R_1 \sin \alpha)^2} - R_2 \sin \alpha}{t \cos \alpha} \quad \ldots \ldots \ldots (6)$$

where $R_1$ and $R_2$ are the pitch-circle radiuses, $R_1$ and $R_2$ are the tip-circle radiuses, $\alpha$ is the meshing pressure angle, and $t$ is the circular pitch, $t = \pi m$.

In the same manners, the efficiency in power transmission in the 2nd stage, $\eta_{2nd}$.

5.2 Theoretical efficiency of EPGD

EPGD in the authors’ development study runs by conversing the power input to the arm C into the power output from the internal gear B in the fixed internal gear A as shown in Fig. 4. In the estimation of the theoretical efficiency of this kind of multi-stage mechanism, a popular manner was employed by dividing the whole behavior of transmission into two stages as below, manner which is often used for a complicated gear mechanism.

As the first step, power input to the internal gear B and power output from the internal gear A is considered under condition that the carrier C is fixed. In this condition, the gear B rotating counter-clock-wise drives the gear A also counter-clock-wise in the same direction to the gear B: the moment of force $T_B$ loaded on the gear B is negative and then the positive moment of force $T_A$ is required to maintain an equilibrium state of steady motion. The efficiency in power transmission $\eta_B$, the reference efficiency, is defined and $T_B$ is derived as

$$\eta_B = (T_B \omega_A)/(-T_B \omega_B) = (T_B n_A)/(-T_B n_B) \quad \ldots \ldots \ldots (7)$$

where $\omega_0 = n_A/n_B = (Z_1 Z_4)/(Z_2 Z_3)$.

In order to keep the EPGD in a stationary state under the above pair of moment of force, $T_A$ and $T_B$, the
clock-wise moment of force \( T_c \) in next formula is required to be added on the carrier C.

\[
T_c = -(T_a + T_b) = \left( \frac{\mu_e}{\eta_0} - 1 \right) T_a
\]

By adding further clock-wise rotation of the entire system back to the initial angular position of the external gear A, the overall operation is equivalent to the one where the gear A is fixed, that is the normal behavior of the drive. Keeping the above equilibrium of moment of force throughout the sequential operations, the carrier C makes clock-wise full turn with \( T_c \) as an input and the internal gear B rotates by \( 1 - 1/\eta_0 \) with \( T_b \) as an output in counter-clock-wise direction. The theoretical efficiency in power transmission of this gear drive system is consequently determined by the next formula as the ratio of the output \( T_b \) \((1-1/\eta_0)\) to the input \( T_c \times 1 \) [6],

\[
\eta_{CB} = \frac{u_0 - 1}{u_0 - \eta_0 - \eta_a - \eta_{2nd}} \quad \text{(8)}
\]

where the reference efficiency \( \eta_0 \) in the above equation is the combination of \( \eta_{1st} \) and \( \eta_{2nd} \), the efficiencies of single gearing between an external and an internal gears in the first and the second stages, respectively, estimated in the previous chapter, as

\[
\eta_0 = \eta_{1st} \cdot \eta_{2nd} \quad \text{(9)}
\]

The theoretical efficiency in power transmission of this gear drive system EPGD is finally expressed by the next formula.

\[
\eta_{CB} = \frac{u_0 - 1}{u_0 - \eta_{1st} \cdot \eta_{2nd}} \quad \text{(10)}
\]

5.3 Estimated theoretical efficiency and discussion

By use of the expression of theoretical efficiency derived in the previous section, the one of EPGD developed in this study was evaluated with the specifications in Table 1. In the calculation, the magnitude of friction coefficient was presumed as \( \mu_e = 0.1 \) considering that the gears of the prototype were shaped by pinion cutter without grinding and an insufficient fluid lubrication between the gear faces in a quasi-static condition in the experiment.

As listed in Table 2, the basic efficiencies in the 1st and the 2nd stages are \( \eta_{1st} = 0.9991 \) and \( \eta_{2nd} = 0.9982 \), respectively, giving the reference efficiency \( \eta_0 = 0.9973 \) and finally the theoretical efficiency \( \eta_{CB} = 0.9493 \). Considering that additional loss caused by windage, friction in bearing and stirring of oil should be considered for a more precise estimation, the expression of the theoretical efficiency of EPGD and the estimated results are considered to be valid.

The estimated theoretical efficiencies under various values of friction coefficient \( \mu_e \) are summarized in Table 2. The results clearly show that the efficiency of gear device having a differential motion is in general strongly depends on the reference efficiency in each stage. Since the lubricant is one of the governing factors affecting the coefficient of friction, the importance of an appropriate selection of lubricating oil is suggested.

6 Conclusions

A new eccentric planetary gear drive has been developed with fewer components and less volume than a hydraulic motor and its performance was measured in a quasi-statistical condition.

1) The gear drive gives the highest efficiency \( \eta_{max} = 0.92 \) with the hypoid gear oil that has the highest viscosity among the tested oils with various additives. A satisfactory behavior in dynamic operating conditions, in addition, shows a possibility for further development.

2) The expression of the theoretical efficiency of EPGD is derived by dividing its operation into two stages.

3) The theoretical efficiency of the prototype developed in this study was estimated with presuming the magnitude of coefficient of friction \( \mu_e = 0.1 \). The estimation \( \eta_{CB} = 0.949 \) was considered to be valid compared to the experimental results with additional loss by friction and stirring resistance.

References


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### Table 2 Theoretical efficiency of EPGD

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<th>Friction coefficient</th>
<th>Theoretical efficiency</th>
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<tr>
<td>( \mu_e )</td>
<td>( \eta_{1st} )</td>
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<tr>
<td>0.05</td>
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