

Consideration about Meshing of Worm Gear Based on MUB (Meshing Under Base-Circle) Theory for EPS

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ABSTRACT

This paper will discuss the stress reduction of the worm wheel for an electric power steering (EPS) system. The research discussed in this paper focused on the worm wheel, the EPS component that determines the maximum diameter of the system. If the stress of the worm wheel could be reduced without increasing in size, it would be possible to reduce the size of the worm wheel and EPS system. In order to reduce the stress of the worm wheel, the conventional design method has extended the line-of-action toward outside of the worm wheel to increase the contact ratio of the gears and these method lead to an increase in the outer diameter.

In order to address this issue, past research proposes the basic concept to extend line-of-action toward the inside of the worm wheel. And this new meshing theory was named MUB (Meshing Under Base-circle) theory.

In this paper, characteristics of meshing of the gear formed by MUB theory are determined in more detail. After that, an example of design guideline of the worm gear profile based on MUB theory is explained. And this worm gear was manufactured, installed into an EPS system, and tested. This paper will report on the findings made during the course of this research.

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INTRODUCTION

Technological initiatives for the protection of the environment are proceeding throughout the world. EPS(1) is in widespread use in automobiles as a technology to increase fuel efficiency, and is contributing to the amelioration of environmental issues. However, with the rapid rise of emerging nations, it is predicted that the number of automobiles in use will increase to a greater extent than previously expected, and the increase in CO₂ emissions will therefore be greater than it has been up to the present. Against this background, it is incumbent on us to consider technological development in order to further ameliorate environmental issues. In the case of EPS, this means realizing further size and weight reductions and increased efficiency.

The research discussed in this paper focused on the worm wheel, the EPS component that determines the maximum diameter of the system. If the stress of the worm wheel could be reduced without increasing in size, it would be possible to reduce the size of the worm wheel and EPS system.

Based on this goal, the research team conducted research towards creating small, highly-efficiency, high-strength and high-durability worm gears. The research outcomes were

reported(2)-(7) and put into practical use. Reference (2) proposes a new profile for the single threaded worms employed in pinion assist EPS and reports that it results in a reduction in the external diameter of the worm wheel. Reference (3) reports that tests demonstrated the contact ratio of worm gears using the new profile increased in accordance with the target. Reference (4) reports on the reduction of contact pressure by the proposed new profile, while reference (5) focuses on the impact loads at the limit of the steering wheel turning range characteristic of EPS systems, and proposes a modification of the width of the worm wheel gear teeth in order to reduce this load. Reference (6) proposes a profile to reconcile impact resistance with high gear efficiency, and reference (7) reports on the reduction in torque fluctuations produced by this modified profile.

This past research has focused on the optimal worm gear profile for leveraging elastic deformation of a plastic worm wheel; there is still a margin for the enhancement of the worm wheel profile.

Conventional design methods have sought to reduce the stress of the worm wheel by increasing the helical angle and the number of teeth in order to increase the contact ratio.

Increased gear modules and profile shifted gears have also been attempted. However, using these methods, there has no option but to extend the line of action toward outside of the worm wheel and to increase the outer diameter.

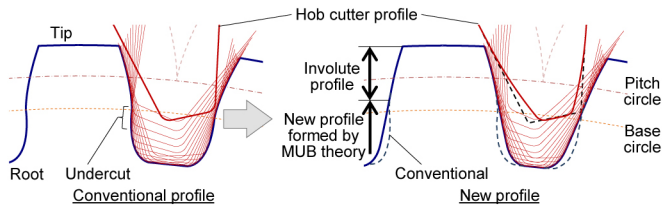


Figure 1. New profile of worm wheel

In order to address this issue, reference (9) proposes the basic concept to increase the contact ratio of the gears by extending line-of-action toward the inside of the worm wheel. To realize this concept, the geometric gear shape formed under the base circle in the machining process was considered, and the theory of meshing under the base circle of gear, which would realize effective meshing under the base circle, was proposed on the basis of these considerations. In this paper, constant speed transmission of the gear pair based on this theory was shown and line-of-action of the gears was derived geometrically. Finally, an actual worm wheel (Fig. 1) was produced based on this meshing theory, demonstrating the theory's validity. This paper will report on the findings made during the course of this research.

1. MESHING UNDER BASE-CIRCLE (MUB) THEORY

Conventionally, the worm wheel contact ratio has been increased by increasing the size of the worm wheel and extending the line of action in the direction of the tips of the teeth. The research discussed here took the opposite approach, considering that it would be possible to increase the contact ratio without increasing the size if the line of action could be extended in the direction of the dedendum. In order to realize this, research was conducted on a new dedendum profile producing effective meshing under the base circle.

Realizing meshing under the base circle using an involute profile would necessitate undercutting the worm wheel using a hob cutter, and the clearance between worm tooth tip and bottom of the worm wheel should be zero. With such gears, the tip of the worm tooth contacts with the bottom of worm wheel as shown in Fig. 2 and effective meshing is not obtained.

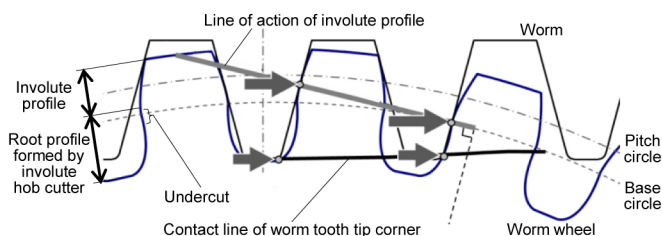


Figure 2. Contact line of involute worm wheel

1.1 Analysis of Conventional Dedendum Profile that does not Allow Meshing under Base Circle

To determine the ideal dedendum profile to realize effective meshing under the base circle, the dedendum profile created by a conventional hob cutter was formulated and analyzed the reason why it could not mesh effectively. When worm wheel teeth are fixed in the absolute coordinate system, datum line of hob cutter profile rolls over pitch circle of worm wheel without slipping. Then the center of hob cutter tooth describes epitrochoid curve and envelope, which is generated by hob cutter tooth moving along the epitrochoid, generates worm wheel profile. As shown in Fig. 3, the dedendum profile under the base circle is created by the tips of the hob cutter teeth.

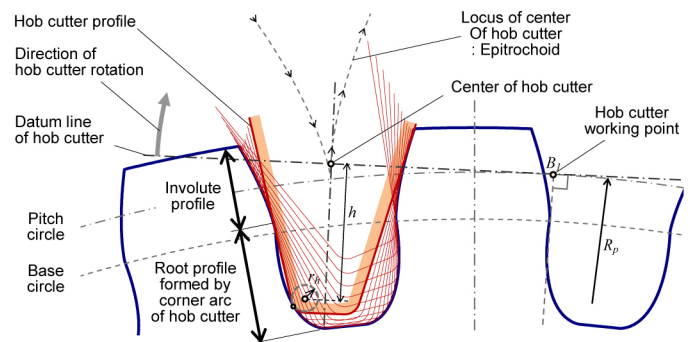


Figure 3. Locus of hob cutter

To formulate the dedendum profile, motion of the hob cutter tooth is modeled as shown in Fig. 4. Firstly, the locus described by the center of the arc of the hob tooth tip corner is calculated, and secondly, envelope curve created when a circle of radius r_h moved along this locus was calculated.

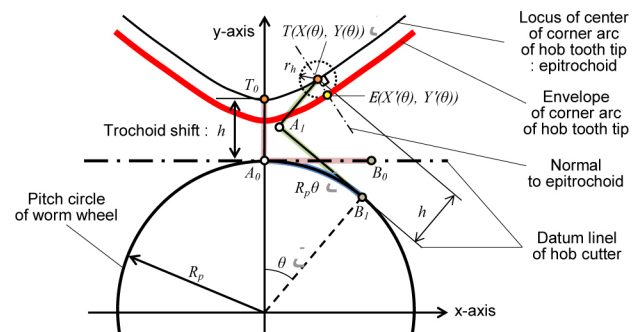


Figure 4. Envelope of hob tooth tip corner radius

When the datum line of the hob cutter profile makes contact with the phase point B_1 , at angle θ on the cutting pitch circle of the worm wheel teeth, the line segment A_1B_1 and arc A_0B_1 are of the same length because segment A_1B_1 is uncoiled arc A_0B_1 . Give this, the coordinates $(X(\theta), Y(\theta))$ of the center of the arc of the hob cutter tooth tip corner, T , can be expressed as

$$X(\theta) = (R_p + h)\sin\theta - R_p\theta\cos\theta$$

$$Y(\theta) = (R_p + h)\cos\theta + R_p\theta\sin\theta \quad (2)$$

with θ as a variable. Next, the envelope created by the arc of the hob tooth tip corner was determined. Because point E on the envelope curve is on the normal line of trochoid curve that crosses point T , and the distance TE corresponds to the radius of hob tooth tip corner, r_h ,

$$\frac{d(X(\theta))}{d\theta} = h\cos\theta + R_p\theta\sin\theta \quad (3)$$

$$\frac{d(Y(\theta))}{d\theta} = -h\sin\theta + R_p\theta\cos\theta \quad (4)$$

$$\sqrt{\left(\frac{d(X(\theta))}{d\theta}\right)^2 + \left(\frac{d(Y(\theta))}{d\theta}\right)^2} = \sqrt{h^2 + R_p^2\theta^2} \quad (5)$$

Point $E(X'(\theta), Y'(\theta))$ on the envelope can be expressed as

$$\begin{aligned} X'(\theta) &= X(\theta) + \frac{r_h}{\sqrt{h^2 + R_p^2\theta^2}} \left(\frac{d(Y(\theta))}{d\theta} \right) \\ &= (R_p + h)\sin\theta - R_p\theta\cos\theta \\ &\quad - \frac{r_h}{\sqrt{h^2 + R_p^2\theta^2}} (h\sin\theta - R_p\theta\cos\theta) \end{aligned} \quad (6)$$

$$\begin{aligned} Y'(\theta) &= Y(\theta) + \frac{r_h}{\sqrt{h^2 + R_p^2\theta^2}} \left(-\frac{d(X(\theta))}{d\theta} \right) \\ &= (R_p + h)\cos\theta + R_p\theta\sin\theta \\ &\quad - \frac{r_h}{\sqrt{h^2 + R_p^2\theta^2}} (h\cos\theta + R_p\theta\sin\theta) \end{aligned} \quad (7)$$

Figure (5) shows trochoid curves produced using the equations above when the center of the arc of the hob cutter tooth tip is shifted. The trochoid curves change depending on the direction of shift, with the hob cutter datum line as the reference. (i) shows negative shifted trochoid describing a circle, (ii) shows zero shifted trochoid describing approximate V-shaped curve whose corner is on the pitch circle and (iii) shows positive shifted trochoid describing approximate V-shaped curve combined concave shape and a low-curvature convex shape. Next, the envelope curve created when a circle moves along these trochoid curves were considered.

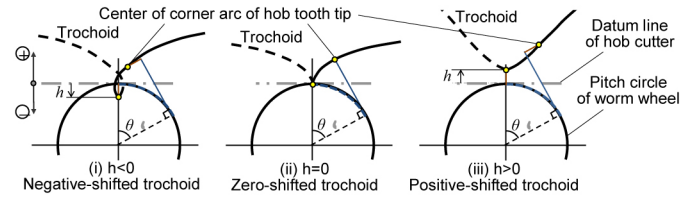


Figure 5. Shifted trochoid

Figure 6 shows the envelope formed by the negative-sifted trochoid and the line of action of gears using this envelope as the tooth profile. The line of action extends under the base circle, but because the pressure angle of the contact point is high, reaching close to 90 deg, the worm wheel is self-locking and unable to move. If the worm gear were provided with a clearance between worm tooth tip and worm wheel bottom to help prevent this, geometrically there would be no contact.

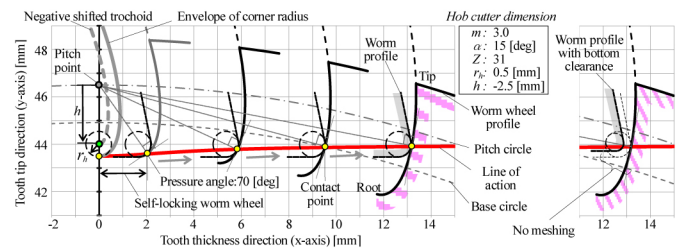


Figure 6. Meshing of gears formed by negative shifted trochoid

The envelope curve formed by zero-shifted trochoid and line of action of the worm wheel are shown in Fig. 7. On the inflection point which changes the slope of trochoid negative and positive, it is not possible to uniquely specify the normal line of the trochoid; the envelope cannot be expressed using Eq. (6) and (7) alone, and the envelope under the base circle is formed by the shape of the tooth tip arc on the inflection point. Because of this, the envelope under the pitch circle is arc-shaped, and the gears mesh intermittently. Therefore, constant speed transmission(8), which is the requirement for gears to mesh effectively, of the gears are not satisfied, because the contact ratio would be less than 1. To achieve this, multiple threaded worm is necessary to increase overlap contact ratio more than 1. This method is unfavorable, because the worm would get bigger.

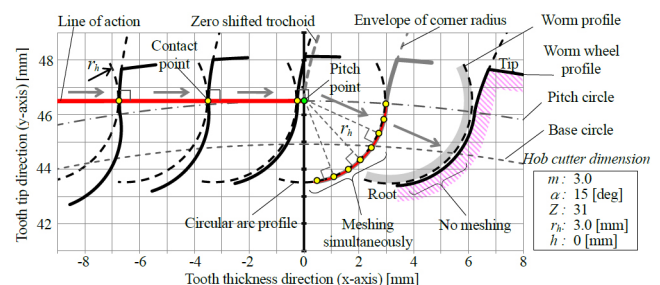


Figure 7. Meshing of gears formed by zero shifted trochoid