

Durability Design Method of New Stopper Bush Using New Theory (Friction and Spring) for Electric Power Steering

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ABSTRACT

In the automobile industries, weight reduction has been investigated to improve fuel efficiency together with reduction of CO₂ emission. In such circumstance, it becomes necessity to make an electric power steering (EPS) more compact and lightweight. In this study, we aimed to have a smaller and lighter EPS gear size by focusing on an impact load caused at steering end. In order to increase the shock absorption energy without increase of stopper bush size, we propose new theory of impact energy absorption by not only spring function but also friction, and a new stopper bush was designed on the basis of the theory. The profile of the new stopper bush is cylinder form with wedge-shaped grooves, and when the new stopper bush is compressed by the end of rack and the gear housing at steering end, it enables to expand the external diameter and produce friction. In this study, we considered the durability in the proposed profile. In particular, the theoretical study was made on the effect due to sliding from the standpoint of the contact pressure and the sliding distance, and durability testing was carried out to check for the durability of the performance after predetermined operation time. After the durability testing, we have examined the force-displacement characteristics, the change in characteristics by thickness reduction agree with the theory. We will report the design method, since the design procedure of the smaller and lighter EPS gear size could be confirmed useful.

CITATION: Shiraishi, T. and Shimizu, Y., "Durability Design Method of New Stopper Bush Using New Theory (Friction and Spring) for Electric Power Steering," *SAE Int. J. Passeng. Cars - Mech. Syst.* 7(3):2014, doi:10.4271/2014-01-0046.

1. INTRODUCTION

Recently a variety of industrial sectors are taking technological approaches to enhance the global environment. In the automobile industries, the fuel economy and the reduction of CO₂ emission are pursued. In the Power Steering Sector, the Power Steering is implemented to be motorized (EPS; Electric Power Steering) and to have higher strength and toughness for the realization of more compact and light weight EPS[1][2]. In such background we have focused on the impact load caused at the steering end and determined the object to create a smaller and lightweight EPS gear with reduction of the impact load. The study is made on the pinion assist type EPS as shown in Fig. 1(A). The EPS of this type is controlled so as to reduce the steering torque during the handling of steering wheel by applying the motor torque which is increased by the worm gear to the pinion shaft of rack and pinion [3][4]. And the steering end is set so that the steering operation does not cause excessive steer angle. As shown in Fig. 1(B), this steering end consists of the protrusion of rack end and the end face of gear housing to confine the stroke of rack. When a driver quickly turns the steering wheel and strikes on the

steering end (Fig. 1(C)), e.g. stationary steering when parking, the kinetic energy from the inertia of motor and the inertia of steering wheel is converted to an impact load and an elastic deformation, and increase the load on the teeth of rack-and-pinion and the worm teeth. Therefore the size of gear is determined to allow stress by the impact load on the steering end. For downsizing of the gear to make EPS more compact and lightweight, a subject is how to reduce the striking velocity resulting in reduction of the impact load by means of a shock-absorbing stopper bush provided at the steering end. Therefore in study[5], In order to obtain higher capacity for the stopper bush by increasing shock absorption energy without increase of the size, this study theoretically showed that the shock absorption capability can be improved in the confined stroke and load by newly adding the friction(Fig.2(B)) to the conventional spring function(Fig.2(A)). Also we have considered a practically applicable profile using this theory, and proposed the cylindrical form with wedge-shaped grooves on lateral side that when the stopper bush compressed in the axial direction it can extend in radial direction and deforms. (Fig.3(B)), Table 1 for parameter).

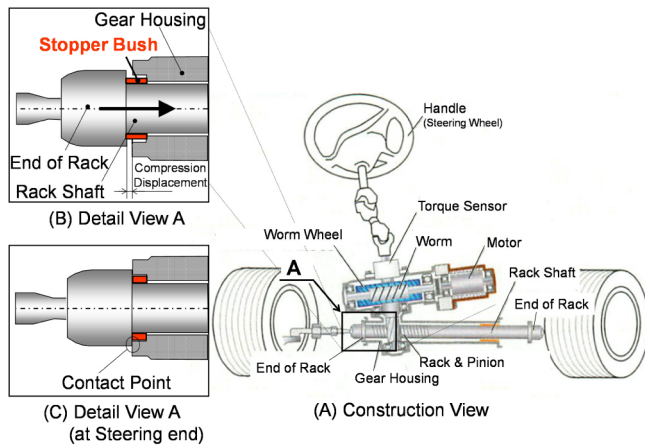


Figure 1. Gear Box of EPS

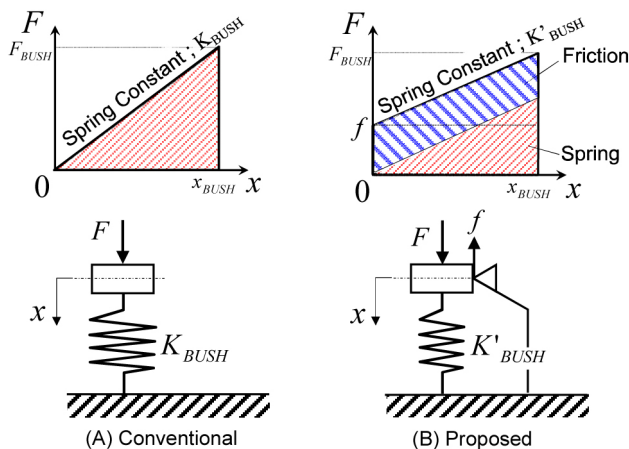


Figure 2. Characteristics of Rigidity with Friction

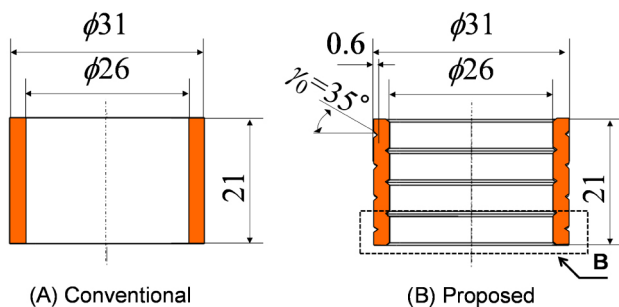


Figure 3. Section of Stopper Bush

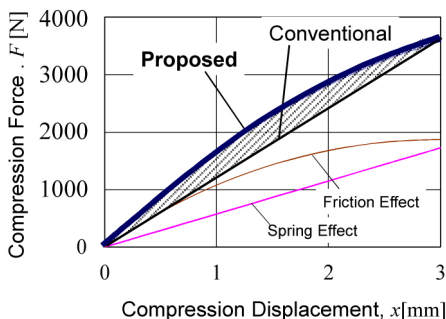


Figure 4. Proposed Force-Displacement Characteristics

Table 1. Specifications of Stopper Bush

Symbol	Parameter	Conventional Design Method	Proposed Design Method
d	Internal Diameter [mm]	26	26
t	Thickness [mm]	2.5	2.5
E	Modulus of Elasticity [N/mm ²]	110	110
b	Depth of Groove [mm]	-	0.6
c	Length of Bend Model [mm]	-	4.5
γ_0	Initial Angle of Groove [deg]	-	35
n	Number of Layers	-	4
μ	Coefficient of Kinetic Friction	-	0.1

The characteristics are shown in Fig.4, and the shock absorption energy is increased by 20% comparing to the conventional one (hatched area in the Fig.4). We have made the proposed stopper bush using this design method, and carried out the demonstration testing. The results show reductions of the impact load by 40%, the diameter of worm wheel by 10% and the weight of EPS by about 1[kg]. In this study, we considered the durability in the proposed profile. In particular, the theoretical study was made on the effect due to sliding from the standpoint of the contact pressure and the sliding distance, and durability testing was carried out to check for the durability of the performance after predetermined operation time. We will report about the durability design method and the usability which could be validated.

2. THEORETICAL EXAMINATION ON CONTACT PRESSURE AND THE SLIDING DISTANCE OF STOPPER BUSH

This chapter shows that the theoretical examination is made on the behavior of the proposed-shape stopper bush and a derivation of the contact pressure and the sliding distance to consider the durability.

2.1 Derivation of Normal Force at Sliding Surface of Stopper Bush

As shown in Fig.3(B), the stopper bush proposed in this study is constructed by multiple layers of the cylindrical profile provided with grooves of wedge angle γ_0 on the inner and outer circumferences. First, the consideration is made on the behavior of a single layer of them (Detail Area B In Fig.3(B), Fig.5(A)). The grooves of stopper bush deform as a compression force applied in the axial direction, and the stopper bush which swelled in a radial direction makes contact to the outer housing and the inner rack, which gradually generates friction force f_r . Then determination is made on derivation of the radial direction force F_r that is the normal force on sliding surface. The examination is made on the deformation behavior of proposed profile. For the deformation from initial profile (A) to the compressed profile (B) shown in Fig.5, the displacement Δx in the axial direction x and the displacement Δr in the radial direction r are geometrically expressed with Eqs. (1) and (2) as shown in Fig.5(C) taking the wedge angle γ . Here γ_0 is a free wedge angle before deformation occurs.

$$\Delta x = 2 \left(\frac{b \sin \gamma_0}{\cos \gamma_0} - \frac{b \sin \gamma}{\cos \gamma} \right) = 2b \left(\frac{\sin \gamma_0 - \sin \gamma}{\cos \gamma_0} \right) \quad (1)$$

$$\Delta r = \frac{b \cos \gamma}{\cos \gamma_0} - b = b \left(\frac{\cos \gamma - \cos \gamma_0}{\cos \gamma_0} \right) \quad (2)$$

The relationship between the wedge angle γ and the axial direction force F_x are expressed as Eq. (3) when considering the bending deformation model as simply supported beam (Fig. 5(A)) which is cut open. I_z is the moment of inertia of area of the bending model.

$$(\gamma_0 - \gamma) = \frac{Mc}{2EI_z} = \frac{(F_x b)c}{2EI_z} \quad (3)$$

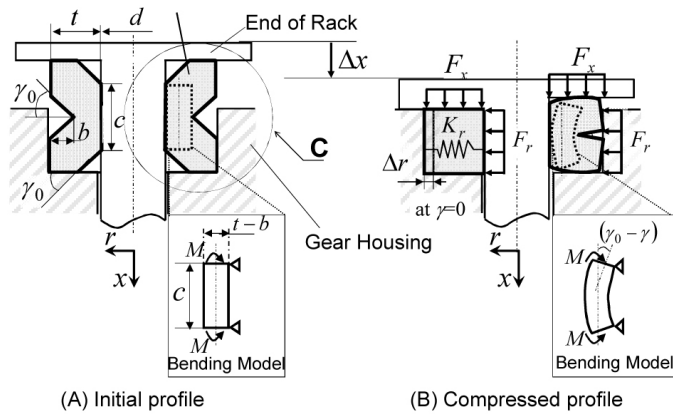


Figure 5. Profile of Proposed Stopper Bush

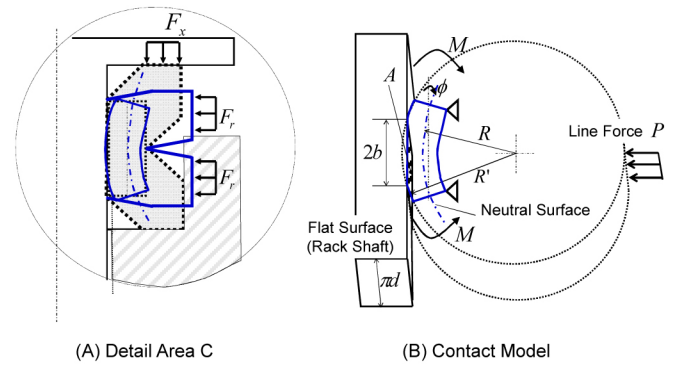


Figure 6. Contact Analysis

When the axial force F_x is applied, the displacement in the axial direction Δx is generated and the same time the displacement in the radial direction Δr is also generated. In this time, the force in the radial direction F_r is formed because the side face of stopper bush makes contact with the gear housing and the rack shaft. Eqs. (4) and (5) show the spring constant in the radial direction K_r of the stopper bush and the force in the radial direction F_r .

$$K_r = \frac{c \times \pi (d+t) \times E}{t + \Delta r} \quad (4)$$

$$F_r = K_r \times \Delta r \quad (5)$$

This force in the radial direction F_r is the normal force on the sliding surface. The friction force f_r , which is generated by the normal force, is $f_r = \mu \times F_r$ with the friction coefficient μ between the stopper bush and the gear housing and the rack shaft. This friction force will increase the shock absorbing energy of stopper bush. The consideration is made on the behavior of a multiple layers (n -layer). The displacement in the axial direction is increased n times. The reaction force of bending deformation is not changed cause of series connection. The friction force is increased n times due to parallel connection, and expressed with Eq. (6)

$$f_r(n) = n \cdot f_r \quad (6)$$