1. Introduction

A clutch mechanism is used to control gears in automobile automatic transmissions (Fig. 1 (a)). The clutch piston is an important mechanical component required in order to shift speeds by fastening the clutch plate hydraulically.

For clutch pistons, a cut aluminum part fitted with D-rings has generally been used (Fig. 1 (b)), but clutch piston seals, that is, a metal component with rubber lip formed by vulcanized adhesion process, have been gaining much notice in recent years because they offer more compact size, high response, and simplified assembly (Fig. 1 (c)).

Performance features primarily demanded of the clutch piston seal include superior pressure resistance (rigidity), sealing performance, and friction characteristics, and these three points are important for design.

Three-dimensional finite element analysis was used to simulate deformation of the clutch piston seal to establish design criteria.

The analytical results showed pressure resistance of the seal was affected by the position and radius bend and the thickness of the metal component. In particular, it was found that there exists a most suitable dimension regarding the position of the bend. The effectiveness of the results was confirmed by experimentation.

2. Analysis Method

2.1 Simulation by the Finite Element Method

The finite element analysis software used for simulation this time was ABAQUS/Standard by Hibbitt, Karlsson & Sorensen, Inc.

Using a three-dimensional model for analysis this time, we used solid elements for the rubber parts and shell elements for the metal component because reduction of nodal points was possible in order to achieve high precision and reduction of resources.

As for physical characteristics, the metal component was treated as an elastic body, and the rubber part as a hyperelastic body (three-dimensional polynomial).

Shell elements used this time were based on a formula based on the primary shear deformation theory of plate
bending (Mindlin/Reissner). This theory takes the effect of shear deformation into consideration and does not require an assumption of normal direction of shell elements, and it has the advantage of theoretically being able to allow even thick plates.

2.2 Analysis Conditions
Plate thickness, spring support width (hereafter S-S width) and bend radius as parameters for the analysis are given in Table 1 and Fig. 2, and seven types were analyzed.

<table>
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<th>No.</th>
<th>Thickness</th>
<th>S-S width</th>
<th>Bend radius</th>
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<tr>
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<td>3.0</td>
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</table>

*Each value is ratio to the dimension of No. 1

The analyzed finite element model is shown in Fig. 3 and boundary conditions in Figs. 4 ~ 6. As shown in the figures, the four points below were taken into consideration. As for material properties, the metal component was treated as a linear elastic material, and the rubber as a hyperelastic material.

1. Constrained axial displacement of clutch plate contact section
2. Hydraulic pressure acting on the seal surface
3. Spring load focused on the position of the spring center
4. Contact between seal lip and shaft or housing

Fig. 3 Finite element model
(Circumferential cross section: Circumferential displacement, radial/axial moment constraint; Cushioning section: Axial displacement constraint)

Fig. 4 Position of constraint

Fig. 2 Shape of model for analysis
2.3 Experiment Verification
Analysis configuration samples were prepared and displacement by hydraulic pressure load measured. The structure of the equipment used for measuring displacement according to Table 1 is shown in Fig. 7.

Axial displacement of the end of the inner diameter of the metal component of the test sample fixed at the clutch plate contact section was measured when manually controlled hydraulic pressure was applied.

Measurement was carried out twice at each point of four equiangular measurement points, and the average was taken as the actual measurement value.

3. Simulation Results
3.1 Effect of Plate Thickness
The relationship between hydraulic pressure and displacement in piston seals of various plate thicknesses is shown in Fig. 8, and the relationship between hydraulic pressure and maximum stress is shown in Fig. 9.

Fig. 8 shows that the calculation values of the analyzed model more or less agree with the experimental values and that the simulation precision was high enough. It was confirmed that displacement decreases as thickness increases.

It can be seen that displacement when various hydraulic pressures are applied tends to be more or less linear with hydraulic pressure, and is nearly proportional to thickness. The reason displacement without a load is negative (direction in which the seal is pushed up to the hydraulic pressure side) is that contact reaction force is produced by interference of the inner seal lip.

As for maximum stress, however, stress is produced by contact of the seal lip even when there is no load.

After that, along with an increase in hydraulic pressure, maximum stress rises more or less linearly after temporarily dropping somewhat.

As for the effect of plate thickness, maximum stress decreases proportionally as plate thickness increases.

The reason maximum stress temporarily decreases when
hydraulic pressure is applied is presumed to be as shown below.

First of all, stress is produced by negative displacement resulting from interference of the inner seal lip due to seal insertion when there is no load.

Maximum stress temporarily drops because stress produced by hydraulic pressure and stress due to interference cancel each other when hydraulic pressure is applied.

The maximum stress of each part is in a linear relationship to the distance of the S-S width, but shows the opposite tendency. This is probably due to the fact that stress produced by hydraulic pressure is concentrated in two S-S widths and the fact that the degree of concentration varies according to S-S width. The ideal way to make maximum stress produced in the seal extremely small is to equalize maximum stress occurring in the two S-S widths.

We therefore sought the distance of the S-S width where maximum stress produced in the two S-S widths shown in Fig. 12 becomes equal, and as a result of displacement analysis of the model, estimated maximum stress (479 MPa) and approximate stress value (484 MPa) were obtained. This is equivalent to a 9% reduction of the maximum stress of the initial design dimensions.

In other words, because maximum stress produced for the entire seal can be reduced by equalizing maximum stress at the various positions of stress concentration, there is an optimal S-S width.

3. 2 Effect of Spring Support Width (S-S Width)

The relationship between displacement and hydraulic pressure concerning piston seals with different S-S widths is shown in Fig. 10.

The figure shows that predicted and experimental values agree fairly well, and that simulation has sufficient accuracy.

Displacement at various hydraulic pressures tends to be more or less linear with hydraulic pressure. It has been also confirmed that the smaller the distance of the S-S width, the smaller the displacement amount.

The relationship between hydraulic pressure and maximum stress is shown in Fig. 11. The figure shows that, just as with the effect on displacement, the effect of distance of the S-S width on maximum stress is such that the shorter the distance of the S-S width, the less the maximum stress is, and stress reduction rate decreases proportionally.

An example of analyzed stress distribution at this time is shown in Fig. 15. The position of maximum stress occurrence is always at part A or B in Fig. 15. The stress distribution produced at the distance of each S-S width separated and arranged at parts A and B is shown in Fig. 12.
3.3 Effect of Dimension of Bend Radius

The relationship between hydraulic pressure and displacement for piston seals of various dimensions of bend radius is shown in Fig. 13.

The figure shows that predicted values and experimental values agree fairly well, and that simulation is sufficiently precise.

The dimension of the bend radius was found to have almost no effect on displacement. Displacement for various hydraulic pressures tended to be in a nearly linear relationship with hydraulic pressure.

The relationship of hydraulic pressure and maximum stress is shown in Fig. 14. This figure shows the effect of the dimension of the bend radius on maximum stress, whereby maximum stress decreases proportionally as the dimension of the bend radius increases.

Effect of the dimension of the bend radius on displacement and maximum stress differs according to the distribution of stress, as shown in Fig. 16. In other words, when the dimension of the bend radius is large, a small amount of stress occurs over a large area. This is because a large amount of stress occurs in a small area when the dimension of the bend radius is small.

Displacement, on the other hand, is the integral value of strain. This is probably why because there is no difference in maximum displacement.
4. Conclusion

The following points became evident as a result of analyzing and verifying the effect of geometrical dimensions on pressure resistance of clutch piston seals by finite element analysis.

1) High-precision FEM displacement analysis of clutch piston seals with axial symmetrical shape is possible.
2) Pressure resistance of clutch piston seals is largely affected by S-S width and plate thickness.

In addition to what is reported herein, we are working to analyze clutch piston seals with complex wavy shapes in the circumferential direction to help in the study of future designs.

Finally, we would like to thank Koyo Sealing Techno Co., Ltd., for its cooperation.

Reference