Development of Rattle Noise Analysis Technology for Column Type Electric Power Steering Systems

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The backlash adjustment mechanism for reduction gears adopted in electric power steering systems aims to reduce rattle noise at gear inverse rotation by decreasing the backlash between worms and reduction gears. However, there is a possibility that another rattle noise could be generated due to the characteristics of the backlash adjustment mechanism itself.

Therefore, we have developed rattle noise simulation for Column type electric power steering (C-EPS) systems to analyze rattle noise caused by the backlash adjustment mechanism. The cause of rattle noise has been inferred from simulation and test results. This simulation can now be used at the design phase to design parameters which will reduce rattle noise and improve design efficiency.

Key Words*: rattle noise, electric power steering, backlash adjustment mechanism, reduction gear*

1. Introduction

The backlash adjustment mechanism for reduction gears used in electric power steering systems whose purpose is to reduce rattle noise during gear inverse rotation from worm gear backlash has a structure with an elastic supporting part for the worm that absorbs the changes of center-to-center distance between the worm and the worm wheel, which occurs depending on operating temperature or deterioration over time, and reduces gear backlash, resulting in reducing the rattle noise to a much greater extent compared with conventional structures.

However, as the worm in this structure does slight translational motion in addition to primary rotational movement, there is a possibility of generating another rattle noise if a sudden change of load occurs in the steering systems.

In order to achieve both reduction of rattle noise caused by the backlash adjustment mechanism of reduction gear and securing fundamental performance of electric power steering systems, there has been much trial and error. It has been a challenge for product development how efficiently and precisely these can be performed.

And so we have developed rattle noise analysis simulation for Column type electric power steering (C-EPS) systems with the purpose of predicting generation of rattle noise caused by the backlash adjustment mechanism for reduction gears and deriving optimum specifications for controlling the impact force by translational motion of the worm which is a cause of rattle noise.

2. Simulation Model

2. 1 C-EPS System Model

The C-EPS system model shown in **Fig. 1** consists of a C-EPS mechanical portion, an assist control portion and a motor current control portion, and is a model of the whole C-EPS which simulates the C-EPS behavior in

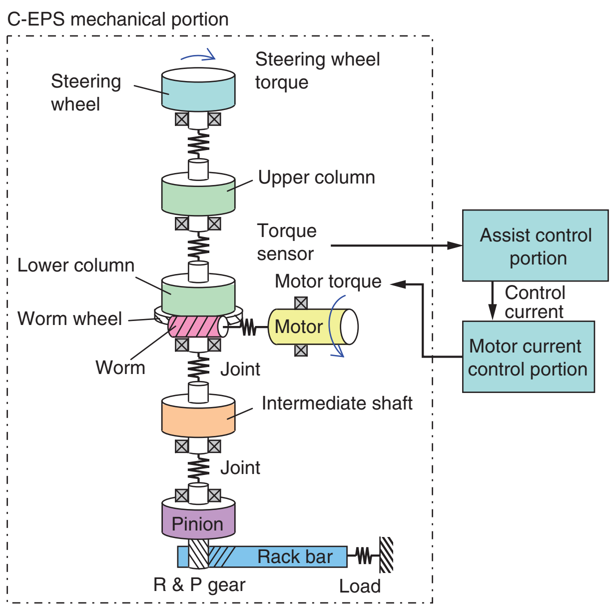

Fig. 1 C-EPS system model

accordance with the steering force from steering wheel and the input force from tires.

Regarding the C-EPS mechanical portion, on the assumption that the major mass and inertia elements constructing C-EPS are rigidly integrated, a equation of motion was defined for each element and the whole construction of C-EPS has been modeled based on the equation.

The assist control portion and the motor current control portion have been modeled based on the control logic used in JTEKT's C-EPS.

2. 2 Model of Backlash Adjustment Mechanism for Reduction Gears

Figure 2 shows the structure of backlash adjustment mechanism for reduction gears. Clearance is provided between the end side bearing of a worm and housing, and a special spring for backlash adjustment is mounted in the clearance, of which structure allows the worm to move toward the center-to-center direction with Q as a fulcrum. This spring presses the worm against the worm wheel side to control the backlash and at the same time acts as a buffer to prevent hitting noise between the bearing and the housing. Also, this structure allows the worm to move in the axial direction due to the narrow axial clearance of the motor side bearing.

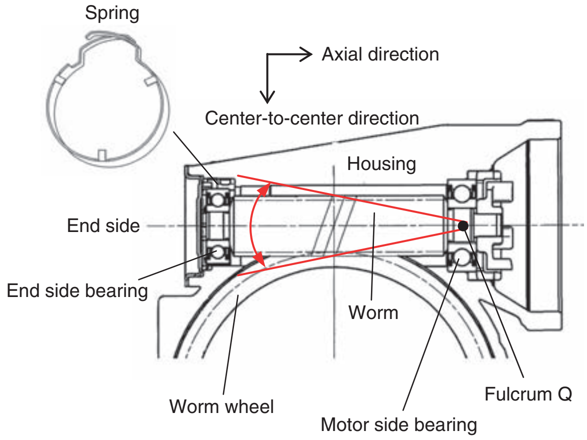

Fig. 2 Backlash adjustment mechanism for reduction gears

Figure 3 shows a schematic diagram of backlash adjustment mechanism model for reduction gears. This model is based on the structure and movement principle of the backlash adjustment mechanism for reduction gears, has a worm, gear meshing load, spring stiffness, contact stiffness of worm gear (stiffness in the center-to-center direction between the worm and worm wheel), specifications of motor side bearing of the worm (axial clearance, ball stiffness), and critical stiffness, which represents the motion limit for each direction, as structural elements, and has been made able to simulate the worm behavior in the center-to-center and axial directions with Q as the fulcrum. The spring stiffness has been made to functionally define the actually measured

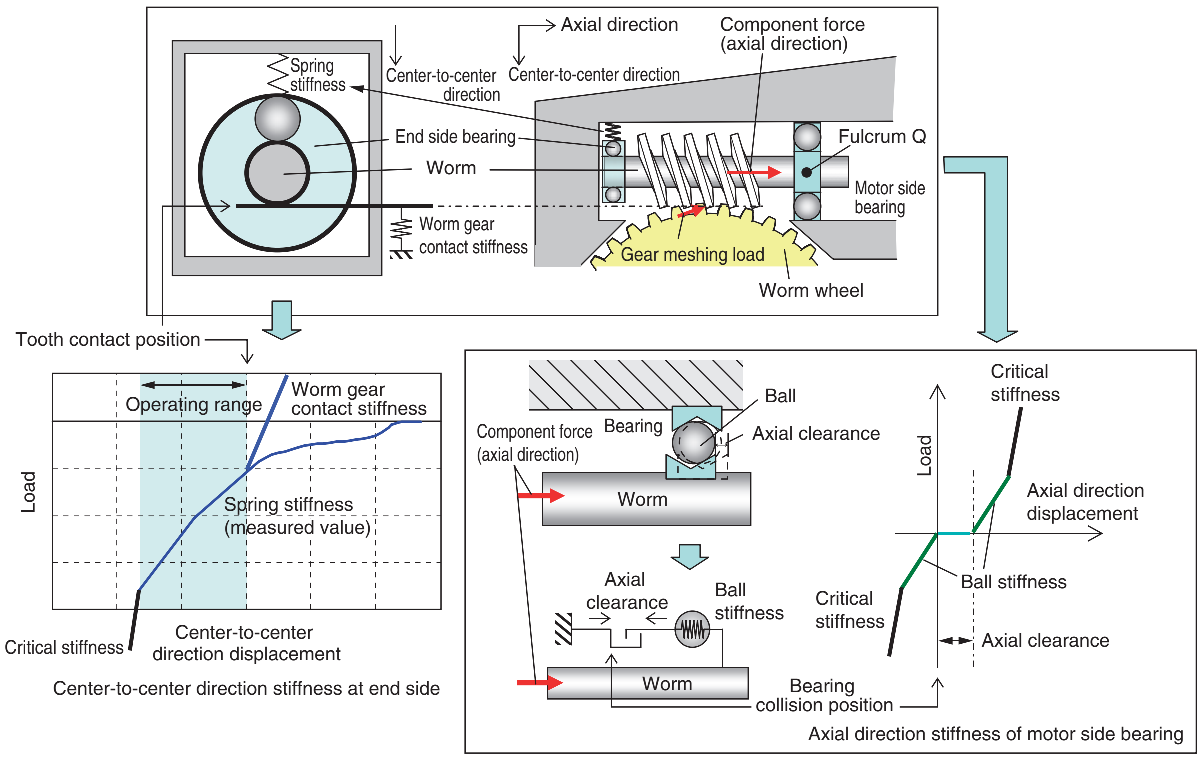

Fig. 3 Model of backlash adjustment mechanism for reduction gears

stiffness as the characteristic value (nonlinear stiffness) by using such factors as operating range in the center-to-center direction. The load (component force) which forces the worm to move toward each direction has been obtained from the gear meshing load, the sliding friction force on the tooth surface and the moment of the worm which depends on the gear meshing load direction and the fulcrum Q.

In order to verify the calculation results obtained from the model, they have been compared with the results of the test.

In the test with C-EPS systems testing equipment as shown in **Fig. 4**, the lower portion of C-EPS below the intermediate shaft was cut, a vibration machine was connected to the lower column, and the center-to-center displacement at the end side of the worm and the axial direction displacement were measured at the time of the reverse input with angle fluctuations of ±180 degree and vibration torque of 15 Hz.

The calculation model has allowed us to mount the model of the backlash adjustment mechanism for reduction gears shown (**Fig. 3**) on the C-EPS model (**Fig. 1**), and cut off the lower portion below the intermediate shaft and input vibration conditions equivalent to the test into the lower column. **Figure 5** shows the results of the center-to-center displacement on the end side of the worm and the axial displacement calculated in this model together with the test results. In **Fig. 5**, "End x disp" stands for the center-to-center displacement at the end side of the worm, "z disp" for the axial displacement, "Sim" for the calculation results and "Exp" for the test results respectively. The calculation results show similar behavior and tendency toward the test results, and thus the calculated model has been confirmed to be valid. (The calculation with the above model and vibration conditions is referred hereinafter as rattle noise simulation.)

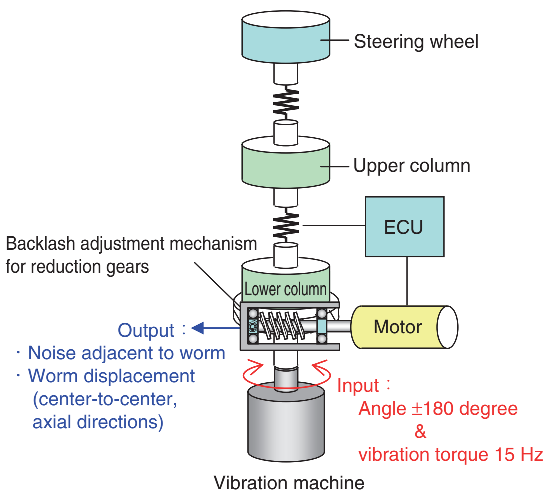

Fig. 4 Testing method for rattle noise

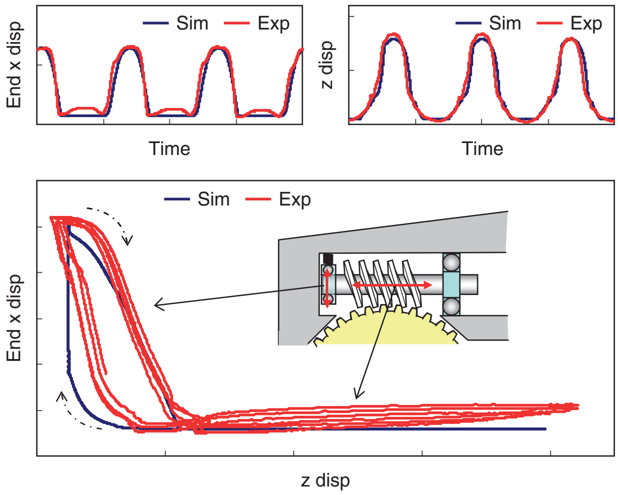

Fig. 5 Comparison of test and calculation results

3. Evaluation of Rattle Noise in Backlash Adjustment Mechanism for Reduction Gears with Test

In order to investigate the rattle noise with the test, a sample which was adjusted to generate the rattle noise (hereinafter called a sample for rattle noise evaluation) was prepared, and the noise near the end side of the worm was measured under the test conditions mentioned in the previous chapter. At the same time, it was examined at which moment the peak of the rattle noise appears against the center-to-center displacement at the end side of the worm and the axial displacement. The measurement result is shown in **Fig. 6**.

In **Fig. 6**, it has been confirmed that the worm is displaced in the center-to-center direction and the axial direction, the peak of the noise appears at the moment of collision, and the rattle noise is caused by the backlash adjustment mechanism for reduction gears.

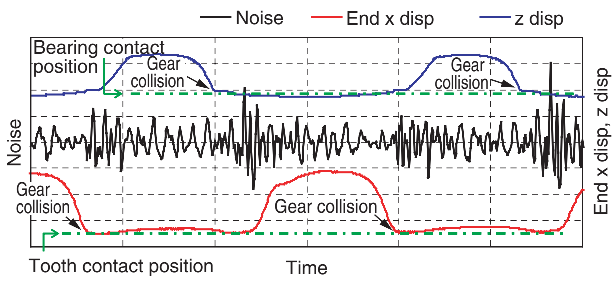

Fig. 6 Results of rattle noise test

4. Study on Influential Factors to Rattle Noise and Countermeasures

Figure 7 shows the results of the rattle noise simulation carried out under the same conditions as the rattle noise test in the previous chapter. As alternative characteristics for the rattle noise, the differentiation value of load which occurs depending on the displacement of worm and the stiffness in each direction (x_dF in the center-to-center direction and z_dF in the axial direction; hereinafter called differentiated forces) and the differentiated forces (G_dF) which occur in the vertical direction of the worm gear tooth surface have been used. In the simulation, the peak of the differentiated forces also appears at the same moment as the test results, and therefore, it is presumed that these sudden differentiated forces have caused the rattle noise.

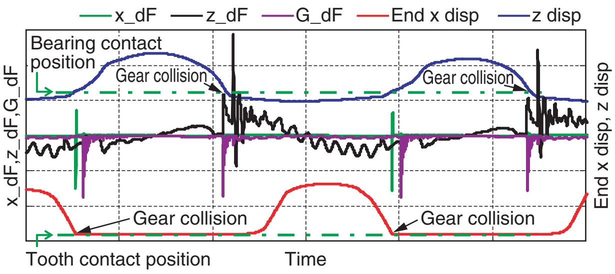

Fig. 7 Result of rattle noise simulation

Based on the above results, countermeasures have been studied by extracting influential factors on the rattle noise generated by the backlash adjustment mechanism for reduction gears.

It is presumed that factors for the peak of the differentiated forces generated by collision in the center-to-center direction of the worm are: the mass of the worm, the spring stiffness, the contact stiffness of the worm gear, and operating range in the center-to-center direction. It is also presumed that, in the axial direction of the worm, factors are: the mass of the worm, the axial clearance of the motor side bearing and the stiffness in the axial direction of the motor side bearing.

In order to reduce the rattle noise generated by the backlash adjustment mechanism for reduction gears, it is necessary to reduce the collision energy in the center-to-center and the axial directions of the worm. Therefore, to mitigate that collision energy, a parameter study regarding the above factors has been done through the rattle noise simulation and optimum specifications to reduce the rattle noise have been investigated.

5. Vibration Analysis by FEM Model and Investigation of Correlation

One issue in studying optimum specifications for rattle noise reduction exists in judging whether the differentiated forces or the vibrations generated by the differentiated forces are appropriate as an evaluation indicator for the rattle noise. Therefore, the correlation between the rattle noise with differentiated forces and the vibrations generated by the differentiated forces was investigated by FEM model.

5. 1 Prediction of Worm Housing Vibration

We have considered, as shown in **Fig. 8**, that the vibration generated by the differentiated forces can be predicted from the column vibration U at an arbitrary position obtained by taking the loads (transmitted from the worm) imposed on the bearings at both ends of the worm, which are calculated in the rattle noise simulation, as the exciting force and multiplying the transfer function expressed in FEM model by the exciting force. In the FEM model shown in **Fig. 9**, the worm housing has been taken as an evaluation point.

The FEM model was made with the C-EPS mounted to the jig and fastening of each part was done based on the spring element with the spring constant calculated from general tightening theory of screws. And, to express the transmission systems only, the modeling was done with the housing only, excluding the internal structures (worm gear, motor shaft, etc.). The transfer function expressed in the FEM model was constructed while confirming the consistency with the analysis on the test mode.

Figure 10 shows the FEM model results and the test results of the worm housing vibration. In **Fig. 10**, the FEM results have been able to duplicate the equal collision waves of the test results, which means that the validity of the column vibration predicted with FEM has been confirmed.

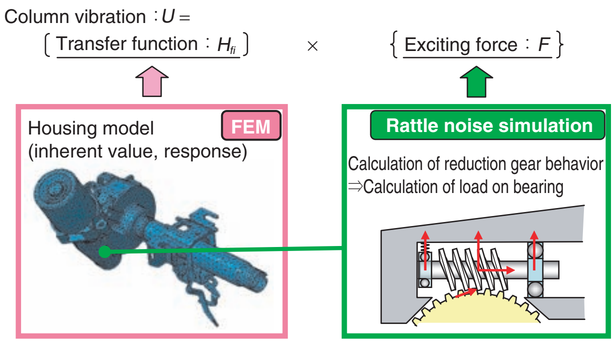

Fig. 8 Prediction of column vibration

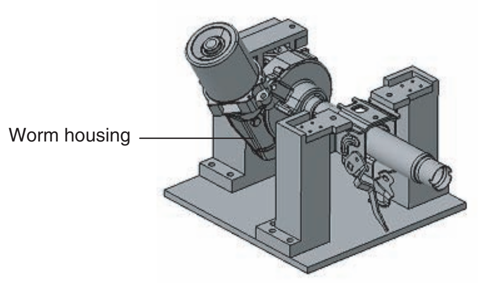

Fig. 9 FEM model

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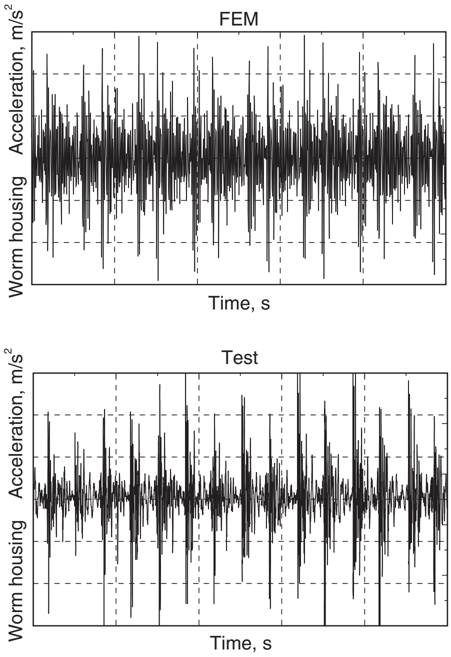

Fig. 10 Vibration results (FEM vs test results)

5. 2 Investigation of Correlation with Rattle Noise

Next, we have investigated the correlation of the worm housing vibration calculated in FEM model and the differentiated forces calculated in the rattle noise simulation with the rattle noise measured in the test. The number of the test and analysis levels where the vibration torque and the vibration frequency vary was eleven in all. Results are shown in **Figs. 11** and **12**. **Figure 11** shows the relation between the worm housing vibration and the rattle noise and **Fig. 12** shows the relation between the differentiated forces and the rattle noise, and in both the approximation lines and coefficients of correlation (R^2) are included respectively.

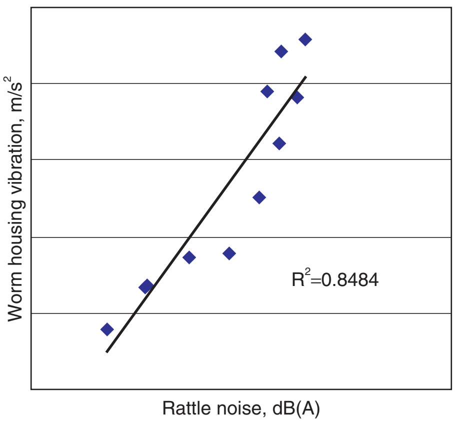

Fig. 11 Relation between worm housing vibration (FEM) and rattle noise

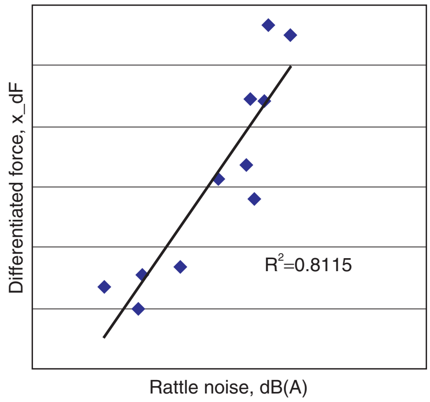

Fig. 12 Relation between differentiated force and rattle noise

As shown in **Figs. 11** and **12**, the correlation of the worm housing vibration with the rattle noise is slightly higher than that of the differentiated forces, however, since the correlation of the differentiated forces with the rattle noise indicates a high R^2 value (over 0.8), it can be judged that there is no problem designating the differentiated forces as an evaluation index for the rattle noise. Moreover, it is more efficient to designate the differentiated forces, which can be calculated solely by the rattle noise simulation, as an evaluation index than to designate the worm housing vibration which needs to be calculated in combination with the rattle noise simulation and FEM.

6. Derivation of Optimum Specifications and Standardization

Based on the results in the previous chapter, a parameter study on the rattle noise simulation was done by taking the differentiated forces (the maximum peak value) as an evaluation index, and the optimum specifications to reduce the rattle noise were derived. As to the factors mentioned in **Chapter 4**, with mitigating collision energy toward the center-to-center and axial directions of the worm in mind and taking into consideration the interaction of influential factors, influence on the fundamental performance of C-EPS, and manufacturing constraints, factors were narrowed down and a level was set up for each factor. By combining those levels, 12 patterns (Pt) were established and simulations were carried out. The results are shown in **Fig. 13**. **Figure 13** shows the maximum peak value of the differentiated forces with the 12 patterns (Pt1 to Pt12) that were studied. The maximum peak value gets to the minimum with Pt11 and this combination has been regarded as the optimum specification.

Next, it was confirmed to what extent the rattle noise could be improved by these optimum specifications. **Figure 14** shows the results of the rattle noise simulation with specifications studied at an early stage of designing (hereinafter called early stage study specifications) and

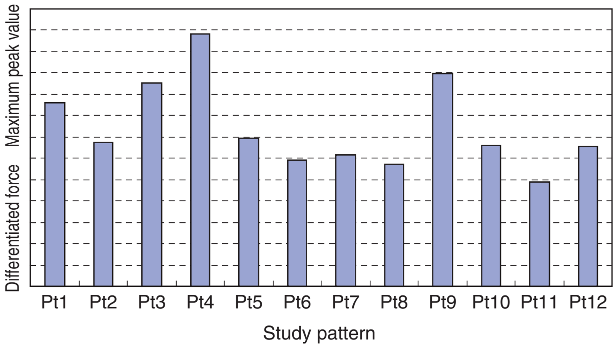

Fig. 13 Factors influencing rattle noise

the optimum specifications. **Figure 14** shows the time varying waves of the differentiated forces, the maximum peak value zone of which is enlarged, and includes the magnitude of the differentiated forces (evaluation criteria level) equivalent to the evaluation standard value of the rattle noise in the test. Also, for reference, the simulation results with the specifications of a sample for rattle noise evaluation conducted in the **Chapter 3** have been included.

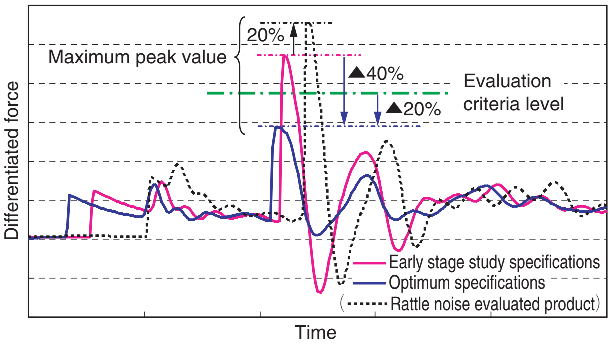

Fig. 14 Effects of optimization

As the maximum peak value under the early stage specifications study is higher than the level of evaluation standard, there is a possibility of rattle noise generation. As for the optimum specifications, the maximum peak value decreases by 40% against the early stage specifications study and by 20% against the evaluation criteria level, and therefore it can be judged that there is no problem with rattle noise. Furthermore, in the optimum specifications, it has been confirmed by the test that the rattle noise is quite small. Also, the optimum specifications have already been implemented in our mass production.

The outcomes obtained from the results of this study have been reflected in our design procedures as follows. These can be applied not only to the design of C-EPS but also to JTEKT's other types of electric power steering systems in which the backlash adjustment mechanism for reduction gears is mounted, and as a result, efficiency in our development and design of electric power steering systems has been improved.

(1) Standardization of the rattle noise simulation which has enabled at the desk study in the design stage
(2) Clarification of the rattle noise evaluation index and method of the rattle noise simulation
(3) Inclusion of guidelines to decrease the rattle noise in the design manual of the backlash adjustment mechanism for reduction gears

7. Conclusion

Starting with installation in light and small size vehicles, the demand for electric power steering systems has remarkably increased due to environmental concerns such as energy saving and not using oil. In addition, backed up by increased environmental awareness, it is currently believed that vehicles such as hybrid and electric cars with higher quietness have gained traction in the market and further quietness will be expected from electric power steering systems. Furthermore, in line with the rapid globalization of markets, product development that adapts in other countries to different infrastructures and different sense of values for vehicles will be required. In order to comply quickly with these market fluctuations and high level demands from customers, we will endeavor not only for further improvement of existing products and development of new products but also the enhancement of our supporting technology.

＊1 C-EPS is a registered trademark of JTEKT Corporation.

Reference

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