THE DEVELOPMENT OF THE SIMULATION METHOD THAT CAN PREDICT THE VIBRATION AS SOURCE OF INTERIOR NOISE FROM THE PARKING SYSTEM OF AUTOMATIC TRANSMISSION

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ABSTRACT – The Parking system of Automatic Transmission(AT) has several mechanical parts to keep stopping vehicle safely in the parking position. When select lever is moved from R to P, vibration is occurred by contact between two parts in the parking system. Then such vibration is transferred to driver as interior noise throughout AT case and select lever.
In this study, We have developed a simulation method by using multi-dynamic analysis and modal transient response analysis. At first we made dynamic model to calculate contact force which became the source of excitation. Then we made Finite Element(FE) model for calculating normal mode analysis and transient response analysis.
We performed experiment in order to validate simulation. We found out that simulation results have a good correlation with experiment.
This simulation method will help to shorten the development period and cost.

1. INTRODUCTION

Reducing vibration and noise in interior of vehicle has been constantly requested by user. In order to meet this request, numerous studies have been carried out using simulation which can predict vibration and noise even from stage of concept design. In the case of Automatic Transmission(AT) which is one of source of interior noise, there have been many studies through simulation such as normal mode analysis and frequency response analysis by assuming model as linear system.
But in case of simulation method about unusual noise caused by contact between two parts, it is difficult to develop simulation method due to nonlinear excitation and numerical problems of convergence of transient analysis.
In this study, we developed a simulation method to predict vibration caused by contact between wedge and tube in parking system in AT (area of green box in fig. 1)
Finally we determine the level that is acceptable or not through transfer sensitivity measured as seen in fig.1.
Experimental data is compared to the predictions to assess the fidelity of the proposed simulation method.

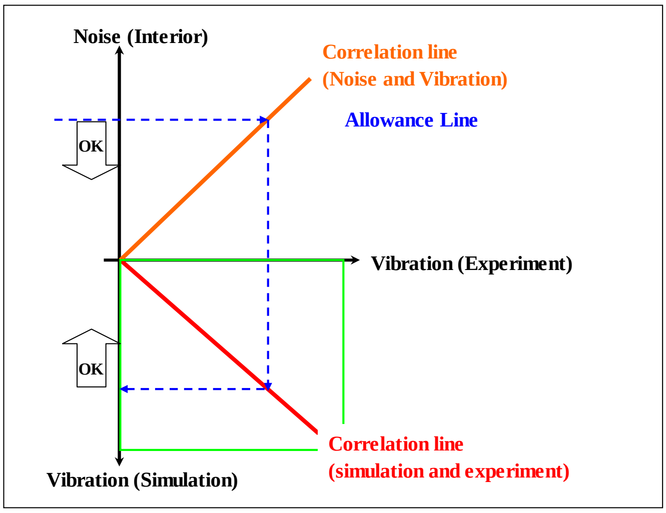

Figure. 1 Correlation line for prediction of interior noise

Fig. 2 shows parking system of AT which consists of several parts such as parking gear, parking pawl, rod, tube, wedge, rod spring, manual shaft and select lever. When select bar connected with select lever is moved from R(reverse) to P(parking) by driver, wedge is pushed by compressed rod spring. Then wedge contacts to tube before vibration is transferred through rod and manual shaft to select lever.

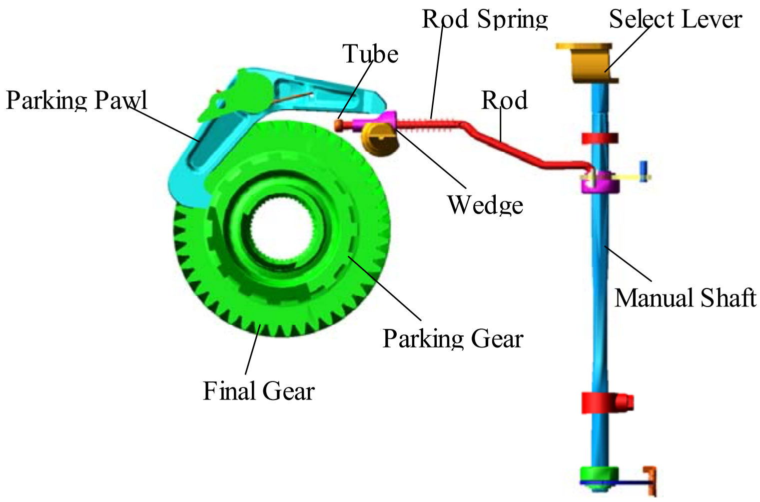

Fig. 2 Parts of parking system in AT

2. MODELING

2.1 DYNAMIC MODELING

For calculating excitation force during contact the dynamic model of a parking system mechanism is constructed. For this dynamic model we used the multi-body dynamic analysis program called ADAMS, which can numerically solve equation of motions as below.

$$M\ddot{q} + \Phi_q^T \lambda = Q$$

M : mass matrix

q : generalized coordinates

Φ : equation of constrant conditions

λ : lagrange multiplies

Q : external force

2.2 CONTACT MODELING

In the contact models, the simplest one is the Kelvin-Voigt model(1), which contains a parallel linear spring-damper element. However, Hunt and Crossely showed that the linear model does not represent the physical nature of energy transfer(2). Instead, they represented the contact forces by the Hertz force-displacement law(3). Therefore, a contact force model considered here is related to the model with parallel nonlinear spring and linear damper elements. The contact force is given by the following equation.

$$F = kx^{p} + c\dot{x} \tag{1}$$

In equation (1), x is the penetration depth, k depicts the spring constant and c means damping coefficient. The discrepancy of this model is that the force model has discontinuity at the moment of impact. Just before impact, the contact force is apparently zero. However, the damping force is applied instantaneously due to initial velocity. The contact force model to overcome above these shortcomings was proposed by Lankarani and Nikravesh(4). This contact force model can be expressed by the following equation (2).

$$F = k(1 + \mu\dot{x})x^{p} \tag{2}$$

where the parameter μis called the hysteric damping coefficient which can be expressed by following equation (3).

$$\mu = \frac{3k(1-e^{2})}{4\dot{x}^{(-)}} \tag{3}$$

where e is the restitution coefficient, $x^{(-)}$ is the velocity before impact.
Figure 3 shows Contact force between wedge and tube during contact using equation (2). In figure 3, recovery phase (duration from maximum contact force to separation of wedge and tube) is longer than compression phase (duration from beginning contact to maximum contact force). This results from effect of hysteresis caused by hysteric damping coefficient. Contact force become as input to transient response analysis, which will be explained next chapter.

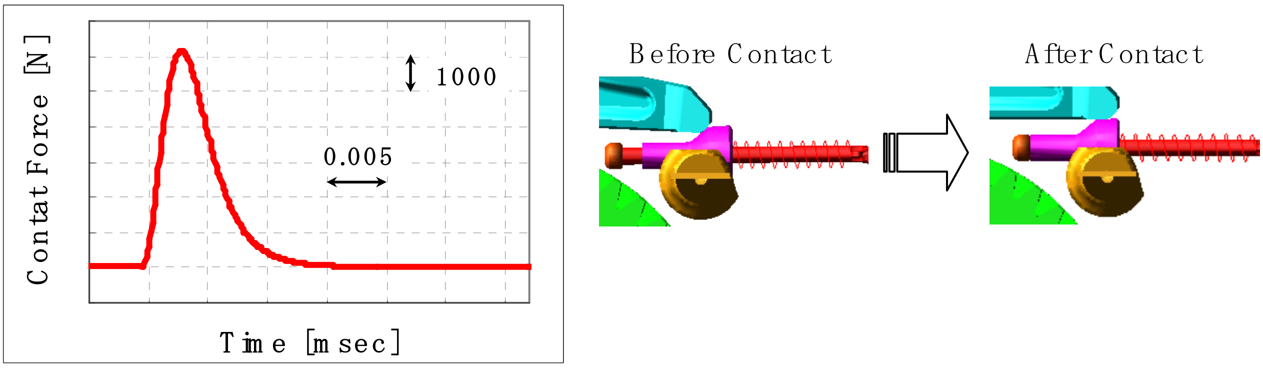

Fig. 3 Contact force between wedge and tube

2.3 FE MODELING

In order to simulate transient response analysis using NASTRAN, FE model is constructed. Fig. 4 shows FE model of parking system and case of AT. After contact force input to tube as excitation, vibration which is transferred through parking system and case can be calculated at the end of manual shaft.

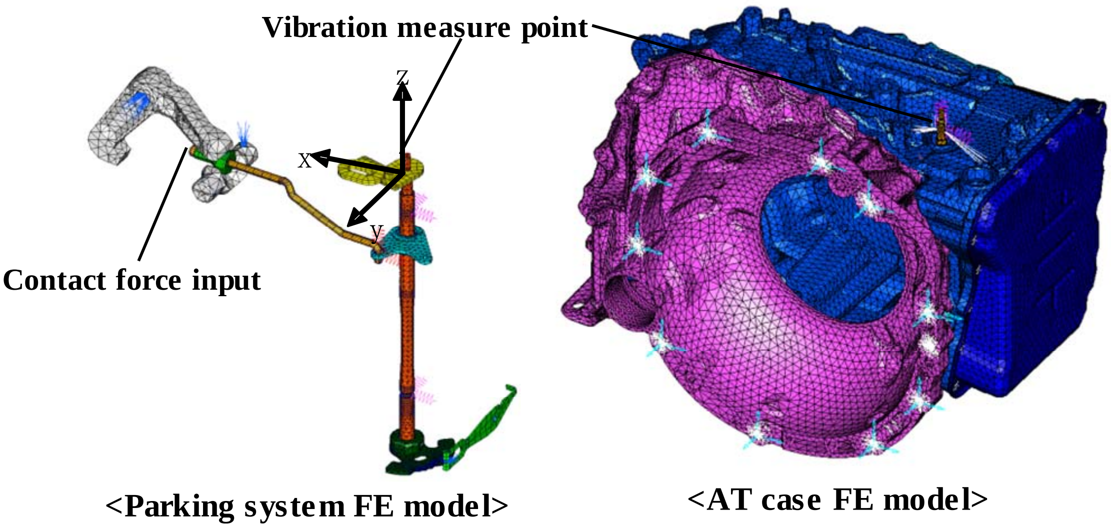

Figure. 4 FE model of parking system and case of AT

Fig. 5 shows constraints of FE model which affect mainly the vibration at the end of rod. In order to determine each value of stiffness at three main constraints, parameter studies were performed. Each constraint has three spring element such as Kr, Kθ, Kz in cylindrical coordinate system. By changing stiffness of spring elements with purpose function of correlation to experiments.

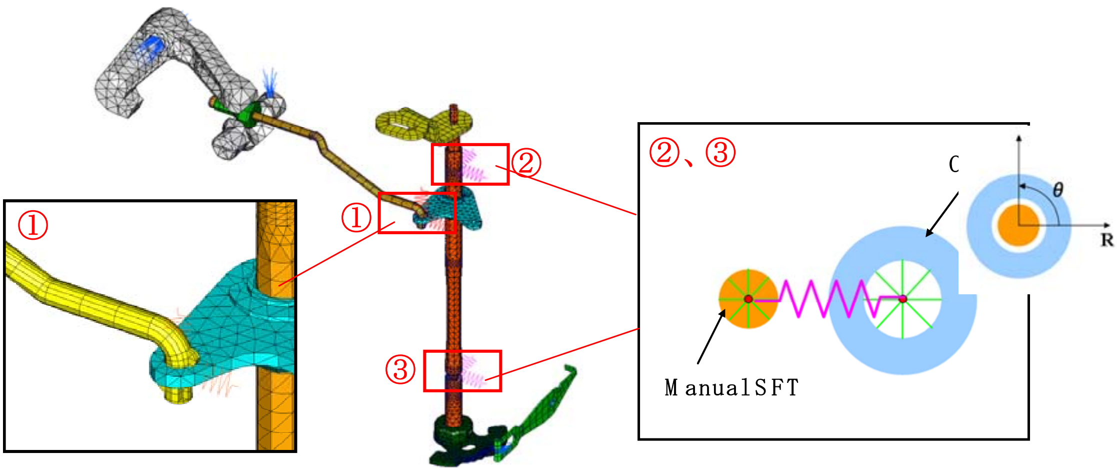

Figure 5. three main constraints affecting vibration

3. EXPERIMENT & RESULTS

3.1 MODAL TEST OF SUB ASSEMBLIES

In order to validate FE model of sub assemblies as seen in fig. 6 we performed comparison of experiment and simulation about modal frequencies of rod subassembly and manual shaft sub assembly. Table 1 shows comparison results of experiment and simulation. Frequencies of first mode and second mode of each assembly are almost same. This results shows that FE model of sub assemblies are reasonable.

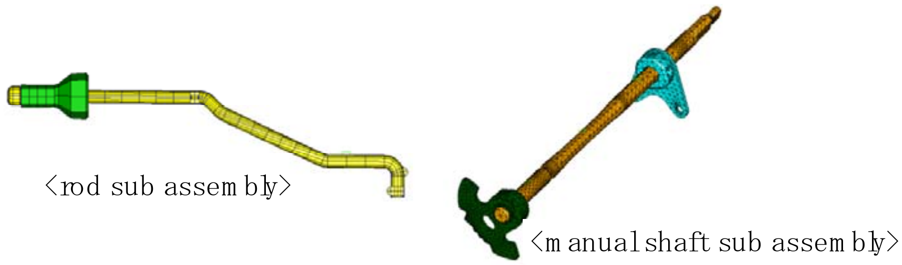

Figure 6. Rod and manual shaft sub assembly

Table 1. Normal mode frequency comparison

	Normal mode	Experiment	Simulation	Difference
rod ass'y	1st mode	580Hz	585Hz	0.9%
	2nd mode	1715Hz	1668Hz	-2.7%
manual shaft ass'y	1st mode	450Hz	450Hz	0.0%
	2nd mode	1250Hz	1298Hz	3.8%

3.2 VIBRATION CAUSED BY CONTACT FORCE

Fig. 7 shows experimental setup to measure vibration at the end of manual shaft. We measured 3 acceleration data as vibration in three translation direction. Then 3 acceleration data is converted value of magnitude by using rms synthesis.

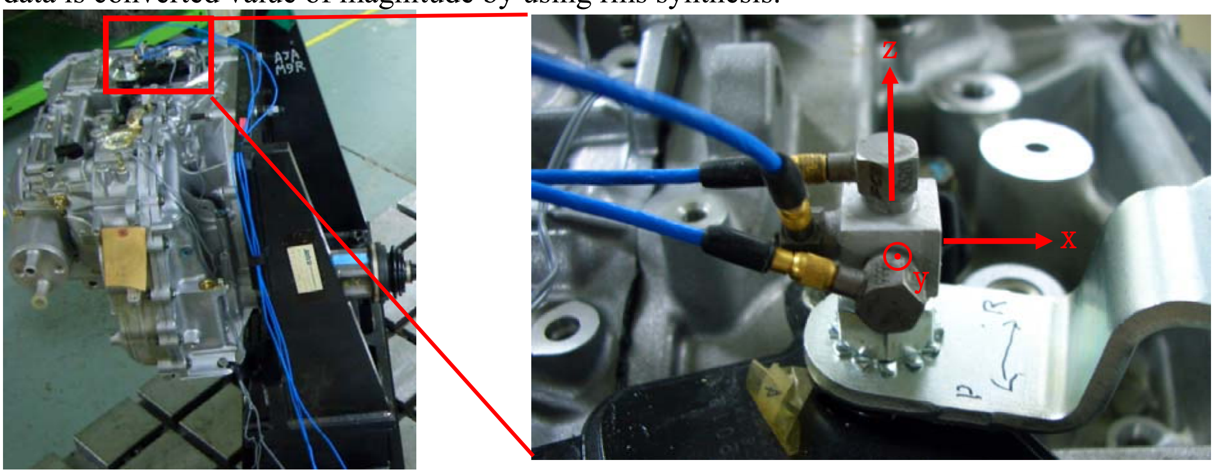

Figure 7. Experimental setup

Experiments was performed below order (from 1> to 5>) to validate simulation. Experiments were also carried out repeatedly at 3rpm, 6rpm, 10rpm of rotation speed of parking gear.

1> putting parking pawl on upper side of parking gear to prevent from rotation of paring pawl
2> moving select lever form R to P while rod spring is compressed
3> rotating parking gear to put parking pawl into lower side of gear
4> occurrence of contact wedge and tube
5> measuring transferred vibration to the end of manual shaft

Figs. 8, 9, 10 are respectively results of vibration at 3rpm, 6rpm and 10rpm of parking gear. The results show that simulation and experiment have same tendency like acceleration Z < acceleration X < acceleration Y. Difference of dB between experiments and simulations seem to be caused by damping in entire system of AT. In this study we have not included considering damping factor as parameters in entire system of AT instead of applying 5% modal damping to all modes.

Fig. 11 represents correlation result between experiments and simulations. As seen in fig. 11, correlation coefficient R^2 is 0.95. From this R^2 value, simulation method has good relation with experiment. It is possible to predict vibration using simulation method.

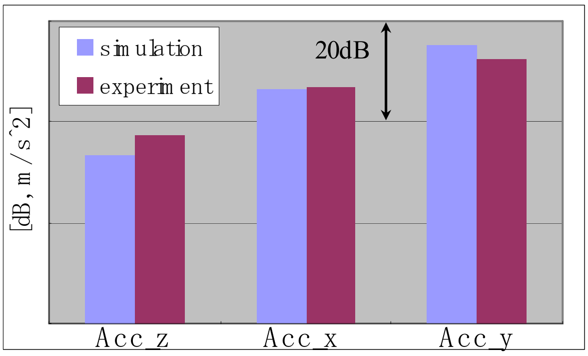

Figure.8 Vibrations at 3rpm of parking gear

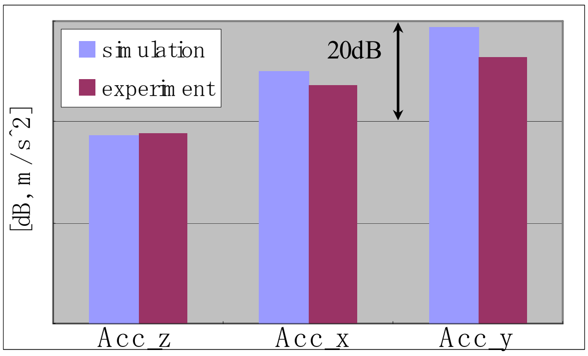

Figure.9 Vibrations at 6rpm of parking gear

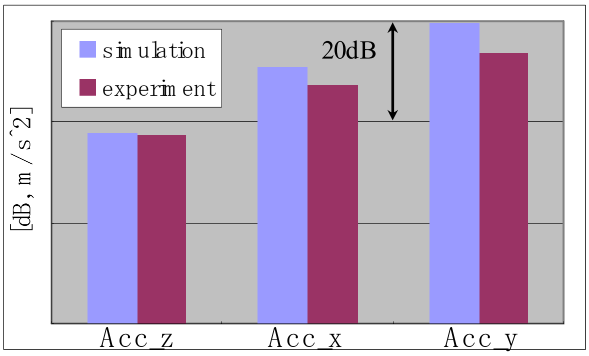

Figure.10 Vibrations at 10rpm of parking gear

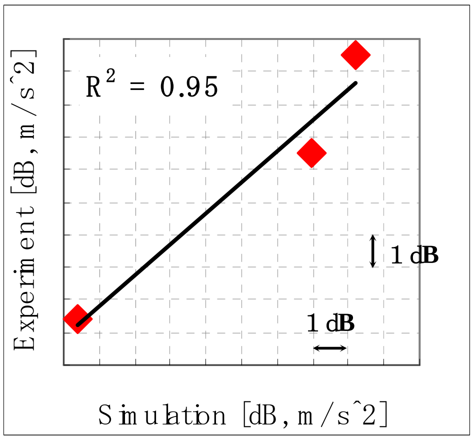

Figure 11. Correlation result of simulation and experiment

4. SUMMARY

First we developed a simulation method that can predict vibration as source of interior noise.
Second, we built experimental set up and performed test to verify simulation.
Third, we were able to identify reliability of the simulation through a correlation between experiment result and simulation result.

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