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EFFECTS OF THE VEHICLE MODEL ON SHIFTING TRANSIENTS OF PASSENGER CARS WITH AUTOMATIC TRANSMISSION

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ABSTRACT−This paper presents a vehicle model for analyzing the transient shifting characterisitics of a passenger car with automatic transmission. Then the presented vehicle model was linked with the dynamic model of an automatic transmission. In order to identify the parameters of the vehicle model, we installed a test equipment with an accelerometer in a conventional vehicle and performed road tests. With the proposed vehicle model, we simulated the dynamic characteristics during shifting, and benchmarked with experimental results. Moreover, a modal analysis was carried out to investigate the effect of the vehicle model in the frequency domain and to obtain the transfer function of the vehicle model. In addition, we showed the numerical results in the time domain for analyzing the effect of each stiffness element, such as engine mountings and suspensions.

KEY WORDS : Automatic transmission (AT), Transient shifting characterisitics, Shifting shock, Shifting quality, Vehicle model

1. INTRODUCTION

The transient shifting characterisitics of a passenger car with automatic transmission are generally investigated by examining the torque of a propeller shaft. However, the shock during shifting that is felt by passengers, is not directly dependent on the output torque of transmission, but on the accelerations of the vehicle body. For that reason, it is essential to consider both the powertrain system and the vehicle system simultaneously when investigating the shifting transients or shift quality.

In this paper, we propose a vehicle model to be used for analyzing transient shifting characterisitics. The vehicle model is composed of the housing of engine and automatic transmission (AT), a simplified suspension system, tires, and a vehicle body. The parameters of the vehicle model were determined based on the experimental results. The vehicle model is then coupled with the detailed automatic transmission model, which we investigated and verified in the previous works (Lim *et al.*, 1994; Kim *et al.*, 1994; Jo *et al.*, 2000).

In this paper, we analyze the transient shifting characterisitics from the 1st to 2nd gear power-on upshift by applying numerical simulation with the proposed vehicle model. To analyze the effects of the vehicle model, we also conduct a simulation with AT model only, which does not contain the vehicle model. In order to investigate both the reliability and the effectiveness of the proposed model, we compare the simulation results with the experimental results. In addition, we separate the vehicle model from the full model and calculate the frequency response to transmission output torque for a more comprehensive examination of the vehicle model (Zhang *et al.*, 2002).

2. DYNAMIC MODELING

2.1. Powertrain System

Figure 1 shows the powertrain system of a target passenger car. This system consists of an internal combustion engine, a torque converter with lock-up clutch, three planetary gear sets, and numerous wet multi-plate clutches and brakes. This power transmission model has been described in detail in our previous work (Jo *et al.*, 2000).

The torque of a gasoline engine was modeled as a function of engine speed and throttle opening, and the one dimensional performance model was used as the

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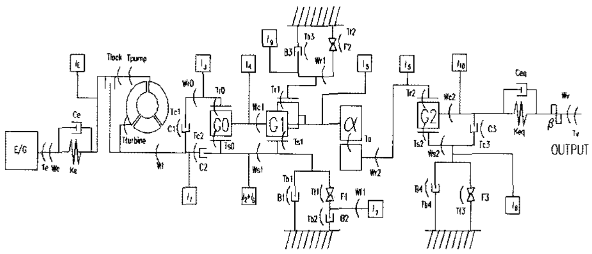

Figure 1. Schematic of the powertrain system.

torque converter model. The dynamic equations of a torque converter model have four state variables as follows: a pump velocity, a turbine velocity, a stator velocity, and a flow rate (Lim *et al.*, 1994). Backlash effect and stiffness of all gears are ignored in this study.

Since friction clutches and brakes in the automatic transmission play a very important role in shifting, we were very careful in developing the appropriate model and simulation method. The clutch pressure profiles, which acts on each clutch, was obtained from dynamometer experiments. For the simulations, we used POTAS-MSM which can analyze a dynamic model whose degree of freedom could change variously (Lim *et al.*, 1997, 2000).

2.2. Vehicle System

Figure 2 shows the vehicle model proposed in this study. This vehicle model is the extended version of a conventional bicycle model. It is composed of 4 inertias; a vehicle body, an engine/AT housing, frontal tires, and rear tires. Frontal (or rear) two tires are regarded as one inertia, because only 2 dimensional motion is considered. Each inertia has 3 DOFs (Degrees Of Freedom), namely are surging, pitching, and bouncing motion. 3-DOF is the maximum number of DOFs that can be considered for a two dimensional motion.

The most important parts of the proposed vehicle model are the connections between a vehicle body and powertrain system such as an engine, an AT, and frontal tires. From Figure 2 and Figure 3, the connections between a vehicle body and engine/AT housing are indicated as ① and ②, and those between a vehicle body

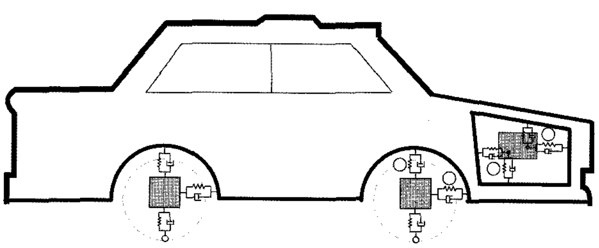

Figure 2. Schematic of the vehicle model.

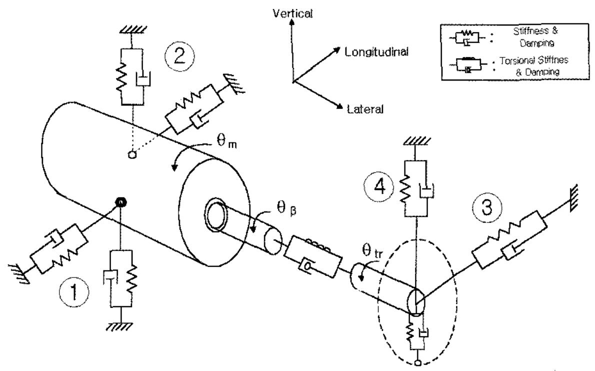

Figure 3. Schematic of engine/AT housing and frontal tires.

and frontal tires are as ③ and ④. These connections are modeled in equivalent stiffness with viscous damping. θ_m is an angular displacement of engine/AT housing, θ_β is one of the propeller shaft, and θ_{tire} is one of the frontal tire. The propeller shaft was modeled in equivalent stiffness with viscous damping; the torsional angle of that stiffness can be described as $\theta_\beta - \theta_m - \theta_{\text{tire}}$, which makes the pitching motion of engine/AT housing to become coupled to a driving force or shift shock. Here, it is assumed that the tires keep contact with the road in vertical direction.

Figure 4 illustrates a FBD (Free Body Diagram) of the vehicle body. F_a is an air drag force assumed to be exerted on the centroid of vehicle body. Other forces (F) or torques (T) are internal forces of equivalent stiffness. d_{mx} and d_{my} are distances between the centroid of vehicle body and the centroid of engine/AT housing. From Figure 4, the force exerted on vehicle body from engine mountings is not located on the centroid of engine/AT housing. The locations of engine mountings are illustrated in Figure 5.

Figure 5 illustrates a FBD of an engine/AT housing.

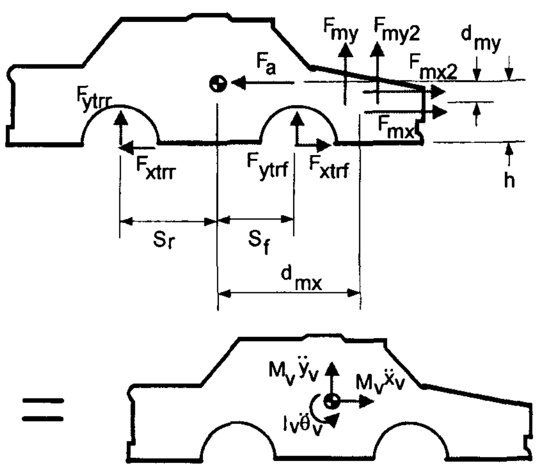

Figure 4. Free body diagram of vehicle body.

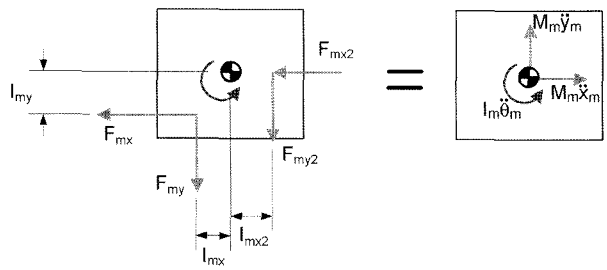

Figure 5. Free body diagram of engine/AT housing.

The pitching motion of engine/AT housing is induced by the output torque of AT and also by moments generated from forces such as F_{mx}, F_{my}, F_{mx2}, and F_{my2}. The constants such as l_{mx}, l_{my}, and l_{my2} are distances between engine mounting and the centroid of engine/AT housing. It is obvious that pitching motion of engine/AT housing does not have any effect on either longitudinal or vertical motion of vehicle body in the case where the above-mentioned distances are zero. Stiffness or damping coefficients of engine mountings and distances such as l_{mx}, l_{my}, and l_{my2} were obtained from the experimental results in the reference (Kim, 1995).

Identification of the unknown parameters such as equivalent stiffness and dimensions of the vehicle model will be described in the next chapter.

3. EXPERIMENTS

3.1. On-road Test

In order to identify the parameters of the vehicle model and to verify the simulation results of shifting transients, some experiments were performed using a real car. We measured the engine velocity, turbine velocity, AT output velocity, and throttle position by using sensors that are already installed in the car. Then, we added three accelerometers and a gyro sensor to measure accelerations of three directions and a pitch rate. The sampling rate was

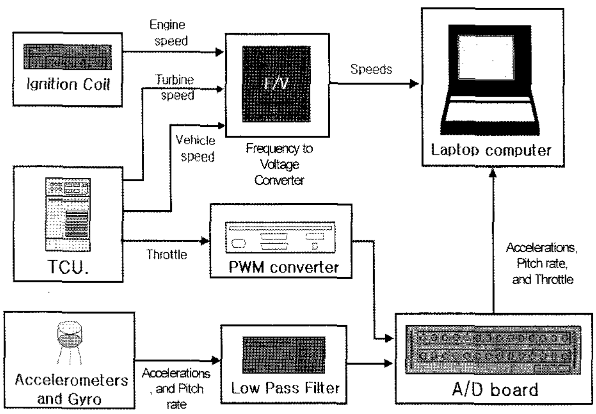

Figure 6. Schematic of on road test system.

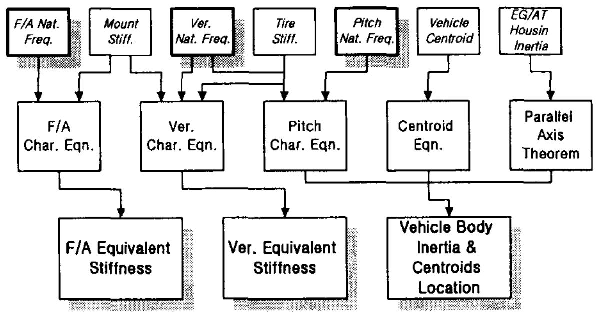

Figure 7. Parameter identification.

100 Hz and all the signals were acquired onto a laptop. Figure 6 shows the experiment system briefly.

3.2. Parameter Identification

All data for modeling powertrain system, such as engine performance, torque converter performance, inertias, gear ratio, and clutch specification, are already known from the previous work (Jo *et al.*, 2000).

Figure 7 shows the procedure of the parameter identification. The top row of Figure 7 displays data used for identification. The thick framed data were measured from experiments, and the others were already known data (Kim, 1995; Fenton, 1998). Using simplified characteristic equations of motion in each direction, and centroid equation with the parallel axis theorem, we calculated fore/aft equivalent stiffness (indicated as ③ in Figure 2), vertical equivalent stiffness (indicated as ④ in Figure 2), vehicle body inertia, and centroid locations.

4. ANALYSIS IN TIME DOMAIN

We investigated shifting transients that occur during 1st to 2nd gear upshift in the case of WOT (Wide Open Throttle). In this case, the magnitude of shift shock is very large and the shifting takes up a longer time compared to other cases; so that this case can be the most discomfort shifting for passengers, which is the reason why we choose this case for this study.

Two simulations were performed. The first simulation was performed with the full model which was composed of AT and the vehicle model. Then, the simulation with AT only model was carried out. Since we measured fore/aft acceleration (in longitudinal direction) as shifting transients in the experiment, we had to convert the output torque data of the simulation with AT only model into fore/aft acceleration. Fore/aft acceleration can be evaluated from the following equation:

$$a_{fore/aft} = \frac{T}{M \cdot r_t} \tag{1}$$

where, is fore/aft acceleration, is output torque of AT, is

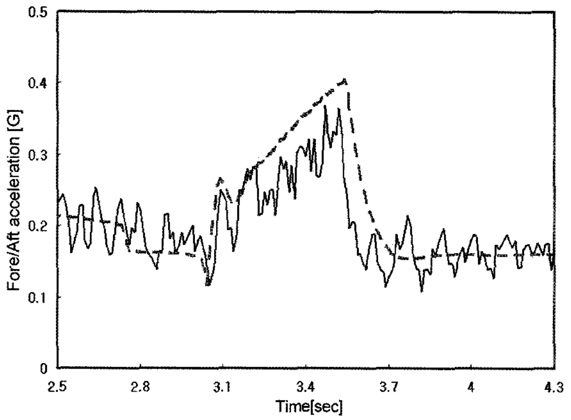

Figure 8. Fore/aft accelerations during 1→2 upshift (solid: experiment, dotted:simulation w/o vehicle model).

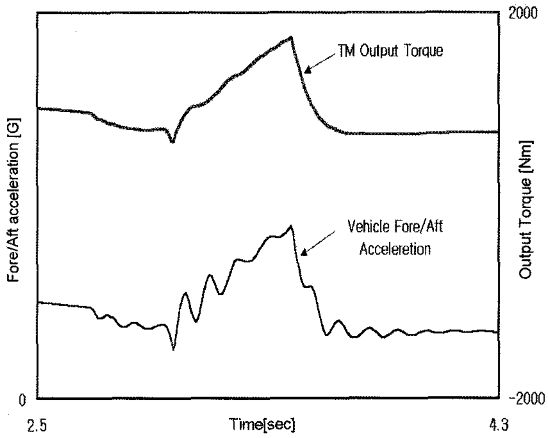

Figure 10. Simulation results - AT output torque and fore/aft acceleration.

total mass of vehicle, and is the equivalent radius of tire.

The solid line denotes experimental result whereas the dotted line denotes the simulation results. The dotted line in Figure 8 indicates the simulation result obtained by analyzing AT only model, which does not have the vehicle model. The gear change can be seen to start at about 2.7 seconds and completed at 3.7 seconds. At 3.1 seconds mark, we can see that the torque phase ends and inertia phase begins; the acceleration increases rapidly. At the end of the shifting, the acceleration drops steeply to a stationary value. In Figure 8, the simulation results agree with those of experiments approximately, but we can still see certain discrepancies. While the vibration around 10 Hz is detected in the experimental result, the result of simulation with AT only model does not show that vibration.

As shown in Figure 9, the vibration observed in experimental result is observed from simulation with the full model which includes the vehicle model.

It is apparent that the existence of the vehicle model is the reason for the difference between the two simulation results. In order to closely analyze the reason of the vibration around 10Hz, we plotted the reaction forces obtained from full model simulation in Figure 10, 11.

Figure 10 shows an output torque of AT and fore/aft acceleration of a vehicle body. Although the simulation was performed with a full model which includes the vehicle model, the vibration around 10 Hz can not be observed in the output torque, which means that the vehicle model hardly affects the output torque. Moreover, that vibration can be simulated only with the vehicle model.

F_{sus} in Figure 11 denotes the reaction force on connection ③ in Figure 3, and F_{mount} is the sum of longitudinal reaction forces of connection ① and ②. Similarly with the output torque in Figure 10, the vibration is not observed in the plot of F_{sus}. That

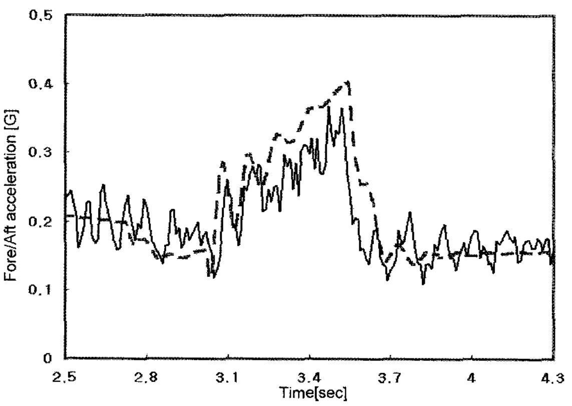

Figure 9. Fore/aft accelerations during 1→2 upshift (solid: experiment, dotted: simulation with vehicle model).

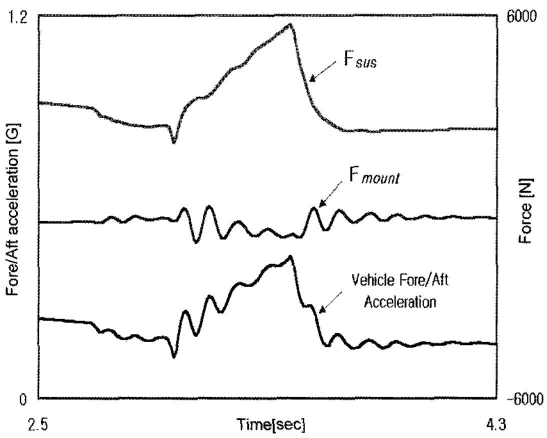

Figure 11. Simulation results - Fore/aft acceleration and forces exerted on vehicle body.

vibration, however, appears in F_{mount}. Based on the fact that most of the parts of the driving force is a sum of those two forces, we can conclude that the transient motion of engine/AT housing is the cause of the vibration around 10 Hz that was observed in vehicle fore/aft acceleration during or immediately after shifting.

5. ANALYSIS IN FREQUENCY DOMAIN

In order to mathematically analyze the effects on shifting transients caused exclusively by the vehicle model, we separated the vehicle model from the full model. The output torque of AT was assumed to be exerted on a propeller shaft like an external excitation, and tires were assumed to roll without slip. The vehicle model, thereupon, becomes an 11-DOF linear vibration system with a rigid mode.

The equation (2) describes 11 displacement variables of the vehicle model. is the displacement in longitudinal (fore/aft) direction, is the displacement in vertical direction, and is the rotating angle or pitching angle. Meanings of subscriptions are as follows:

v : vehicle,
m : engine/AT housing,
trf : frontal tire,
trr : rear tire,
β : propeller shaft.

The equation of motion for the vehicle system can be obtained to be equation (3). For formulating eigen-value problem, damping terms are ignored. The size of inertia matrix M or stiffness matrix K is 11 by 11. Fourier transforming equation (3) yielded equation (4). The frequency response function $\alpha(\omega)$ whose size is 11 by 11 can be defined as equation (5).

$$x = [x_v \;\; x_m \;\; x_{trr} \;\; y_v \;\; y_m \;\; y_{trf} \;\; y_{trr} \;\; \theta_v \;\; \theta_m \;\; \theta_\beta]^T \tag{2}$$

$$M\ddot{x} + Kx = f \tag{3}$$

$$X = (K - \omega^2 M)^{-1} F \tag{4}$$

$$\alpha(\omega) \equiv (K - \omega^2 M)^{-1} \tag{5}$$

Since our object is to analyze the effects on the fore/aft acceleration caused by the output torque of AT, it is important to choose an appropriate element of $\alpha(\omega)$ and calculate an inertance.

The output torque exerted on the vehicle-only model as an external excitation affects the engine/AT housing and the propeller shaft simultaneously. As already shown in Figure 11, however, the shifting transients transmitted via propeller shaft or tires are very similar to the output torque of AT. In order to investigate the effects of engine/AT housing, we chose the (1,10) element of $\alpha(\omega)$ to be defined as $\alpha^*(\omega)$.

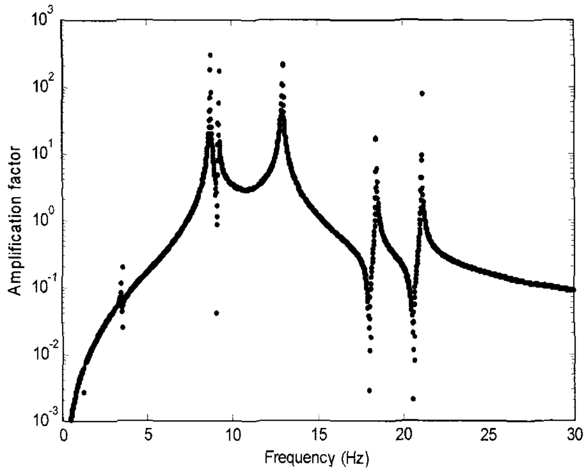

Figure 12. Amplification factor ϕ.

$$|\omega^2 X_v| = |\omega^2 \alpha^*(\omega)||\tau_{out}| \tag{6}$$

The equation (6) describes an inertance equation, where X_v is a Fourier transform of x_v, τ_{out} is a Fourier transform of T_{out}. The part of right side $|\omega^2\alpha^*(\omega)|$ is the magnitude of the inertance, but it is difficult to understand from a quantitative point of view. For this reason, a dimensionless number ϕ was defined as shown in equation (7).

$$|\omega^2 X_v| = |M_v r_t \omega^2 \alpha^*(\omega)|\left|\frac{\tau_{out}}{M_v r_t}\right| \equiv \phi\left|\frac{\tau_{out}}{M_v r_t}\right| \tag{7}$$

The vehicle total mass M_v and a radius of tire were chosen as scaling factors to non-dimensionalize the inertance. The magnitude of ϕ is not an exact amplification factor, but merely an approximate. If ϕ is much greater than 1 in some frequency bands, that band of the output torque can be amplified.

In Figure 12, ϕ is plotted against frequency (Hz). As mentioned before, ϕ is a scaled response of vehicle fore/aft acceleration excited by the output torque of AT via engine mountings. Since stiffness of engine mountings are relatively smaller than that of other vehicle part, the vibration of low frequency about 0~7 Hz is isolated as shown in Figure 12. It is worthy to note in Figure 12 that the dominant natural frequencies are clustered around 10 Hz. Moreover, the magnitude of ϕ around 10 Hz is much greater than 1, which means that this specific band of the output torque can be amplified. This analysis supports the simulation results as presented in Chapter 4 in a frequency-based point of view.

6. CONCLUSIONS

In this paper, a vehicle model consisting of equivalent stiffness with viscous damping was developed to be coupled with a conventional AT model. By performing experiments with a real car, we obtained the various parameters of the vehicle model and shifting transients during the 1st to 2nd gear upshift.

The transient behavior during shifting was simulated with the full model. In addition, a simulation with AT only model was also carried out; and by comparing the two simulation results and experimental results, we could make the following conclusions.

(1) Peak-to-peak magnitude and trend of shifting shock from two simulation results were similar to each other.
(2) However, adding the vehicle model makes it possible to simulate the vibration around 10 Hz.
(3) The vibration was caused by the transient behavior of engine/AT housing.

For analyzing the effects of the vehicle model in mathematical manner, we separated the vehicle model from the full model and formulated an eigen-value problem. The inertance of the vehicle model was calculated and scaled to dimensionless amplification factor ϕ. By examining ϕ in the frequency domain, we could conclude that the band around 10 Hz of the output torque was amplified, and transmitted to vehicle body via engine mountings.

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