

Technology to Predict Input Force to Engine Mount at Rapid Start of MT Vehicle

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ABSTRACT

Technology was created that gives the capability to predict engine mount input force during rapid starts. This input force is used for the strength design of the engine mounts for cars with a manual transmission (MT). Clutch engagement during a rapid start was modeled from time series characteristics of the clutch clamping force derived from actual measurements of clutch pedal delay and engagement time. The engine mount was modeled by applying force-displacement curves that had been tested up to high load. Wheel torque and engine mount input force were successfully predicted by means of full vehicle simulation using multi-body analysis. The parameters of that model were then changed to the upper limit and lower limit values within their obtainable range to confirm the rate of change in wheel torque and engine mount input force.

1. Introduction

There is demand for further shortened development times in order to respond to customer needs. In order to meet this demand, design accuracy will have to be increased in the initial stages of development. In strength areas, strength is evaluated on the basis of input force to the vehicle that occurs while actually driving. Therefore, the input force to each region has to be predicted accurately. There have been reports of techniques for accomplishing this by input force prediction technology with full-vehicle simulation using multi-body analysis⁽¹⁾⁻⁽⁵⁾. These are being used as replacement techniques for measurement of inputs on actual vehicles and for simple input prediction based on vehicle specifications. Many of these cases, however, focus on suspension input force and body input force from uneven road surfaces such as when driving on a rough road or running over a curb. None of them appear to have focused on engine mount input force from driving reaction force, which is needed for reinforced engine mount design.

The full-vehicle model used for strength areas was, therefore, augmented with a drivetrain model that includes a clutch and an engine-mount model that takes account of non-linear high loads. The addition of these models created a technology that would be able to predict engine mount input force during rapid starts in an MT vehicle. That model was used to study the influence of vehicle parameters on engine mount input force, and those results will also be reported.

2. Rapid Starts in MT Vehicles

In an MT vehicle, a standing start involves half-clutch operation, but the rapid start addressed in this paper involves keeping the engine at a higher speed than usual and, while in that state, releasing the clutch pedal and simultaneously pressing down on the accelerator. This causes the clutch to engage suddenly so that input force exceeding that from engine torque occurs in the drivetrain and the engine mount. The vehicle's rapid start is accompanied by wheel spinning. This scenario assumes misuse in the actual use environment.

Figure 1 shows the main input force and the behavior that occurs. The arrows indicate the direction of the force from the road surface side received on the body side. The

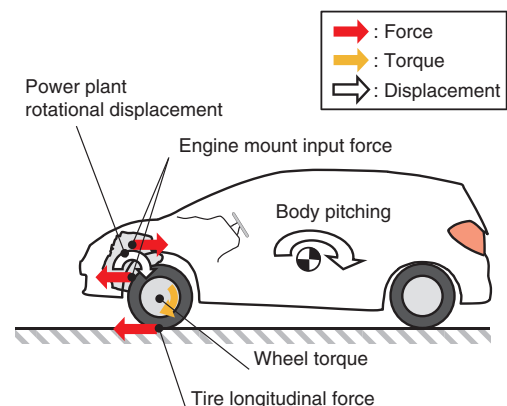


Fig. 1 Rapid start of MT vehicle

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drive torque transmitted from the engine and clutch to the wheels causes a longitudinal force to be exerted at the tire contact patch, and the vehicle body exhibits pitching behavior while the rapid start occurs. At the same time, the power plant displays a rotational behavior caused by the driving reaction force, and the engine mount, which supports the power plant, receives an input force.

Engine mount strength is evaluated with bench tests using the peak value for input force during rapid starts. Therefore, input force peak values have to be predicted accurately.

3. Overview of the Models

The engine mount input force that occurs during rapid start in an MT vehicle is influenced by numerous physical elements of the vehicle. Multi-body analysis was, therefore, used to create a full-vehicle model (Fig. 2).

3.1. Engine Model and Clutch Model

The engine was modeled with the engine torque map for each throttle position, as shown in Fig. 3. The time series data for throttle position corresponding to accelerator pedal operation during rapid start is input, the engine torque is calculated, and that load is applied to the flywheel.

Figure 4 shows the clutch on an MT vehicle. The clutch model uses the clutch coefficient of friction μ , clamping

force F , and effective radius R in Eq. (1) to calculate the transmission torque T .

$$T = 2\mu FR \quad (1)$$

It was decided to use the time series characteristics of the clamping force to model the clutch engagement resulting from clutch pedal operation during rapid start. In order to examine the time series characteristics, therefore, actual measurements were made of the clutch pedal stroke and mainshaft torque during rapid starts. One example of this is shown in Fig. 5(a). When the driver's foot is taken off the clutch pedal, the pedal stroke begins. It was found, however, that mainshaft torque does not rise at the same time. There is a delay of approximately 0.04 sec before it rises. It was also found that clutch engagement takes the relatively short time of approximately 0.05 sec. The time series characteristics obtained from these results, shown in Fig. 5(b), were input to the model. The delay and the time taken to engage the clutch may vary with the model of vehicle, but the time series characteristics can be changed in

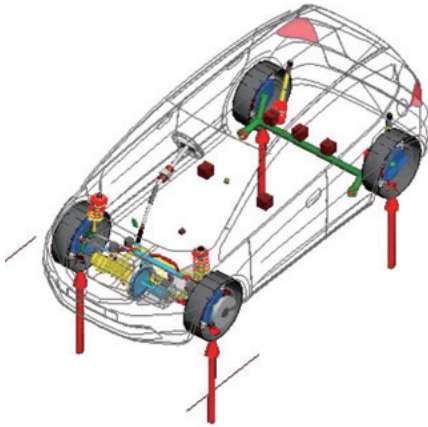


Fig. 2 Full vehicle model

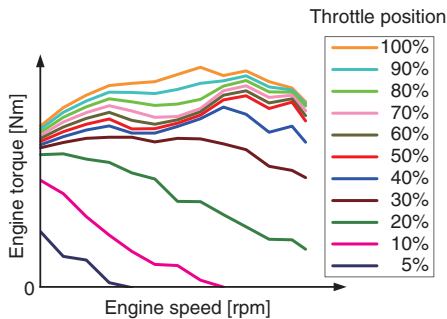


Fig. 3 Engine torque map

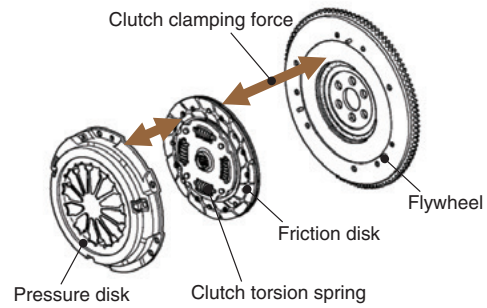
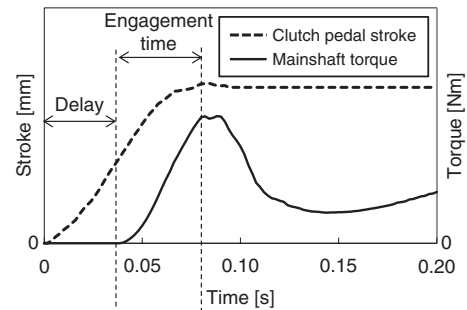
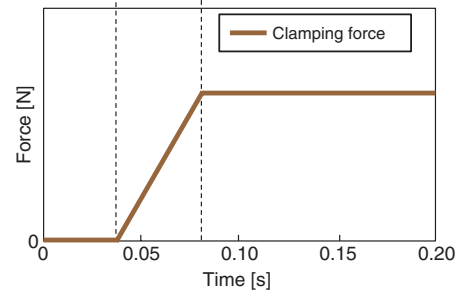


Fig. 4 Main structure of MT clutch



(a) Measurement data



(b) Model input data

Fig. 5 Identification of clutch clamping force characteristic

order to model such vehicles.

The clutch torsion spring and the driveshaft were modeled by their respective torsional characteristics.

3.2. Other Models

The engine mount was modeled using force-displacement curves tested up to high load, as shown in Fig. 6, to account for the magnitude of the loads that occur with a rapid start. This took account of the influence from the bush spring rise characteristics.

The suspension was modeled as a multi-body model with rigid-body links (Fig. 7).

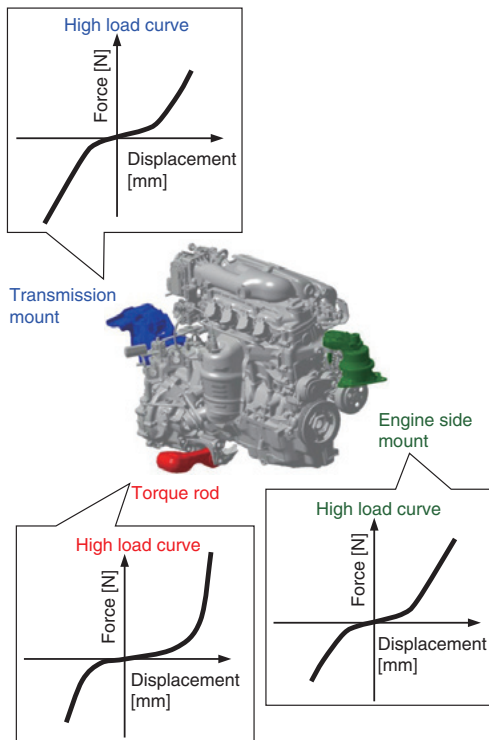


Fig. 6 Engine mount model

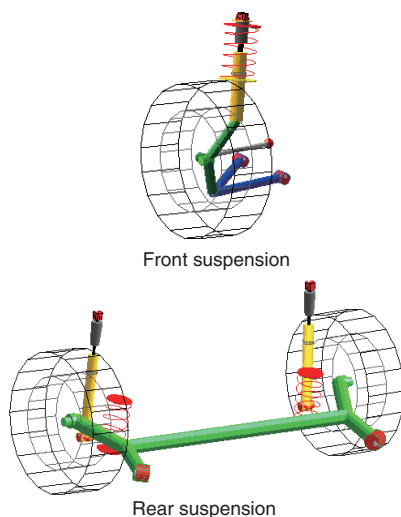


Fig. 7 Suspension model

The tire model uses the Magic formula capable of calculating the longitudinal force of the tire from the wheel load, the slip ratio, and so on.

4. Simulation Verification Results

A comparison was made of the simulation and actual measurement values for wheel torque and engine mount input force, which are driving reaction forces.

4.1. Small Vehicle Comparison Results

Figure 8 shows the results of wheel torque comparison for a small vehicle with a 1.5 L-displacement gasoline engine. Wheel torque rises due to the rapid start, and the first peak on the waveform is the maximum peak. It was confirmed that the peak value differential is slight and the time-series waveforms also matched closely.

Figure 9 shows the results of engine mount input force comparison. With the engine mount input force, as with wheel torque, the first peak on the waveform was the maximum peak. The peak value differential, which is needed for strength evaluation, was slight, and this model was confirmed to have adequate predictive accuracy.

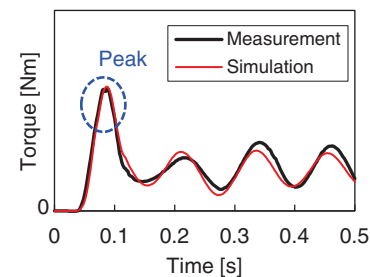


Fig. 8 Wheel torque comparison for small vehicle

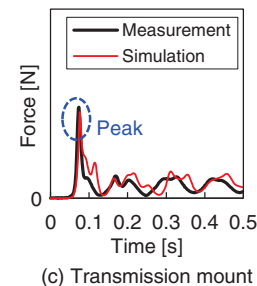
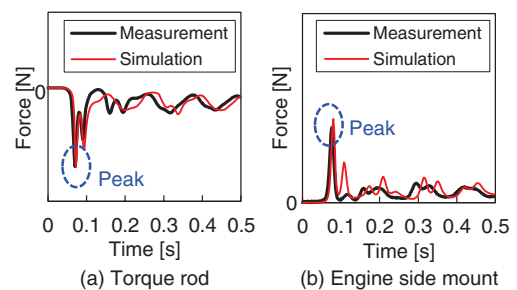


Fig. 9 Engine mount input force comparison for small vehicle

4.2. Large Vehicle Comparison Results

The simulation and actual measurement values were also compared on a large vehicle with a 2.2 L-displacement diesel engine, which has greater torque. Wheel torque comparison results are shown in Fig. 10 and the engine mount input force comparison results in Fig. 11. The peak value differentials for both wheel torque and engine mount input force can be seen to be slight. This confirmed predictive capability with large vehicles as well as with small vehicles.

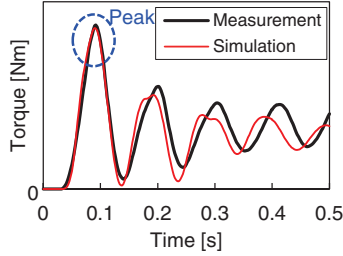


Fig. 10 Wheel torque comparison for large vehicle

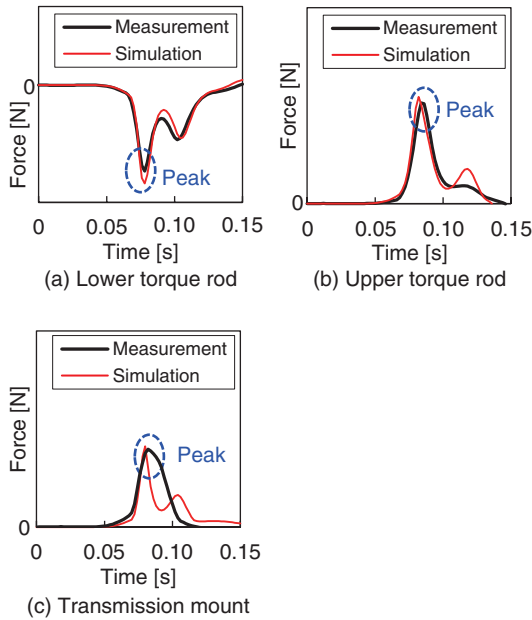


Fig. 11 Engine mount input force comparison for large vehicle

5. Influence of Parameters on Input Force

5.1. Input Force Reduction Parameters

It would be useful for design work in the initial stages of development to start with a grasp of the vehicle parameters that have greater influence on input force and of the tendencies for changes to these parameters to increase or reduce input force. The vehicle parameters shown in Table 1 were, therefore, changed to their upper-limit and lower-limit values within their respective obtainable ranges. The

Table 1 Vehicle parameter

Power plant	Power plant center of gravity coordinate
	Power plant mass
	Power plant inertia moment
	Flywheel inertia moment
Drivetrain	Clutch clamping force
	Clutch engagement time
	Mainshaft inertia moment
	Driveshaft torsional stiffness
Engine mount	Engine mount coordinate
	Engine mount bush spring
	Engine mount stopper clearance
Suspension	Suspension coil spring constant
	Suspension buffer clearance
Tire	Wheel inertia moment
	Tire friction
	Tire radius
	Tire inertia moment
Vehicle body	Body inertia moment

rates of change of the wheel torque peak value and engine mount input force peak value were confirmed. The focus in this paper is on the reduction rates.

Figure 12 shows the parameters that had a higher wheel torque reduction rate, together with their reduction rates. It can be seen that the clutch engagement time, clutch clamping force, and driveshaft torsional stiffness have higher reduction rates.

Figure 13 shows the parameters that had a higher torque rod input force reduction rate, together with their reduction rates. It is apparent that parameters with higher wheel torque reduction rates generally also have higher torque rod input force reduction rates. Others are the torque

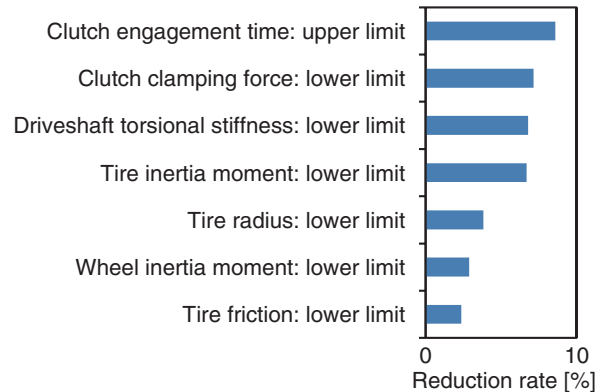


Fig. 12 Wheel torque reduction rate

rod stopper clearance, vertical coordinate, bush spring, and power plant inertia moment. This suggests that engine mount input force can be reduced by drivetrain parameters and engine mount parameters.

5.2. Influence of Torque Rod Stopper Clearance

As described above, the stopper clearance has a relatively large influence on torque rod input force. When the clearance is smaller, the input force is reduced. In order to verify this simulation result, different torque rods with stopper clearance of 2 mm, 6 mm, and 11 mm were fabricated and measurements were made of the input force on an actual vehicle.

Figure 14 shows the results for comparison of the simulation and actual measurements. The vertical axis shows the rate of input force change with respect to torque rod input force with a stopper clearance of 6 mm. The horizontal axis shows the stopper clearance. This confirmed that the actual measurements showed the same tendencies as in the simulation to increase or reduce the input force.

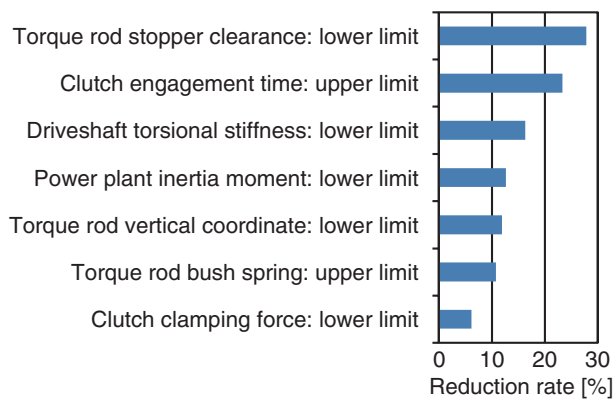


Fig. 13 Torque rod input force reduction rate

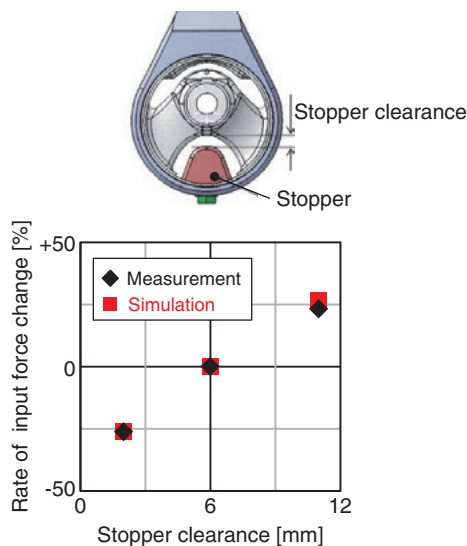


Fig. 14 Rate of input force change to stopper clearance

This also confirmed that the rate of change was predicted with adequate accuracy.

The inertial force of power plant dynamic rotational behavior during rapid starts has a relatively large influence on torque rod input force. Consequently, the input force is thought to be smaller when the stopper clearance is smaller, because the stopper then takes effect before the angular velocity of the power plant has increased. The way in which a smaller power plant inertia moment results in a reduced torque rod input force can also be considered to be due to the influence of the inertia force.

5.3. Influence of Suspension System Parameters

When suspension system parameters are changed, change also takes place in the wheel load movement that results from vehicle body pitching behavior during rapid starts. This is accompanied by change in the longitudinal force of the tire, which is thought to cause increase or decrease in the wheel torque and engine mount input force. The influence of suspension system parameters, however, was slight.

Measurements were therefore made of vertical force at the front wheel center during rapid starts and the wheel load movement was confirmed. Figure 15 shows the actual measurement results together with the simulation results. It is apparent, first of all, that the simulation corresponds closely with the actual measurements, and that the model successfully predicts wheel load movement. The focus was next placed on wheel load from before rapid start to the time when wheel torque peaks, and it was found to remain virtually unchanged. Thus, the occurrence of an input force peak, as indicated in chapter 4, is a briefer phenomenon when wheel load movement does not take place. This is thought to be the reason that suspension system parameters have such little influence.

The fact that engine mount input force can be predicted regardless of the suspension indicates that input force can be predicted at an early stage of vehicle development, for example, even if the suspension characteristics have not been decided.

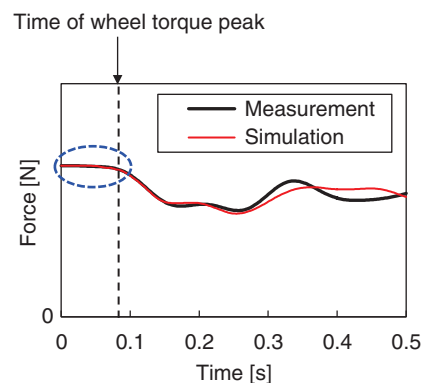


Fig. 15 Vertical force at front wheel center

6. Conclusion

A full-vehicle model of an MT vehicle was created, rapid start simulation was conducted, and the following results were obtained.

- (1) Wheel torque peak values and engine mount input force peak values on a small vehicle and a large vehicle were compared with actual measurements, and this model was confirmed to have adequate predictive accuracy.
- (2) Clutch engagement time, clutch clamping force, and driveshaft torsional stiffness were found to have relatively large influence on wheel torque peak values.
- (3) Other than wheel torque, the engine mount stopper clearance, vertical coordinate, bush spring, and power plant inertia moment were found to have relatively large influence on engine mount input force peak values.

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