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Development of Rigid-Body HEV Model for HILS Certification Test Method

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Abstract

There are many vehicle types such as different gear ratios and different bodies in heavy-duty conventional vehicle (HD-CV) category. Therefore, in Japan, following simulation method is used for evaluating fuel economy and exhaust emissions of HD-CVs: 1) Engine speed and torque is calculated with dedicated program; 2) After that, fuel economy is calculated by adding fuel consumption map, and exhaust emissions are measured with engine unit on engine dynamometer. Concerning the test method for heavy-duty hybrid electric vehicles (HD-HEVs), must be evaluated as a vehicle system because they consist of not only an engine but also other components such as a motor generator and a rechargeable energy storage system featuring regenerative braking, and operating condition of these components are controlled by hybrid electronic control unit. However, chassis dynamometer testing method for HD-HEVs takes thousands of man-hours and expensive. Therefore, a hardware-in-the-loop simulator (HILS) method is developed as technical guidelines in 2007. Currently, the HILS method is being discussed for developing national technical regulations and globally-harmonized technical regulations. Therefore, all information of the HEV model which has currently non-disclosure part must be fully disclosed. This open-source HEV model corresponding to parallel HEVs equipped with an automated manual transmission or a manual transmission.

Keywords: hardware-in-the-loop (HIL), HEV (hybrid electric vehicle), simulation

1 Introduction

One of the causes of global warming is considered to be greenhouse gases such as carbon dioxide emitted from motor vehicles. As a means of reducing automotive CO_2 emissions, hybrid electric vehicles including heavy-duty ones (HEVs) have been placed in the market. An HEV is comprised of a motor generator and a rechargeable energy storage system in addition to an engine, which are each mobilized separately according to running conditions. As a consequence the exhaust emission and fuel economy performances of an HEV must be evaluated on the whole vehicle rather than on the engine alone. The man-hours and costs required by HEV exhaust emission and fuel economy tests using a chassis dynamometer are further increased by the great diversity of body shapes and gear ratios involved and by the necessary correction of state of charge (SOC) before and after the test.

Concerning the method of testing the HEV's emissions and fuel economy by hardware-in-the-loop simulation (HILS)[1], the Japanese Ministry of Land, Infrastructure, Transport and Tourism in March 2007 issued HILS Technical Guidelines “Kokujikan No.281 and No.282”[2]. Currently preparations are underway to develop a domestic HILS technical regulation and to introduce a harmonized global HILS regulation. Although to this end all information on HEV simulation models must be disclosed, such disclosure has been hindered by the software makers who hold copyrights on their drive train models, making it necessary for other parties to develop an original drive train model that will be opened to the public.

Developed in the present study was an open-source rigid-body drive train model that corresponded with the parallel HEVs having an automated manual transmission (AMT) or a manual transmission (MT). This drive train model was evaluated by an HIL simulator at JE05 driving cycle, and the results confirmed that the new rigid-body drive train model gives sufficient accuracy for use in heavy-duty HEV certification tests.

2 Outline of the HILS Method

replace the conversion program of the existing test method with an HIL simulator.

In the case of an HEV whose engine load conditions are controlled by control logic, engine load conditions are determined by an electric control unit (ECU) and an HIL simulator consisting of various HEV element models. Once engine load conditions are derived, the HEV's fuel economy and emissions are measured in a way similar to conventional vehicles.

According to the existing fuel economy test method for heavy-duty vehicles, engine load conditions (rotational frequency, torque) are derived from the conversion program[3] issued by the Ministry of the Environment; then the fuel economy of a vehicle is calculated from the derived engine load conditions and a fuel consumption map. Similarly according to the existing exhaust emissions test method for heavy-duty vehicles, the same engine load conditions are reproduced by a stand-alone engine placed on an engine dynamometer, where the measured amount of test emissions is equivalent to the amount of emissions from an actually driven vehicle. As shown in Figure 1, the HILS method is designed to.

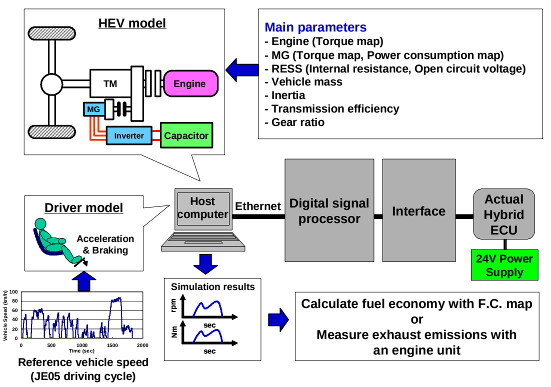

Fig. 1 Outline of the HILS method

3 Development of a Rigid-Body Drive Train Model

For the development of a drive train model in the present study, three development principles were adopted: (1) observe physical laws, (2) simulate the transient motion of vehicles, (3) do not use any uncertain elements. Developed on the concept of a spring-mass system, conventional drive train models involve axle torsional rigidity and damping coefficient as parameters. Since these parameters can be uncertainty factors, however, the present drive train model was developed on the basis of rigid body mechanics to preclude uncertainties.

The new rigid-body drive train model for parallel HEVs consisted of three principal elements: a final gear model, a transmission model, and a clutch system model (Fig. 2). The transmission model was adaptive to both AMT and MT, while the development of a transmission model accommodative to torque converter AT is still underway. The calculation formulas used in those elements are explained in the succeeding sections.

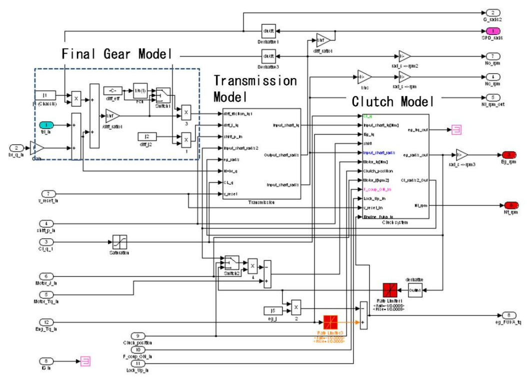

Fig. 2 Rigid-body drive train model

3.1 3.1 Final gear model

In the final gear model, transmission output shaft load torque was calculated using the formulas given below.

During drive:

$$T_O = \frac{1}{\eta_{diff}\, n_{diff}}\left(T_{br} + T_{trl} + j_{chassis}\frac{d}{dt}N_S\right) + j_{diff}\frac{d}{dt}N_O \quad (3\text{-}1)$$

During regeneration:

$$T_O = \frac{\eta_{diff}}{n_{diff}}\left(T_{br} + T_{trl} + j_{chassis}\frac{d}{dt}N_S\right) + j_{diff}\frac{d}{dt}N_O \quad (3\text{-}2)$$

In the above formulas, T_O, T_{br}, T_{trl}, η_{diff}, n_{diff}, $j_{chassis}$,

j_{diff}, N_S, and N_O denote respectively transmission output shaft load torque, brake torque, running resistance, differential gear efficiency, differential gear ratio, chassis inertia, differential gear inertia, vehicle speed, and output shaft rotational frequency.

To calculate the required driving power of the rigid-body drive train model, first, the value of acceleration resistance was derived from chassis inertia and tire inertia at identical speeds; second, this acceleration resistance value was added to the sum of brake torque and the torque equivalent with the running resistance; third, this total sum of torques, after modification by final gear transfer loss, was added to the acceleration resistance value derived from final gear inertia; finally, the value of transmission output shaft load torque was obtained.

3.2 Transmission model

In the transmission model (Fig. 3), calculated from the above-obtained transmission output shaft load torque and input shaft rotational acceleration were input shaft load torque, input shaft rotational frequency, and output shaft rotational frequency. Assuming the gear shift position to be "m", the formulas given below were used to calculate input shaft load torque.

During driving:

$$T_I = \frac{1}{\eta_m n_m} T_O + j_m \frac{d}{dt} N_I \qquad (3\text{-}3)$$

During regeneration:

$$T_I = \frac{\eta_{deff}}{n_{deff}} T_O + j_m \frac{d}{dt} N_I \qquad (3\text{-}4)$$

In the above formulas, T_I, η_m, n_m, j_m, and N_I stand respectively for input shaft load torque, transmission transfer efficiency at gear shift position m, transmission gear ratio at gear shift position m, transmission gear inertia at gear shift position m, and input shaft rotational frequency.

Transmission transfer loss and transmission acceleration resistance were first derived from the transmission gear ratio, gear efficiency, and gear inertia all at shift position m, and then were modified by the output shaft load torque inputted from the above-explained final gear model in order to calculate the value of input shaft load torque.

Next, the value of input shaft rotational acceleration obtained from the clutch system was multiplied by gear ratio to derive output shaft rotational acceleration; this output acceleration was integrated to give output shaft rotational frequency. This output frequency was multiplied by transmission gear ratio to derive input shaft rotational frequency at the gear shift positions other than neutral. This is because, since the rotational mass from the output shaft to the tires were greater than the input shaft rotational mass, input shaft rotational frequency was influenced by output shaft rotational frequency. When the gear shift was in the neutral position, the integrated value of input shaft rotational acceleration was outputted as input shaft rotational frequency.

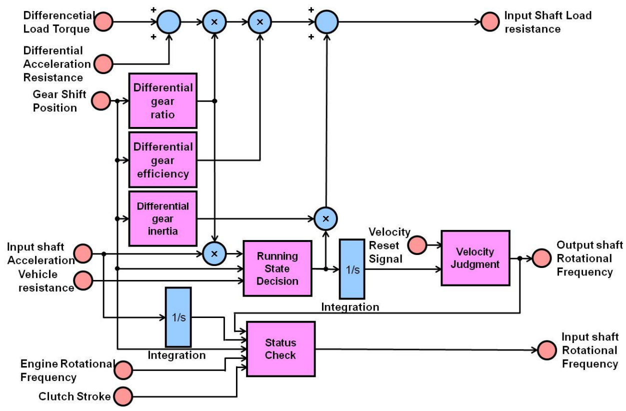

Fig. 3 Diagram of the transmission model

3.3 Clutch system model

In the clutch system model, two values derived from the transmission model (i.e., input shaft load torque and input shaft rotational frequency) along with engine torque and motor torque were used to calculate engine rotational frequency and input shaft rotational acceleration. As shown by the clutch system model diagram in Figure 4, this model was comprised of a clutch model, a fluid coupling/lock-up clutch (FC/LC) model, and an engine revolution calculation model. Motor torque to be transferred to the input shaft was added on the engine side or on the input shaft side, depending on the location of the motor. The calculation methods for the various element models are explained in the succeeding sections.

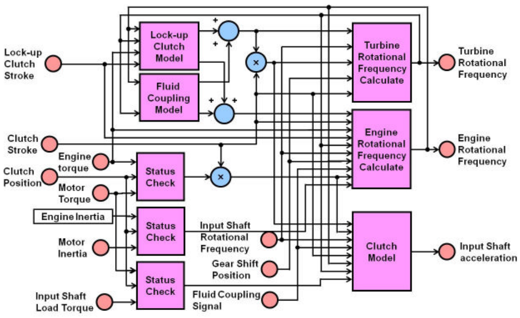

Fig. 4 Diagram of the clutch system model

3.3.1 Clutch model

In the clutch model, the value of input shaft rotational acceleration was calculated from the input shaft load torque inputted from the transmission model and from the difference between engine torque and motor torque both transferred to the input shaft, using the formulas given below.

Clutch fully engaged:

$$\alpha_I = \frac{T_T - T_{IL}}{j_I} \tag{3-5}$$

Clutch being engaged:

$$\alpha_I = \frac{NT_T - N_I T_{IL}}{j_I N_I} \tag{3-6}$$

In the above formulas α_I, T_T, T_{IL}, j_I, N, and N_I denote respectively input shaft rotational acceleration, clutch transferred torque, input shaft load torque, input shaft rotational inertia, clutch rotational frequency on the engine side, and input shaft rotational frequency.

Torque on the engine side was transferred according to clutch strokes. However, when the clutch was fully engaged, rotational frequency was identical between engine side and input shaft side so that the amount of transfer could be calculated in terms of torque. When the clutch had yet to be fully engaged, rotational frequency could differ between engine side and input shaft side so that the amount of transfer was calculated in terms of output power and was then converted into torque.

3.3.2 FC/LC models

In the FC/LC models, fluid coupling input/output torque, and engine output torque were calculated from engine revolution and turbine revolution.

The calculated fluid coupling output torque was added to lock-up clutch output power, and the sum was multiplied by clutch stroke to derive the torque transferred to input shaft. The torque not transferred was treated as surplus torque from which turbine rotational frequency was calculated; at the same time, the calculated turbine rotational frequency was synchronized with input shaft rotational frequency according to clutch strokes.

The resultant fluid coupling input torque and lock-up clutch input torque were added together to give engine load torque which was in turn used to calculate engine revolution.

a)FC model

In the FC model, FC characteristics were expressed in terms of e (the ratio of turbine rotational frequency to pump rotational frequency), t(e) (the ratio of output shaft torque to input shaft torque for e), and C(e) (capacity coefficient for e). The values of e, t(e), and C(e) were obtained by the formulas given below.

During forward drive:

$$e = \frac{N_O}{N_I} \tag{3-7}$$

$$t(e) = \frac{T_O}{T_I} \tag{3-8}$$

$$C(e) = \frac{T_I}{N_I^2} \tag{3-9}$$

During reverse drive:

$$e = \frac{N_I}{N_O} \tag{3-10}$$

$$t(e) = \frac{T_I}{T_O} \tag{3-11}$$

$$C(e) = \frac{T_O}{N_O^2} \tag{3-12}$$

In the above formulas, N_I, N_O, T_I, and T_O stand respectively for input shaft rotational frequency, output shaft rotational frequency, input shaft torque, and output shaft torque. Forward drive is define as condition N_I-$N_O > 0$, and reverse drive as N_I-$N_O < 0$.

The values of input shaft torque and output shaft torque were derived by the following formulas which were modified from above formulas (3-8), (3-9), (3-11), and (3-12).

During forward drive:

$$T_I = C(e) N_I^2 \tag{3-13}$$

$$T_O = t(e) T_I \tag{3-14}$$

During reverse drive:

$$T_O = C(e) N_O^2 \tag{3-15}$$

$$T_I = t(e) T_O \tag{3-16}$$

Maps of t(e) and C(e) for e were prepared by using the above four formulas. As a result it became possible to calculate the values of input shaft torque and output shaft torque at any input shaft rotational frequency and output shaft rotational frequency by using the FC model diagramed in Figure 5.

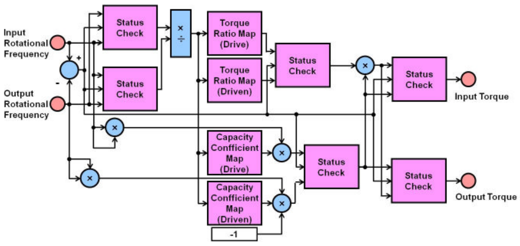

Fig. 5 Diagram of the FC model

b) LC model

In the LC model the torque from input shaft was transferred to the output shaft according to clutch strokes. When the clutch had yet to be fully engaged, rotational frequency may differ between input shaft and output shaft so that the amount of transfer was calculated in terms of output power, and was then converted into torque.

$$P_{LCO} = LC\_qN_{LCI}T_{LCI} \tag{3-17}$$

$$T_{LCO} = \frac{1}{N_{LCO}} P_{LCO} \tag{3-18}$$

In the above formulas, P_{LCO}, LC_q, N_{LCI}, N_{LCO}, T_{LCI}, and T_{LCO} denote respectively LC output power, LC stroke, LC input rotational frequency, LC output rotational frequency, LC input torque, and LC output torque.

c) Turbine rotational frequency calculation model

In the turbine rotational frequency calculation model, the increase of turbine rotational frequency was calculated, using the surplus FC/LC output torque that had yet to be transferred; at the same time, the calculated turbine rotational frequency was synchronized with input shaft rotational frequency according to clutch strokes.

3.3.3 Engine revolution calculation model

In the engine revolution calculation model, the increase of engine revolution was calculated, using the surplus engine torque that had yet to be transferred; at the same time, the calculated engine revolution was synchronized with input shaft rotational frequency according to clutch strokes. In the case of an HEV model incorporating an FC/LC, the calculated engine revolution was synchronized with turbine revolution according to lock-up clutch strokes.

Adopting the above-explained elements, the rigid-body drive train model developed in the present study was able to represent both parallel HEVs having a motor on the engine side and those having a motor on the transmission side, thus covering all the parallel HEVs existing in the Japanese market.

4 Verification of Accuracy

The accuracy of an HEV model incorporating the developed rigid-body drive train model was examined, using software-in-the-loop simulation (SILS) and HILS. The SILS was applied to examine if the behaviors of the HEV model and an existing spring-mass model were equivalent. Then the HILS was employed to compare the values obtained through simulation and those obtained through actual measurement in order to verify an accuracy level necessary for HEV certification tests.

4.1 Accuracy verification by SILS

For the SILS-aided accuracy verification of the developed model, the government-prescribed standard ECU model[2] was employed in the place of an ECU model regularly used in the control of a vehicle simulation model. This standard ECU model, programmed on MATLAB and Simulink, outputted fixed values according time as command values for the motor, engine, and brakes. In addition the standard ECU model outputted fixed values according to vehicle speed, as shift change commands. Consequently it was possible in the present study to check if each vehicle manufactures HILS system would respond differently to the same command value.

The SILS-aided verification results are shown in Figure 6, where the broken lines represent a spring-mass drive train model and the solid lines represent the developed rigid-body drive train model. Except for the shift change regions marked by circles in the graphs of motor revolution, motor torque, engine revolution, engine torque, and electric current, the rigid-body drive train model indicated operations similar to those of the spring-mass drive train model. Similarly, verification results of HILS were considered to fall within the permissible range since the conventional vehicle test method[2] requires to exclude from calculations the data obtained in the one-second periods before and after each shift change.

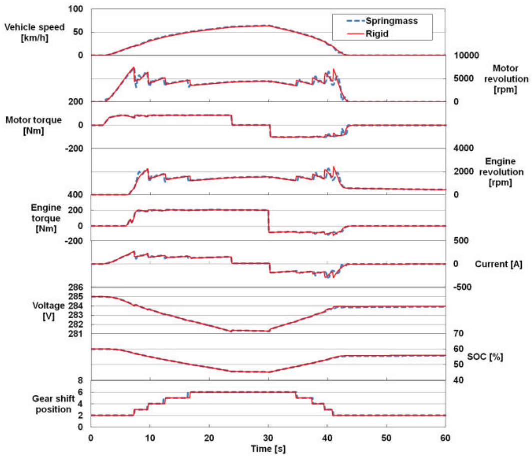

Fig. 6 SILS results

4.2 Accuracy verification by HILS

With the cooperation of automakers in Japan, the developed rigid-body drive train model was applied to four types of HEV models shown in Figure 7 (Vehicles A, B, C, D), and HILS-calculated values and measured values were compared. As defined in the HILS test method, verification was performed using the first-trip and the whole JE05 cycle. The first-trip verification was aimed to check if transient motion equivalent to that of a realworld vehicle could be reproduced by using the measured values of acceleration and brake operation amount in the verification. In the whole-cycle verification, the measured values or driver model values of acceleration and brake operation amount were used in order to check if the HILS method could ensure sufficiently high accuracy for certification tests. The driver model was able to output acceleration and brake operation amount from a targeted vehicle speed and from the values of vehicle speed, engine revolution, and running resistance outputted by an HEV model.

Table 1 shows the verification results and HILS determination coefficients obtained from the JE05 cycle first-trip verification on the four types of HEV models combined with the present rigid-body drive train model. Table 2 shows the corresponding results and determination coefficients obtained from the whole-cycle verification. The checkmarks in Tables 1 and 2 mean that the corresponding verification results and determination coefficients satisfied the criteria established in the HILS Technical Guidelines[2].

In the first-trip verification by HILS, the determination coefficients of measured and simulation values for vehicle speed or engine revolution, motor torque, motor output power, engine torque, engine output power, and rechargeable energy storage system (RESS) output power were judged against the criteria specified in the HILS Technical Guidelines. As given in Table 1, vehicle speed or engine revolution recorded min. 0.979, thus exceeding the criterion of 0.97≤. Similarly, motor torque (min. 0.902), motor output power (min. 0.945), engine torque (min. 0.892), engine output power (min. 0.931), and RESS output power (min. 0.935) all satisfy the criterion of 0.88≤.

In the whole-cycle verification by HILS, (1) the determination coefficients of measured and simulation values for vehicle speed or engine revolution and engine torque; (2) the ratios of measured value to simulation value for engine positive work and fuel economy were judged against the criteria specified in the HILS Technical Guidelines. As shown in Table 2, the determination coefficient for vehicle speed or engine revolution stood at min. 0.976, thus exceeding the criterion of 0.97≤. Similarly, engine torque (min. 0.881) exceeded its determination coefficient criterion of 0.88≤, engine positive work (min. 0.988) was above 0.97≤, and fuel economy (max. 0.999 or less) stood below 1.03≥, so that all the criteria were met.

Because both first-trip and whole-cycle verifications indicated compliance to the determination coefficient criteria by all the four HEVs, it was concluded that the developed rigid-body drive train model had sufficient simulation accuracy for use in certification tests.

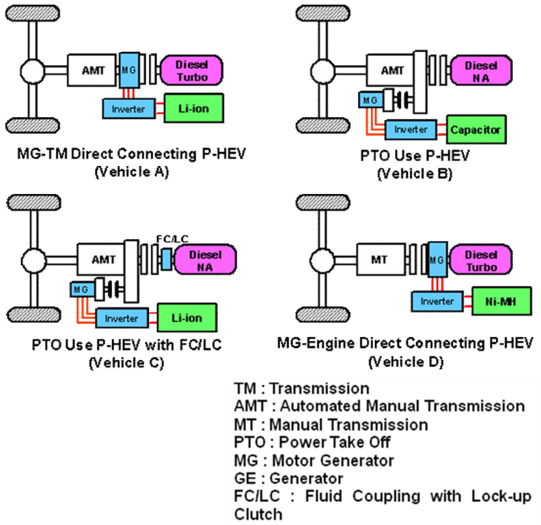

Fig. 7 Types of Japanese heavy-duty HEVs

Table 1 Verification results, JE05 first strip

	Vehicle speed or Engine revolution		Electric motor			
			Torque		Power	
Criteria	0.97≤		0.88≤		0.88≤	
Vehicle A	0.998	✓	0.979	✓	0.976	✓
Vehicle B	0.999	✓	0.902	✓	-	
Vehicle C	0.979	✓	0.961	✓	0.945	✓
Vehicle D	0.996	✓	0.981	✓	0.978	✓
	Engine				RESS	
	Torque		Power		Power	
Criteria	0.88≤		0.88≤		0.88≤	
Vehicle A	0.961	✓	0.973	✓	0.941	✓
Vehicle B	0.892	✓	0.973	✓	0.935	✓
Vehicle C	0.951	✓	0.976	✓	0.936	✓
Vehicle D	0.892	✓	0.931	✓	0.972	✓

Table 2 Verification results, JE05 whole cycle

	Vehicle speed or Engine		Engine torque		Engine positive work		Fuel economy	
Criteria	Determination coefficient		Determination coefficient		W_{eng} HILS / W_{eng} vehicle		FE_{HILS} / $FE_{vehicle}$	
	0.97≤		0.88≤		0.97≤		1.03≥	
Vehicle A	0.998	✓	0.961	✓	1.017	✓	0.986	✓
Vehicle B	0.995	✓	0.881	✓	1.047	✓	0.989	✓
Vehicle C	0.976	✓	0.915	✓	0.988	✓	0.995	✓
Vehicle D	0.994	✓	0.895	✓	1.003	✓	0.999	✓

5 Conclusion

A rigid-body drive train simulation model adaptive to both AMT and MT was developed for open-source HEV models. To verify the accuracy of the new drive train model, JE05 first-trip and whole-cycle verifications were conducted according to the fuel economy test method for heavy-duty HEVs prescribed in the HILS Technical Guidelines. Since all the verification results satisfied the criteria established in the Technical Guidance, it was confirmed that the developed drive train model had sufficient accuracy for use in certification tests. The Ministry of Land, Infrastructure, Transport and Tourism and the National Traffic Safety and Environment Laboratory will therefore be asked to use this model in certification operations. Furthermore, efforts will be exerted to develop a torque converter AT model to increase the types of heavy-duty HEV models usable for HILS-aided certification tests.

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References

[1] K. Morita et al., Development of a Fuel Economy and Exhaust Emission Test Method with HILS for Heavy-Duty HEVs, SAE Int, J. Engines, Volume 1, Issue 1, p.873-887 (2008)

[2] Official notices "Kokujikan No.281 and No.282" of the Ministry of Land, Infrastructure, Transport and Tourism

[3] Website of the Ministry of the Environment (http://www.env.go.jp/air/car/program/index.html)

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