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CORRECTION OF DYNAMIC WHEEL FORCES MEASURED ON ROAD SIMULATORS

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ABSTRACT

In a typical test procedure involving a road simulator, an elevation profile of a selected road section is measured and recorded using a computer data acquisition system. In general, actuators are unable to follow the input profile signals exactly due to an inherent inertia of the road simulator-vehicle system. As a result, the response of a vehicle tested on a road simulator is different from the response of the same vehicle tested on an actual road section. Several procedures have been developed to compensate for the actuators' dynamics and to ensure that the vehicle response on a road simulator is the same as in road tests. In general, there are two groups of methods that can be used to ensure that the results of road simulator tests are the same as the results obtained in tests on actual roads. In the first group of methods, called *Input-Correction Methods*, the original actuator input signal is adjusted iteratively until the vehicle response matches the response obtained in road tests. These methods are commonly known as Remote Parameter Control (RPC) methods. This paper presents an alternative method which can be classified as an *Output-Correction Method.* In the proposed method, only one road simulator test is performed using the original road profile as the input signal and the vehicle response measured in the test is subsequently processed to compensate for the effects of the road simulator's dynamics. In both groups of methods a certain knowledge of the system dynamics is required. In the input-correction methods, a transfer function of the entire system, i.e. the road simulator with the test vehicle, needs to be evaluated whereas in the output-correction methods only the transfer function of the road simulator itself is needed. In the proposed method, a nonlinear mathematical model of the simulator is first derived analytically and then linearized assuming small deviations from the normal operating point conditions. Results from the road simulator tests processed using the output-correction method are presented and compared with the results of tests conducted on actual roads.

INTRODUCTION

Road simulators are servo hydraulic actuation systems consisting of several (usually two, four, or six) hydraulic actuators whose vertical displacements are controlled by a computer system to simulate road profile excitations applied to the wheels of a vehicle traveling over the particular section of road. Road simulators have been used extensively by vehicle manufacturers in durability tests of vehicles and/or vehicle components [1, 2]. In several recent studies, road simulators were employed in tests of dynamic wheel loads generated by heavy vehicles traveling over uneven road surfaces [3, 4, 5,6]. The main reason for road simulators to be an attractive research tool in vehicle testing is that they allow for conducting vehicle tests under well controlled laboratory conditions without the logistical and weather problems that often arise in tests performed on actual in-service roads.

A truck-road simulator testing setup is shown in **figure 1**. The input to the system is the vector of measured road profiles, $\underline{x}_p$, which usually has two components, one for each (left and right) wheel track. The performance of the actuators is controlled by two analog controllers, a proportional gain controller and a differential pressure controller. The two controllers are tuned to obtain the best dynamic performance of the actuators such that the vector of the actuator displacements, $\underline{x}_a$, is as close as possible to the road profile vector, $\underline{x}_p$ [7].

In tests conducted on actual roads, the vehicle traveling over the particular section of road generates a dynamic response, which can be represented mathematically by a vehicle response vector $\underline{y}$. In tests performed on a road simulator, the measured road profile, $\underline{x}_p$, (which will be assumed here to be the same as the actual road profile) is used as input to the road simulator but the vehicle is now subjected to the displacements of the simulator actuators, represented by $\underline{x}_a$, which are different from the input profile signal due to inherent inertia of mechanical and hydraulic components of the actuators. Since the vehicle is subjected to different input signals in road simulator tests, its response, $\underline{y}^*$, is also different from the response, $\underline{y}$, generated in road tests. Block diagrams in **figure 2** illustrate the two testing arrangements. In order to obtain the same response in road simulator tests as the response that the vehicle would generate in actual road tests, the system shown in figure 2b has to be modified to compensate for the actuators' dynamics. In most existing methods developed for this purpose, the input road profile signal is modified iteratively until the desired system performance is obtained. These methods can be referred to as *Input Correction Methods*. A block diagram of the most common testing system utilizing an input correction method is shown in **figure 3**. In this system, the input signal is modified iteratively until

the response of the vehicle being tested, $\underline{y}$, is sufficiently close to the response, $\underline{y}_d$, measured in actual road tests. This system is known as the Remote Parameter Control (RPC) system and the algorithm employed in correcting the input signal in this system is discussed in [8]. Another input correction method was developed by Moran et al. [5]. In this method, the input signal is modified to generate actuator displacements, $\underline{x}_a$, that match the measured road profiles, $\underline{x}_p$, as illustrated in the block diagram shown in **figure 4**.

This paper presents an alternative method that can be used to ensure that results of road simulator tests agree with results of corresponding road tests. In the proposed method, which can be classified as an *Output Correction Method*, the input signal is not altered but, instead, the output, or the vehicle response, is processed to eliminate the effects of the actuator dynamics. A block diagram of the system with an output correction method is shown in **figure 5**. There is no need for iterative modifications of any of the signals when this method is used. It does, however, require the knowledge of the actuators transfer functions. The output correction algorithm is described next, followed by the analytical derivation of a suitable mathematical model of the actuator. Finally, some experimental results obtained in tests conducted on the Federal Highway Administrations simulator, called DYNTRAC (DYNamic Truck Actuation system), are presented.

OUTPUT CORRECTION ALGORITHM

It should be noted that both input and output correction algorithms are derived assuming that the testing system consisting of the actuators and the test vehicle is linear. Under this assumption the dynamics of the actuator and of the test vehicle can be represented by transfer functions $\mathbf{T}_a(s)$ and $\mathbf{T}_t(s)$. The response of the vehicle in road tests can be expressed as

$$\mathbf{Y}(s) = \mathbf{T}_t(s)\mathbf{X}_p(s) \tag{1}$$

where $\mathbf{X}_p(s)$ is a Laplace transform of the road profile signal. In road simulator tests, the vehicle response, $\mathbf{Y}^*(s)$, is additionally affected by the actuator dynamics

$$\mathbf{Y}^*(s) = \mathbf{T}_t(s)\mathbf{T}_a(s)\mathbf{X}_p(s) \tag{2}$$

The two vehicle responses given by equations 1 and 2 are obviously different. The vehicle response in road simulator tests can be modified to match the response obtained in road tests by multiplying it by the inverse transfer function of the actuator. The corrected vehicle response, $\mathbf{Y}^c(s)$, is then, theoretically, the same as the vehicle response recorded in road tests,

$$\mathbf{Y}^c(s) = \mathbf{Y}^*(s)\mathbf{T}_a^{-1}(s) = \mathbf{Y}(s) \tag{3}$$

To perform the output correction described by equation 3, the transfer function of the actuator must be known. The actuator transfer function can be identified from experimental results following a procedure similar to that employed in the input correction methods. In this paper, the actuator transfer function will be obtained by linearizing a nonlinear mathematical model of the actuator, assuming small deviations from the actuators normal operating point conditions.

MATHEMATICAL MODEL OF ACTUATOR

As indicated earlier, the road simulator used in this analysis is DYNTRAC, which was specifically designed to simulate dynamic forces applied by heavy trucks to pavements. It uses four 156 kN hydraulic actuators controlled by 380 lpm 3-stage servovalves. The actuators displacements are controlled using LVDT feedback for position control with a stroke of 15.2 cm. In addition, a differential pressure feedback is used for stabilization of large inertial loads. A schematic of a DYNTRAC actuator is shown in **figure 6**. A detailed nonlinear model of the actuator and a computer simulation package was described in [7]. The mathematical model has ten states and includes a second-order model of servovalve dynamics, oil compliance on each side of the actuator piston, the mass of the wheelpan and piston assembly, and a quarter-vehicle model which is excited by the wheelpan position. This nonlinear model has been simplified and linearized as described by Wang [9]. The following overall transfer function, relating actuator displacement to road profile signal, was derived for the linearized model

$$T_a(s) = \frac{X_a(s)}{X_p(s)} = \frac{b_1 s + b_0}{a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + a_0}$$

where:

$$b_0 = b_1 = a_0 = K_{sv} K_q K_e$$

$$a_1 = a_0 + (K_c + K_{dp} K_{sv} K_q) k_t / A_p^2$$

$$a_2 = (K_c + K_{dp} K_{sv} K_q) \frac{b_p}{A_p^2} + \frac{V_f k_t}{4 \beta_e A_p^2} + 1$$

$$a_3 = (K_c + K_{dp} K_{sv} K_q) \frac{m_p}{A_p^2} + \frac{V_f b_p}{4 \beta_e A_p^2}$$

$$a_4 = \frac{V_f m_p}{4 \beta_e A_p^2}$$

The above equations express the coefficients of the actuator transfer function in terms of parameters of the linearized model of the actuator (static gain of servovalve, K_{sv}, gains of the linearized model of servovalve, K_q and K_c, piston cross - sectional area, A_p, total volume of fluid in the actuator chambers, V_f, effective bulk modulus of fluid, β_e, total mass of piston and load, m_p, and coefficient of viscous friction, b_p), parameters of the analog controllers (K_e and K_{dp}), and vehicle tire stiffness, k_t [9].

In order to verify the accuracy of the linearization, the step response of the linearized model of DYNTRAC was compared with that of the nonlinear model. **Figure 7** shows the two step responses. It can be seen that the two models produce very similar responses.

The linearized model of DYNTRAC was further evaluated by comparing the dynamic wheel forces simulated with the nonlinear and linearized models with the forces measured in actual road tests. The dynamic wheel forces are characterized by a Dynamic Load Coefficient (DLC) defined as the ratio of standard deviation over mean wheel force. From the results shown in table 1, it can be seen that the value of DLC obtained in the road simulator test is considerably lower than the value calculated from the wheel forces measured in the road test due to the actuators' inability to follow the input road profile with sufficient accuracy, as discussed earlier. The results shown in table 1 also indicate an excellent agreement between the DLC's obtained in the road simulator test and the DLC's calculated from the nonlinear as well as the linearized model of the road simulator.

Table 1. Comparison of DLC's obtained from experimental and simulated data.

	Road Test	DYNTRAC Test	Nonlinear Model	Linearized Model
DLC	0.2105	0.1754	0.1748	0.1724

Figure 8 shows sample traces of wheel forces measured on DYNTRAC and processed using the output correction algorithm.

RESULTS OF OUTPUT CORRECTION METHOD

In the output correction algorithm, given by equation 3, the actuator displacement signal is multiplied by the inverse of the actuator transfer function defined by equation 4. It is quite clear that the fourth-order transfer function with only one zero, such as that given by equation 4, will lead to an inverse transfer function whose gain increases rapidly with frequency. This will make the correction algorithm extremely sensitive to a high frequency noise that is likely to contaminate the actuator displacement signal and may even result in a numerical instability of the correction process. To eliminate this problem, a low-pass filter was inserted in series with the inverse transfer function to supress the high-frequency components. The transfer function of the fourth-order low-pass filter used for this purpose is

$$T_F(s) = \frac{\omega_1^2}{s^2 + 2\zeta_1\omega_1 s + \omega_1^2} \bullet \frac{\omega_2^2}{s^2 + 2\zeta_2\omega_2 s + \omega_2^2}$$

The two corner frequencies, ω_1 and ω_2, were chosen to be approximately 16 and 24 Hz, respectively and both damping ratios, ζ_1 and ζ_2, were set to be 1. So, the modified inverted transfer function of the actuator is

$$T_a^{-1}(s) = \frac{a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + a_0}{b_1 s + b_0} \bullet \frac{\omega_1^2}{s^2 + 2\zeta_1\omega_1 s + \omega_1^2} \bullet \frac{\omega_2^2}{s^2 + 2\zeta_2\omega_2 s + \omega_2^2}$$

The output correction algorithm was applied to three data sets representing dynamic wheel forces measured on the road simulator for various combinations of road roughness, vehicle speed, and payload. The measured and corrected values of DLC were compared with the DLC's obtained in actual road tests under the same test conditions. The results are shown in table 2.

Table 2. Values of DLC from road, road simulator, and corrected wheel forces.

	Road	DYNTRAC	Corrected DYNTRAC
Roughness: medium Speed: 72 km/h Payload: 6360 kg	0.0985	0.0854	0.0994
Roughness: high Speed: 48 km/h Payload: 3640 kg	0.1912	0.1497	0.1794
Roughness: high Speed: 72 km/h Payload: 3640 kg	0.2105	0.1754	0.2139

It can be seen that the DLC's of the corrected wheel forces are very close to the DLC obtained from measurements conducted in road tests in two of the cases. In the third case the agreement between the road and corrected DLC is not that close but the corrected result is still much closer to the road test DLC than the DLC obtained from the forces simulated on the road simulator.

CONCLUSIONS

The output correction algorithm presented in this paper as an alternative method for processing dynamic wheel forces measured in tests conducted on road simulators was shown to produce values of DLC that are much closer to the DLC's obtained in tests on actual roads. The proposed correction algorithm uses a transfer function of the linearized mathematical model of the actuator, which was derived analytically from the basic equations of motion of the entire road simulator system. The inverted transfer function of the actuator is modified by a fourth-order low-pass filter to eliminate numerical instability of the correction algorithm caused by high-frequency components in the signals being processed. While the results obtained with the proposed method have been very promising, more work is needed in order to further validate this method, particularly with other road simulators systems.

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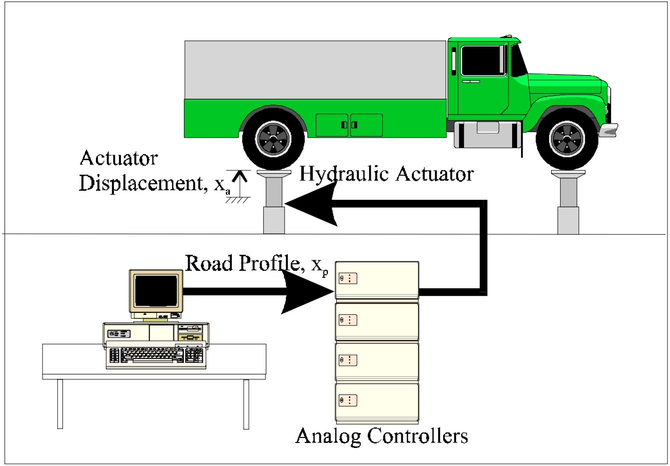

Figure 1. Truck-road simulator testing setup.

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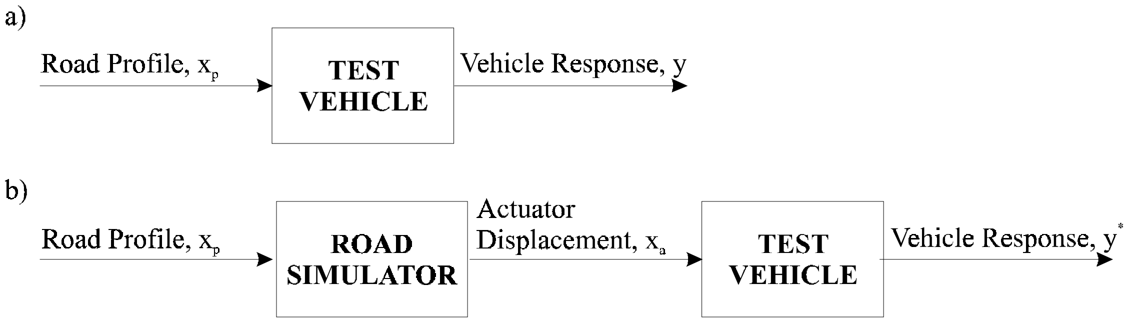

Figure 2. Block diagrams of road (a) and road simulator (b) testing setups.

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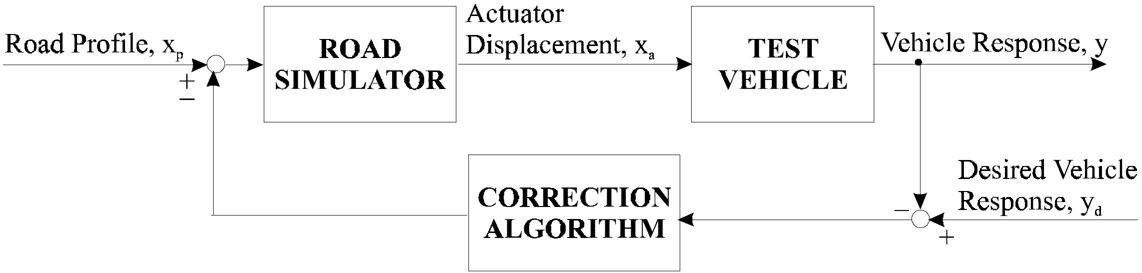

Figure 3. Block diagram of input correction system with RPC.

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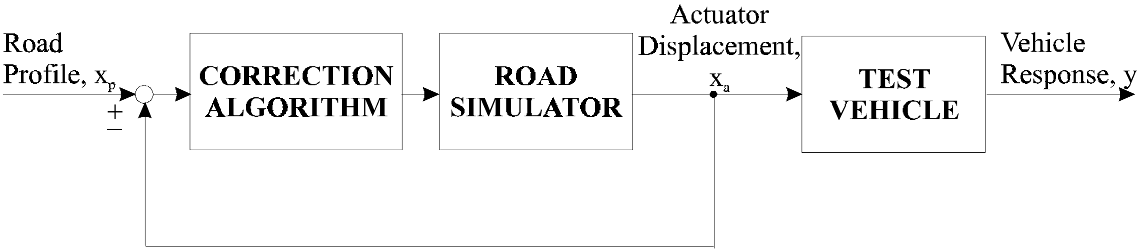

Figure 4. Block diagram of input correction system with actuator displacement control.

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Figure 5. Block diagram of output correction system.

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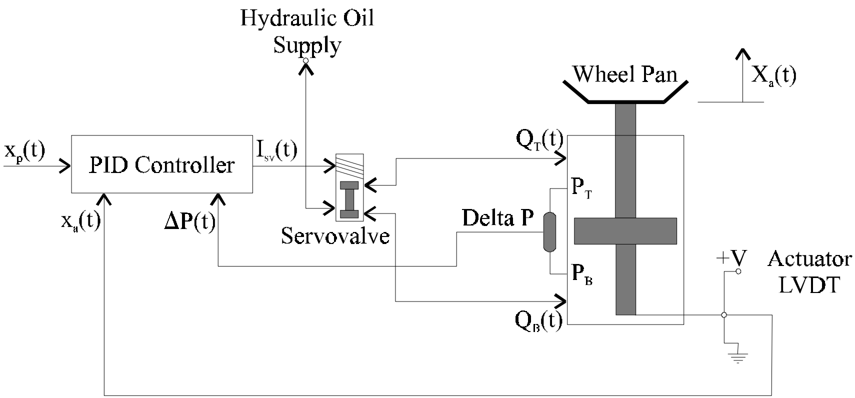

Figure 6. Schematic of the DYNTRAC actuation system.

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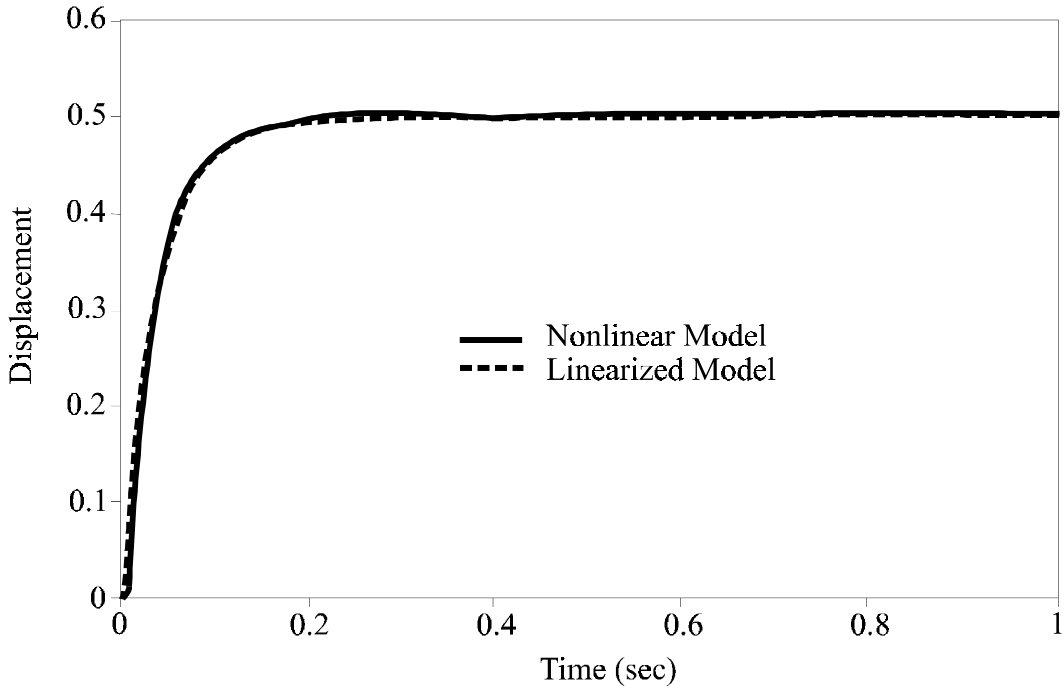

Figure 7. Step responses of nonlinear model and linearized model.

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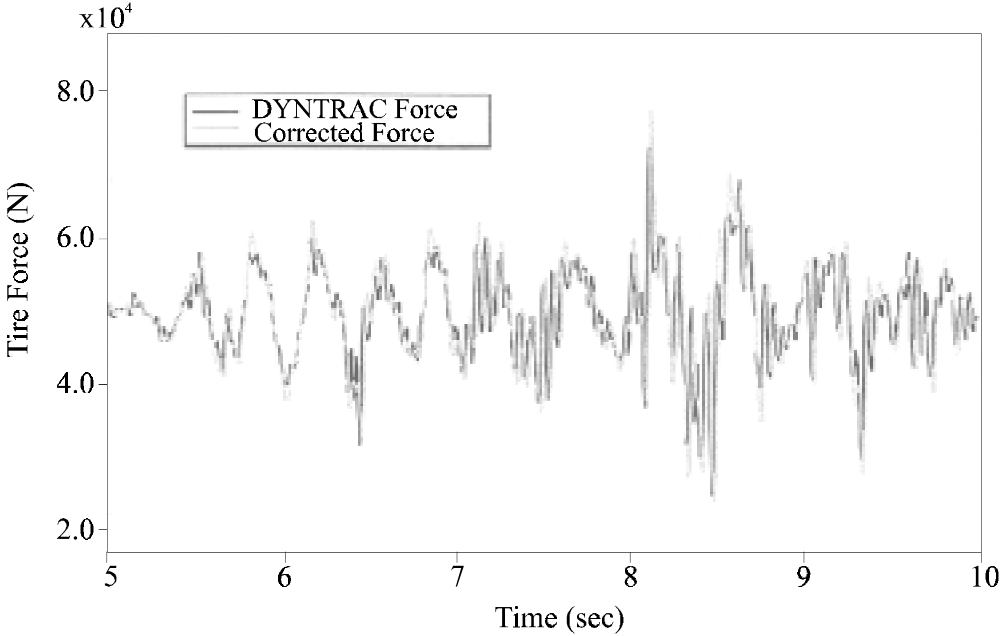

Figure 8. Measured and corrected tire forces.

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