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The Effect of Front and Rear Ride Rate Ratios to the Pitch Angle of a Midibus

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Abstract

Ride comfort and handling of a vehicle are relevant to front and rear ride rates. Consistency of comfort and handling come with the harmony of these two phenomenon. A vehicle has several reactions against the base excitations of irregular road inputs. Pitch motion is one of the most annoying reactions according to human perception. Since the road inputs excide the front wheels of the vehicle first, pitch motions occur even during small obstacle transitions. The effect of front and rear ride rate ratios to the pitch angle of a midibus is investigated in this paper. Several ratios are defined and a half vehicle model is generated. Modal analysis is performed, the natural frequencies of pitch modes are found. A pulse input is defined as base excitation in order to simulate road profile. Transient analyses are carried out and the vertical displacements of front end and rear end of the vehicle body are obtained. Pitch angles are calculated from the vertical displacements. Change in the pitch angle according to change in the ride rate ratios is demonstrated. Thus, an optimum ratio is determined that allows to form minimum pitch angle.

Keyword: Ride Rate, Pitch Angle, Modal Analysis, Half Vehicle Model

INTRODUCTION

Vehicles travel at various speeds, and are excited by irregular road inputs. Form of reactions, occurred due to these inputs can be judged as ride by passengers, and handling by drivers against the dynamic behaviors of the vehicle. Several studies have been carried out about the influence of the suspension parameters over ride and handling of the vehicles. Els et al. [1] mentioned the effect of spring-damper characteristics over ride and the inverse effect of the same characteristics over handling. The conflict caused by suspension stiffness and damping parameters was also described by Gobbi and Mastinu [2].

Ride of a vehicle can be examined by means of bounce and pitch motions. As the front wheels of the vehicle go over a road obstacle before the rear wheels, different form of bounce motions occur on front and rear end of the vehicle with respect to time delay. Delayed bounce motions shown in front and rear end, generate a pitch motion in the vehicle body. While time delay is related with the wheelbase and speed of the vehicle, the form of bounce is relevant with front and rear ride rates. Ride rate refers to the natural frequency of the vehicle in vertical bounce mode. Bounce and pitch reactions of a vehicle can be simply studied by half vehicle model. The simple mechanics of the quarter-car model do not fully represent the rigid-body motions that may occur on a motor vehicle. Because of the longitudinal distance between the axles, it is a multi-input system that responds with pitch motions as well as vertical bounce [3].

At the beginning of modern vehicle dynamics creation, experiments have been performed in test rigs by Olley [4]. He concluded his studies with a consideration that the front suspension should have a 30 % lower ride rate than the rear suspension. [5,6] To minimize pitch motion, the equivalent spring rate and the natural frequency of the front end should be slightly less than those of the rear end [7]. The rule that rear suspension should have a higher spring rate (higher natural frequency) is rationalized by the observation that vehicle bounce is less annoying as a ride motion than pitch [8]. Wenkui et al. [9] also mentioned the sensitivity of human body to pitch motion rather than vertical vibration.

As the vehicle traverses a road, the roughness excitation at the different axles is not independent. The rear wheels see nearly the same input profile as the front wheels, only delayed in time. The time delay is equal to the wheelbase divided by the speed of travel. The time delay acts to filter the bounce and pitch excitation amplitude, and has been called "wheelbase filtering" [10]. Sharp [11] also demonstrates the reactions of a vehicle changes by means of change in vehicle speed.

In this paper, several ride rate ratio cases that refer to an existing front ride rate and different rear ride rates are defined. Ratios between the front and the rear ride rates are calculated. Calculations started with three initial cases. For the first case, the front and the rear ride rates are equal. At the second case, the front ride rate is 30 % lower than the rear ride rate. The third case demonstrates a scenario that the rear ride rate is twice as the front ride rate. Modal analysis is performed and natural frequencies for pitch modes of the vehicle are specified for all cases via half vehicle model. A road excitation is introduced, and vehicle responses are examined for 10 m/s vehicle speed. Pitch angles of the vehicle body are calculated over vertical displacements of front and rear end of the vehicle. Results are examined for three cases, and the dynamic reaction of the vehicle is discussed against the change in front and rear ride rate ratios.

Dynamic Characteristics of the Vehicle

Half vehicle model

A four-degree-of freedom, half vehicle model is generated in order to examine bounce and pitch responses of the vehicle (Figure 1). Li [12] also defines this model as a 2-D bounce-pitch model. The general coordinates of the vehicle are Z_{fa}, Z_{ra}, Z and ϑ, while the inputs of the system are Z_{ft} and Z_{rt}. Primary suspension system, exists between the axles and the vehicle body is modeled by spring-damper couples. Unsprung masses are lumped to axle centers, and the vehicle body is modeled as a rigid beam. Damping property is also added to form a spring-damper couple while modeling the tires of the vehicle.

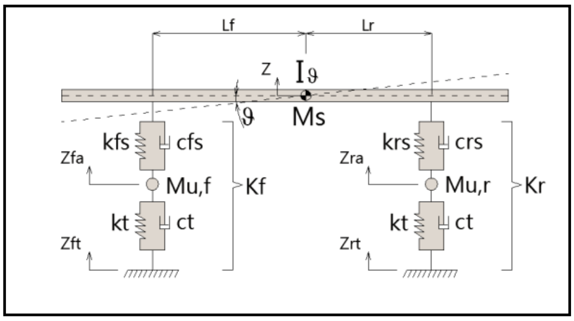

Figure 1. Half vehicle model

Bounce-Pitch dynamics of the vehicle

Bouncing and pitching effects play an increasingly significant role in a vehicle design [13]. The vehicle ride quality is strongly related to the pitch and vertical modes of the sprung masses [14]. On most vehicles there is a coupling of motions in the vertical and pitch directions, such that there are no pure bounce and pitch modes [8]. Bounce-pitch dynamics of a vehicle can be determined from the differential equations given below. Half vehicle model (Figure 1) can be used by simplifying the front and the rear suspension systems. Simplification can be done by assigning single stiffness to each suspension systems. Consider, K_f as front suspension stiffness, K_r as rear suspension stiffness, I_ϑ as pitch moment of inertia, k as radius of gyration. α, β, γ are the parameters given in Equations 1-4 in order to have a simple bounce and pitch motions of equations.

$$k = (I_\vartheta / M_s)^{1/2} \quad (1)$$
$$\alpha = (K_f + K_r) / M_s \quad (2)$$
$$\beta = (K_r L_r - K_f L_f) / M_s \quad (3)$$
$$\gamma = (K_f L_f^2 + K_r L_r^2) / M * k^2 \quad (4)$$

The differential equations for bounce (Z) and pitch (ϑ) can be described as given below.

$$\ddot{Z} + \alpha Z + \beta\theta = 0 \quad (5)$$
$$\ddot{\theta} + \beta Z / k^2 + \gamma\theta = 0 \quad (6)$$

Of the several coefficients in these equations, only β appears in both and is appropriately called the coupling coefficients [8]. As Gillespie [8] described, without damping, the solutions to the differential equations will be sinusoidal in form.

$$Z = Z \sin \omega t \quad (7)$$
$$\vartheta = \vartheta \sin \omega t \quad (8)$$

So, Equation 5 and Equation 6 will form, respectively;

$$- Z \omega^2 \sin \omega t + \alpha Z \sin \omega t + \beta \vartheta \sin \omega t = 0 \quad (9)$$
$$- \vartheta \omega^2 \sin \omega t + \beta Z / k^2 \sin \omega t + \gamma \vartheta \sin \omega t = 0 \quad (10)$$

From Equation 9;

$$Z / \vartheta = - \beta / (\alpha - \omega^2) \quad (11)$$

From Equation 10;

$$Z / \vartheta = - k^2 (\gamma - \omega^2) / \beta \quad (12)$$

From Equation 11 and Equation 12, we get;

$$(\alpha - \omega^2)(\gamma - \omega^2) = \beta (\beta / k^2) \quad (13)$$
$$\omega^4 - (\alpha + \gamma)\omega^2 + \alpha\gamma - \beta^2 / k^2 = 0 \quad (14)$$

The roots of Equations 14 will give the frequency of the pitch modes. Two of the roots are imaginary and can be ignored. The other two roots are given below;

$$\omega_1 = \sqrt{\frac{\alpha+\gamma}{2} + \sqrt{\frac{(\alpha-\gamma)^2}{4} + \beta^2 / k^2}} \quad (15)$$

$$\omega_2 = \sqrt{\frac{\alpha+\gamma}{2} - \sqrt{\frac{(\alpha-\gamma)^2}{4} + \beta^2 / k^2}} \quad (16)$$

Defining ride rate ratios

The study will be detailed by investigating three different cases that have different ride rate ratios (Table 1). Case 1 indicates the situation that the front and the rear ride rates are equal to each other. Case 2 demonstrates the Olley's [4,5,6] experiments that he strongly advocates his decision over 30 % stiffer rear ride rate than front ride rate. Case 3 represents the situation that the vehicle has a stiff rear suspension, which has a two times bigger ride rate than the front suspension. Front ride rate is the same in all cases for the comparison.

Table 1. Ride rate ratio cases

Case	Front Ride Rate (Hz)	Rear Ride Rate (Hz)	Ratio
1	1.557	1.557	1
2	1.557	2.024	1.3
3	1.557	3.114	2

Modal Analysis and Road Excitation

Modal analysis

Modal analysis is performed by considering half vehicle model (Figure 1). Natural frequencies of pitch modes are listed for all cases (Table 2).

Table 2. Natural frequencies of pitch modes

Case	Ratio	Pitch Mode-1 (Hz)	Pitch Mode-2 (Hz)
1	1	1.1371	1.3571
2	1.3	1.2027	1.5241
3	2	1.2285	1.8394

The vehicle has two pitch modes for each cases as given in Table 2. Both pitch modes have the natural frequencies close to the ride rates as indicated in the study of Hossain and Chowdhury [15]. While the first pitch mode has a limited increment with increasing ride rate ratio, the increment of the second pitch mode is more perceivable.

Road excitation

A rectangular pulse is applied as road excitation in order to trigger the pitch and bounce motion of the vehicle. The dimension of the pulse is 0.1 m in height, and 4.15 m in length (Figure 2). The vehicle is passing over the obstacle with a speed of 10 m/s.

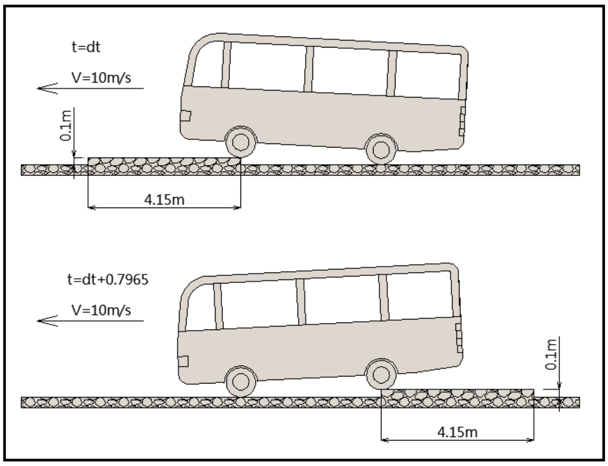

Figure 2. The dimension of the pulse

The first road excitation is applied to the front wheels at t=dt. The rear wheels are excited at t=dt+0.7965 due to the speed and wheelbase of the vehicle (Figure 2). The time step is defined as $dt=1/20/f_i$ and f_i is natural frequencies (Table 2).

Transient Analysis

Bounce displacements of front and rear end of the vehicle body

Transient analysis is performed for defined ride rate ratio cases (Table 1) by considering rectangular pulse road excitation (Figure 2). The displacement results are demonstrated for front and rear end of the vehicle body for all cases, respectively (Figure 3).

Figure 3 demonstrates the bounce motion belongs to front and rear end of the vehicle body. The maximum displacement values are also listed below (Table 3).

Results given in Table 3 demonstrate that maximum displacement occurs in Case 2 for both upward and downward motions. As shown in Figure 3, the period of time is longer in Case 1 than Case 2, and Case 3.

Pitch motion can be calculated from the bounce displacements of front and rear end of the vehicle (Figure 4).

Pitch angle "ϑ" can be calculated as given in Equation 17. Pitch angle results are shown below for all cases (Figure 5).

$$\vartheta = (Zf - Zr) / (\text{Vehicle Length}) \quad (17)$$

From Figure 5, the results show that the highest amplitude occurs in Case 2. However, the amplitude of pitch angles are nearly the same for Case 2 and Case 3, Case 3 has the lower values in peak points, and also the motion in Case 3 is damped quicker than Case 2. Case 1 has the longest period of all cases.

Since the excitation ends at dt+07965, it means that the vehicle is on the starting groundline from that time, the comparison can be made from that time to the end of the motion in order to see the effect of ride ratio cases to the pitch angle.

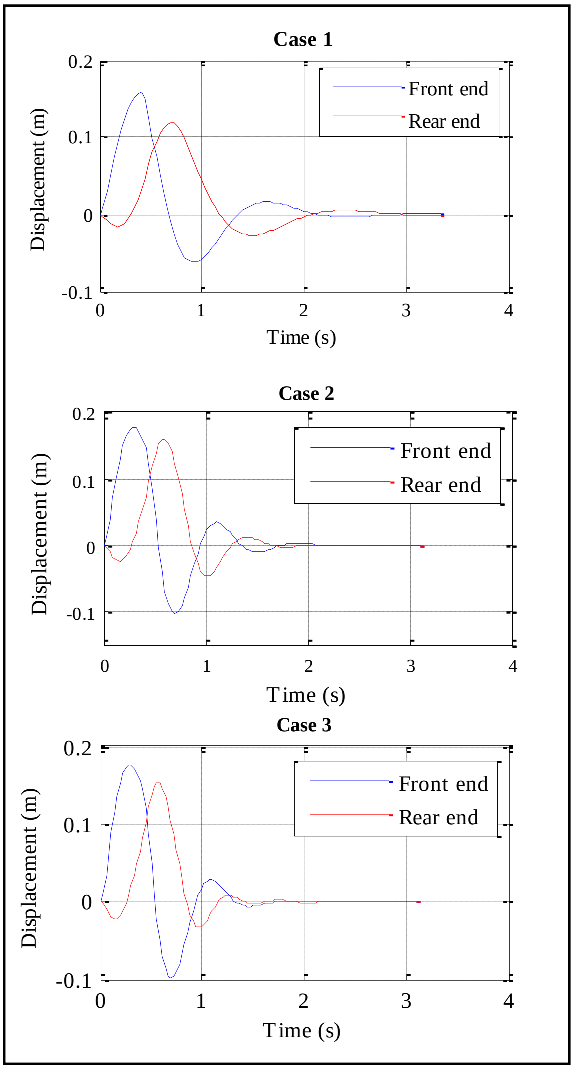

Figure 3. Displacement results of all cases

Table 3. Maximum-minimum displacement values

Case	Max Displacement Values [Upwards/Downwards] (m)	
	Front End	Rear End
1	+0.158 / -0.062	+0.119 / -0.027
2	+0.178 / -0.105	+0.159 / -0.047
3	+0.176 / -0.099	+0.153 / -0.034

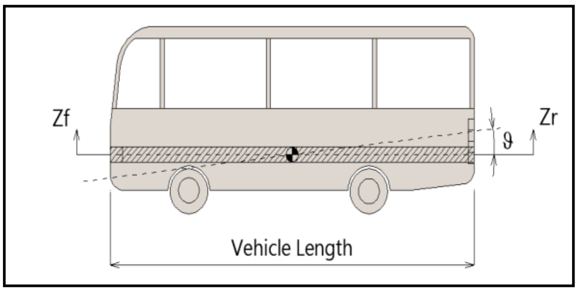

Figure 4. Pitch angle "ϑ"

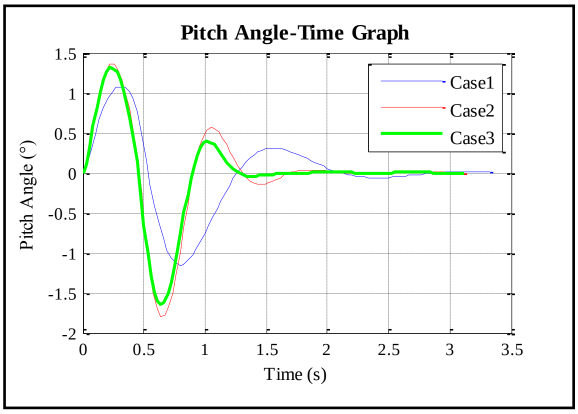

Figure 5. Pitch angle-time graph

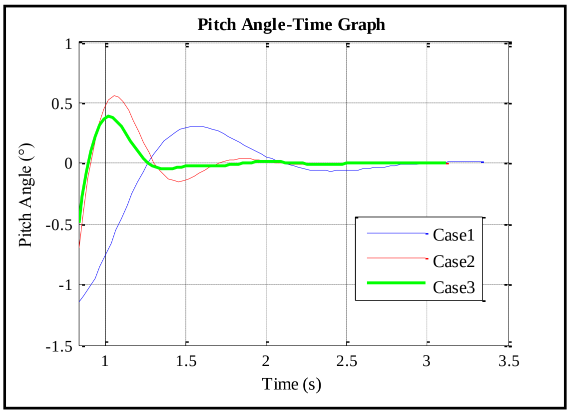

Figure 6. Pitch angle-time graph after the excitation finishes

As shown in Figure 6, the maximum pitch angle occurs in Case 1. Case 3 has the minimum amplitudes among the results, and also the pitch motion in Case 3 is damped most quickly.

Maximum pitch angle values during excitation and after excitation are given in below table (Table 4).

Table 4. Maximum pitch angle values

Case	Maximum Pitch Angle (°)	
	During Excitation	After Excitation
1	1.157	1.14
2	1.795	0.70
3	1.652	0.50

The results shows that pitch angle increases with increasing ride rate ratio during the road excitation. Bounce displacement also increases as the reaction of vehicle body against road excitation for increasing suspension stiffness. Case 2 and Case 3 are the examples of this situation as their stiff rear suspensions resist against the deflection caused by load transfer of inclined body, so the vehicle body will bounce in front and rear end against the inability of suspension elements absorbing the excitation forces transmitted from road irregularities. As a result, the high degrees of pitch angles are shown during road excitations. However, the situation is just the opposite as the road excitation finishes. Stiff rear suspension seems more successful in decreasing the period of pitch motion and causing the vehicle to be stable quicker.

As Case 1 is assumed, representing the soft suspension characteristics for this study, the bounce of the vehicle body is less than other cases due to the allowance of softer rear suspension for deflection against the load transfer. The excitation forces are absorbed by the suspension elements, and the bounce of the vehicle body is limited. On the other hand, when excitation finishes, the equality of the front and the rear suspension ride rates are insufficient for damping the pitch motion.

CONCLUDING REMARKS

In this paper, a study is performed by investigating how the ratios of the front and the rear ride rates affect the pitch angle of a midibus. Half vehicle pitch-bounce model is generated. Three rear ride rates corresponding to three cases are defined by assuming the front ride rate is the same in all cases for the comparison.

Modal analysis is performed, and pitch modes are indicated of all cases. The results show that each cases have two natural frequencies in pitch mode. While the first pitch mode has a limited increment with increasing ride rate ratio, the increment of the second pitch mode is more perceivable.

Transient analysis is performed applying a rectangular pulse excitation to the vehicle. Bounce displacements of front and rear ends of the vehicle are obtained, and pitch angle-time graphs are calculated from the bounce displacements.

The dynamic results indicate that, stiffer rear suspension causes an increment in bounce displacements during the excitation and also an increment in pitch angle. The highest bounce displacements and pitch angle values occur in the case that rear suspension ride rate is 30 % bigger than front suspension ride rate (Case 2).

As the ride rate ratio is increased, pitch angle of the vehicle body increases, but the period of pitch motion decreases. The damping capability of the vehicle is higher against the pitch motion when the ride rate ratio is higher.

The situation of equal front and rear suspension ride rates demonstrate a low frequency ride from bounce point of view, on the other hand inadequate damping characteristics against the pitch motion may lead a motion sickness for passengers during repetitive road irregularities.

This study indicates the positive effect over damping the pitch angle of a vehicle excited by a rectangular pulse, as the rear suspension ride rate is higher than front suspension ride rate.

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