

Vibration Characteristics Analysis in Vehicle Body Vertical Plane of Motorcycle during Turning

Yoshitaka TEZUKA*
Yoshinobu SHIOMI*

Shinichi KOKUBU*
Satoru KIYOTA*

ABSTRACT

A vibration phenomenon generated in the vertical plane of a motorcycle during high-speed turning has been analyzed using actual running tests and simulation. As a result, it was found that the phenomenon is a self-excited vibration caused by the sprung part resonance and the unsprung part resonance of the rear wheel coupled with a frictional force variation at the contact patch of the tire. The simulation model developed reproduced the same vibration phenomenon.

1. Introduction

Motorcycles sometimes generate a vibration during high-speed turns. The dominant component of the vibration is in the vertical plane of the motorcycle. Riders have complained that it is uncomfortable and that it slows down the turning velocity or interferes with other active performances. Few studies of this vibration have been carried out; currently, it is handled by adjusting the motorcycle dimensions, in accordance with experience and the rider's preferences, by adjustment of the suspension parameters, by choice of tires, and by other measures.

The authors have conducted actual running tests under conditions which are known to cause the vibrations. The test results allowed an investigation of the mechanisms underlying the vibration and creation of a simulation model for reproducing similar vibrations.

2. Measurement Method for Actual Running Tests

As depicted in Fig. 1, 14 accelerometers were mounted on the test motorcycle, 7 in the sprung portions and 7 in the unsprung portions to obtain rigid-body behavior of the motorcycle. Motorcycle conditions were simultaneously observed: Engine speed, throttle opening, rotation velocity of the front and rear wheels, motorcycle body roll angle, and lap marker (in order to grasp which area of the course the motorcycle is being run). These were transmitted over a data

link for measuring vibration.

The vibratory phenomenon does not derive until the motorcycle turns at a limit causing lateral tire slip, so the measurement device was installed inside the fuel tank. This was due to considerations of its effect on motorcycle body conditions including additional weight to the motorcycle and the change in weight distribution on the wheels, and reducing the inconvenience to the rider.

3. Actual Running Test Results and Analysis

3.1. Analysis of Time Domain Data

Figure 2 shows the time domain waveforms for front axle vertical acceleration, engine speed, velocity difference between the front and rear wheels, and the throttle opening during the vibration event observed in the running test.

The vibration was prominent over the period (for about 2 seconds) as shown in the figure. There is a visible increase in the vertical acceleration trace amplitude for the front axle in comparison with time before and afterward. The throttle was opened just before the event began, and when the throttle was closed slightly, the vibration converged toward normal levels. Then, the vibration began again as the throttle was re-opened. Rear wheel slip was visible as the difference in wheel velocities appeared in sequence with the throttle opening changes. The motorcycle body roll angle has been nearly constant throughout this period.

* Asaka R&D Center

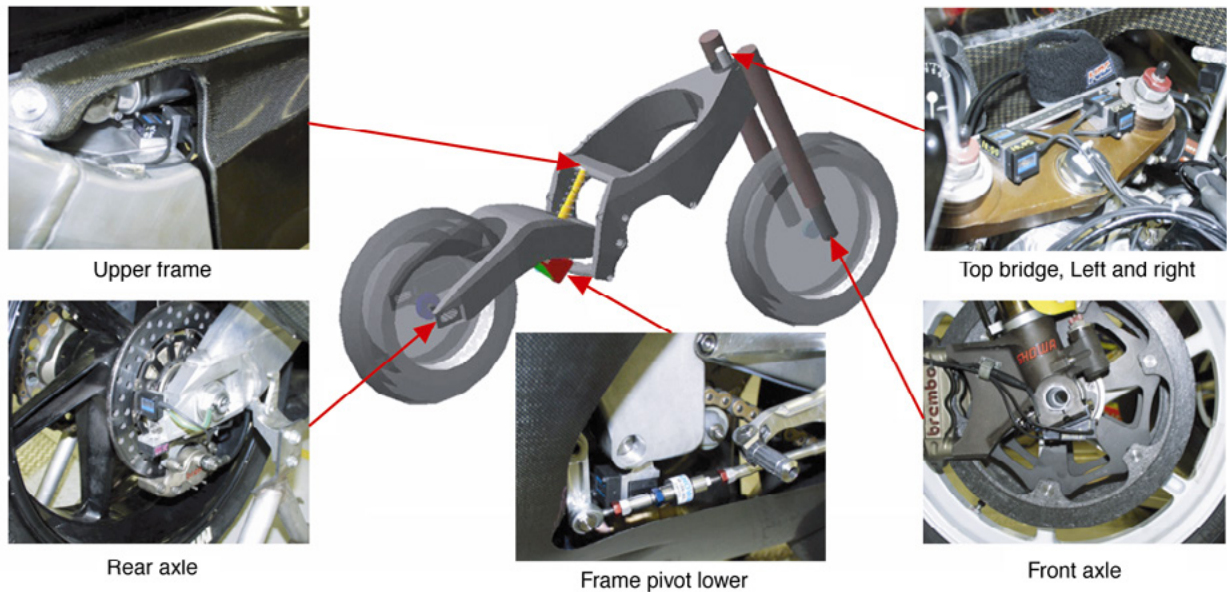


Fig. 1 Locations for acceleration meter

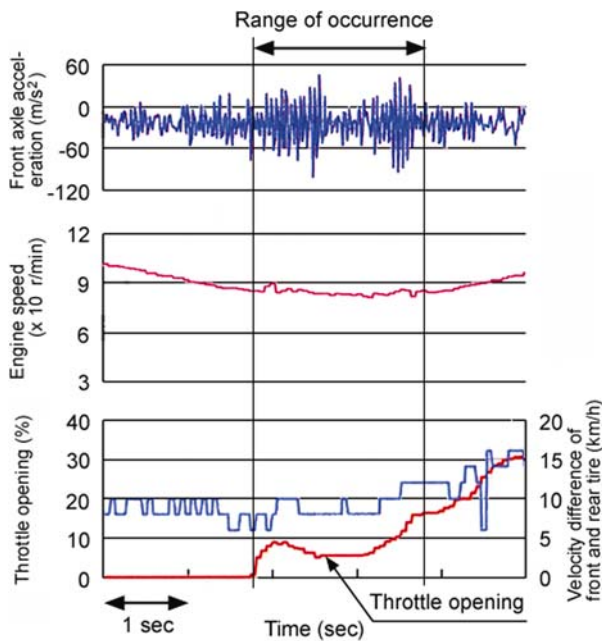


Fig. 2 Time domain data

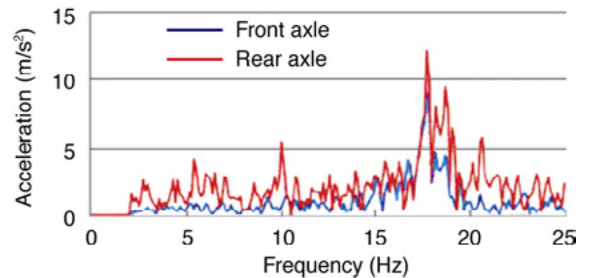


Fig. 3 Front and rear axle acceleration

wheel. It is assumed that the sprung resonance of the body accompanying the vertical motion of the front wheel couples with the resonance in vertical motion of the rear wheel, and the two modes excite each other; the combination appears to act as a self-excited system with a continuous energy input.

Close consideration of the time domain data in the running tests and the analysis of the vibration modes indicated that these are self-excited vibrations that occur during the steepest body roll angle with the driving force applied to the rear wheel, and that mutually promote the behavior of the resonance body modes.

3.2. Analysis of Real Vibratory Motion

A frequency analysis of the data observed during the running test showed vibrations occurring around a prominent peak at 17.7 Hz, as shown in the power spectrum density (PSD) of the vertical acceleration for the front and rear axles in Fig. 3. A motion animation of the 17.7 Hz vibration was made, using a transfer function in reference to the vertical acceleration of the rear wheel (see Fig. 4).

Observations of the actual vibration motion showed that it could be decomposed into three motions, in a continually repeated sequence. Motion 1: The entire body moves forward, forcing the front wheel against the ground. Motion 2: The body moves back, forcing the rear wheel against the ground. Motion 3: The front wheel moves upward, followed after a slight delay by the rise of the rear

4. Models Used for Simulation

4.1. Motorcycle Body Model

A spring and rigid-body mass model with nine degrees of freedom (DOF) was constructed, on the basis of the authors' deductions about the frequency of the actual vibration that the chassis, front fork, and swing arm accompany no elastic behavior.

Figure 5 shows the model used for the simulation. It was composed of ten elements: Sprung steering components, unsprung steering components, front tire, engine, frame body, rider body, rear suspension, and its linkage portion, swing arm, and rear tire. The nine DOF consisted of six DOF for the body component, one DOF for steering and two DOF for

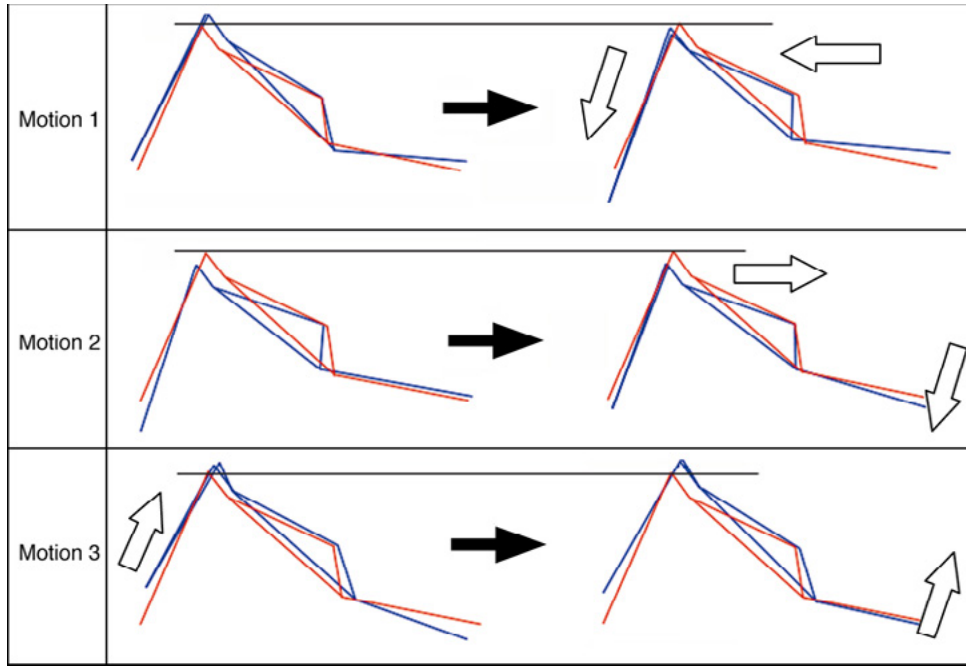


Fig. 4 Sample of modes animation

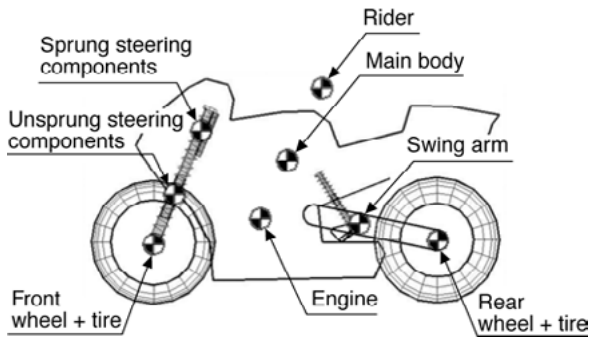


Fig. 5 Motorcycle model

the front and rear suspensions. The actual spring and damping parameters for suspension are non-linear, but since the vibration occurred while the motorcycle was in a stable configuration, these were approximated with linearized parameters. The motorcycle body was assumed rigidly bound, including the rider, who was considered only as a single mass.

4.2. Tire Model

The vibration is caused by a frictional interaction between the tires and the road surface, so modeling of the tires was an essential part of this simulation. Since this model was linearized, however, the tire characteristics do not express dynamic attitude such as cornering characteristics and defines spring constant and damping factor of the tire from the tire tester. The roll angle is high in this model and the motorcycle is in the midst of cornering, so frictional forces act at the contact patch between the tire and the ground in the centripetal direction. Analysis of the time domain data showed evidence of the driving force at the rear tire, in addition to the centripetal force, during the period when the vibration was occurring. The frictional force F_μ was defined as shown in Fig. 6⁽¹⁾⁻⁽³⁾.

$$F_\mu = \mu F_z \quad (1)$$

$$F_z = K_g u_z \quad (2)$$

Here,

μ : Friction coefficient

u_z : Vertical displacement of tire

Figure 7 shows the definitions of the vertical direction spring constant K_g , the spring constants of the tires in terms of the vehicle coordinate system Y, Z axes K_y , K_z , and the camber angle θ . These are required to solve Eq. (3):

$$K_g = K_z \cos \theta + K_y \sin \theta \quad (3)$$

4.3. Equation of Motion for Motorcycle Body Model

Since the vibration was attributed to self-excitation by the analysis of the running test results, a stability criterion analysis was carried out using complex eigenvalues analysis to investigate the potential for reproduction of the actual phenomenon^{(4), (5)}.

In the first stage of a complex eigenvalue modal analysis, the real parts of the eigenvalues are sought, assuming no damping or exterior loads on the system. In the second stage, these real eigenmodes are used to transform the asymmetric equation of motion in the physical coordinate system into modal coordinate system and execute complex eigenvalue analysis. The complex eigenmode thus obtained expresses the extent to which each real mode is excited. By showing the displacement components of the complex modes as vectors, it is easily seen how much each real-component mode contributes to the unstable mode⁽⁶⁾⁻⁽⁹⁾. The equation of motion for the body model is given in Eq. (4):

$$[M]\{\ddot{x}\} + [K]\{x\} = \{F\} + \{F_\mu\} \quad (4)$$

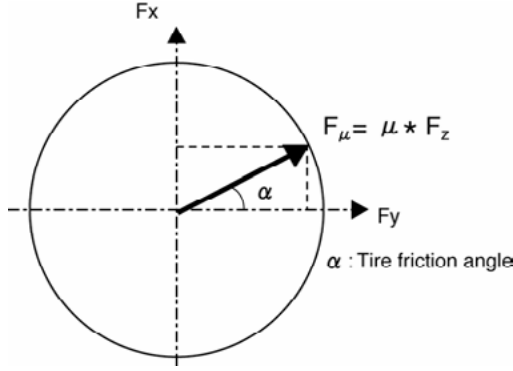


Fig. 6 Tire friction circle concept

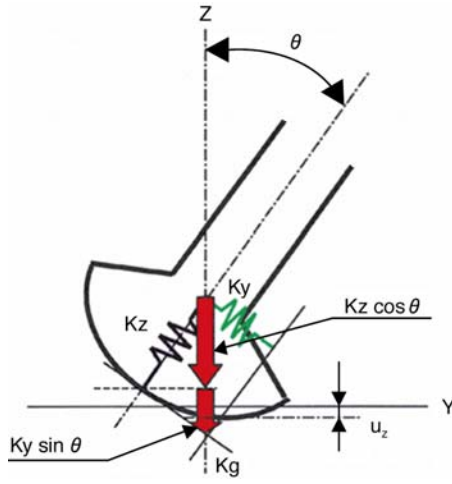


Fig. 7 Tire condition

Here,

$\{F\}$: Force vector
 $[M]$: Mass matrix
 $[K]$: Stiffness matrix
 $\{x\}$: Displacement vector

Exterior forces due to friction $\{F_\mu\}$ are calculated using Eq. (5):

$$\{F_\mu\} = [\Delta K]\{x\} \quad (5)$$

$[\Delta K]$ is an asymmetric modal stiffness matrix since $\mu * k_g$ expressing the frictional force exerted on the tire, contains a row with the Y-direction component and a column with the Z-direction component only in the respective body-coordinate directions.

$$[\Delta K] = \begin{bmatrix} & \mu * k_g \\ & \end{bmatrix} \quad (6)$$

Substituting Eq. (5) into Eq. (4) and moving the force term which is a function of position onto the left side, the asymmetric equation of motion in Eq. (7) is obtained:

$$[M]\{\ddot{x}\} + [K - \Delta K]\{x\} = \{F\} \quad (7)$$

If we examine the real eigenmodes of a symmetric version of Eq. (7), with the asymmetric stiffness term $[\Delta K]$ removed, we obtain the relational expression between the physical and modal DOF in Eq. (8).

$$\{x\} = [\phi]\{\lambda\} \quad (8)$$

Here,

$[\phi]$: Mode shape matrix
 $\{\lambda\}$: Modal vector

Substituting Eq. (7) into Eq. (8) and multiplying both sides by the transposed matrix $[\phi]^T$, Eq. (9) is obtained:

$$[m]\{\ddot{\lambda}\} + [k - \Delta k]\{\lambda\} = \{f\} \quad (9)$$

This equation uses the terms

$[m]$: Modal mass matrix
 $[k]$: Modal stiffness matrix
 $[\Delta k]$: Asymmetric modal stiffness matrix
 $\{f\}$: Modal force

where each term is composed of the following:

$$[m] = [\phi]^T [M] [\phi] \quad (10)$$

$$[k] = [\phi]^T [K] [\phi] \quad (11)$$

$$[\Delta k] = [\phi]^T [\Delta K] [\phi] \quad (12)$$

$$\{f\} = [\phi]^T \{F\} \quad (13)$$

To add a little further explanation of Eq. (9): Complex eigenvalue analysis reveals complex eigenmodes, which consist of the complex eigenvalue and the real eigenmodes modal DOF for the model. This allows the real eigenmodes which cause the unstable mode(s) to be identified.

5. Results of Simulation and Observations

A parametric study was conducted in an attempt to identify the variables which had correlations with the vibration phenomenon in order to assess the validity of this model. Since the vibration occurred under a combination of circumstances of near-the steepest roll angle, driving force applying to the rear wheel and low throttle opening of around 10%, the body roll angle and the angle α which establishes the frictional force F_μ under driving force at high body roll angles (hereafter called the “tire friction angle”) were chosen as the parameters. The body roll angle was varied in steps of 5° from 30° to 60° and the tire friction angle was varied from 0° to 10° for each body roll angle.

Figure 8 shows the root loci of the parameter study results. The real part of the complex eigenvalues becomes unstable around the roll angle of 45° and the tire friction angle of 8°. This clearly was the region in which the self-excitation began. The resonance frequency was indicated at around 19 Hz here, a value reasonably close to that observed in the running test. No vibration was indicated at lower body roll angles, regardless of the tire friction angle; the real part of the complex results was stable. These results vindicate the model as valid for reproducing the vibration.

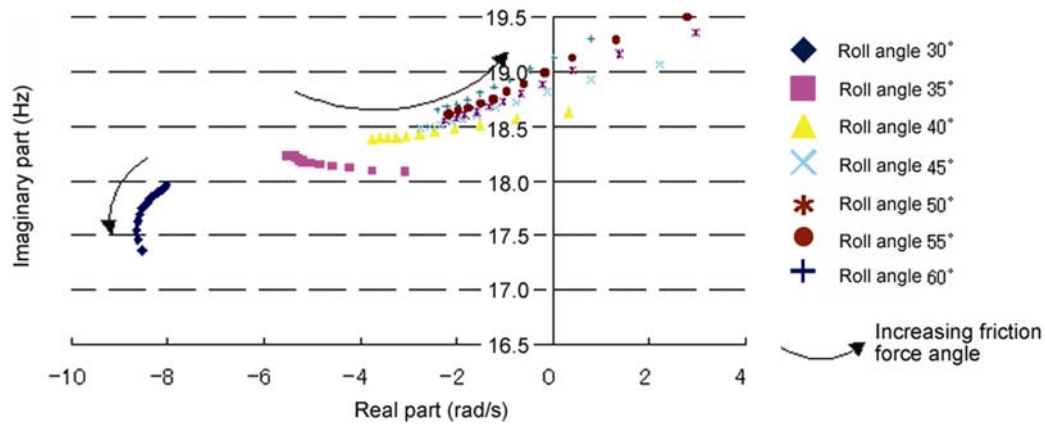


Fig. 8 Simulation result

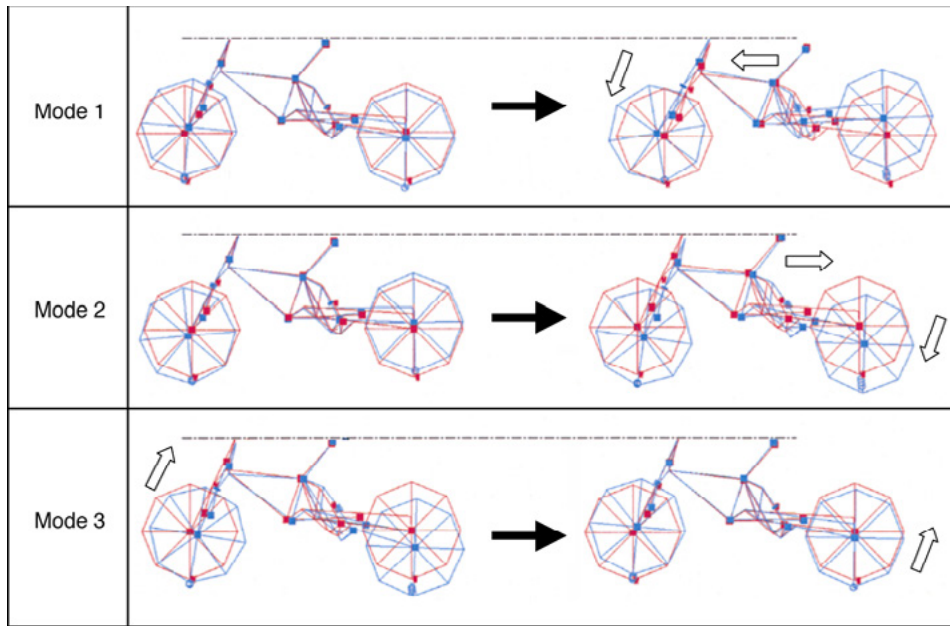


Fig. 9 Sample of simulation result animation

Next, the causes of the vibration were analyzed.

The modal complex eigenvalue analysis allows quantitative identification of the contribution ratios of the real eigenvalues, which cause the unstable eigenmodes. Figure 9 shows the results of a simulation at the roll angle of 50° and the tire friction angle of 8°. These are nearly the same as the vibration modes seen in the running test, and the resonance frequency was 19 Hz. The components of the complex eigenmode displacements were obtained by the analysis. It is indicated that the vibratory phenomenon is mainly made up from three real eigenmodes A, B and C, which are shown as vectors in Fig. 10.

Table 1 provides these real eigenmodes and their strain energy ratios. Real eigenmode A was the opposite phase mode of the sprung components on both the front and rear wheels; the strain energy chart indicates a high contribution from the front suspension. Mode B, the opposite phase mode of the unsprung components on both the front and rear wheels, showed a dominant contribution from the rear tire. Mode C, the in-phase mode of the unsprung components on both wheels, showed a dominant contribution from the front tire. The vector relations depicted in Fig. 10 show that

Modes B and C were nearly equal in magnitude and were mixed. Since they were out of phase with each other and chiefly affected the front tire, the two phases cancelled, reducing displacement of the front tire. They appear to have formed a synthetic mode of the unsprung components of the

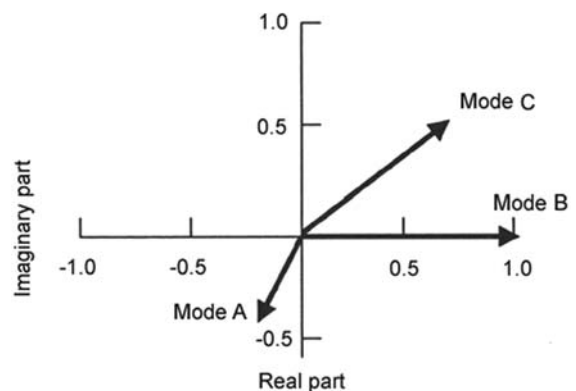
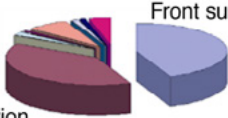
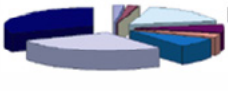


Fig. 10 Mode vector

Table 1 Real inheritance mode of simulation model

	Modes	Percent of strain energy
A	 Opposite phase mode of sprung	 Front suspension 44% Rear suspension 39%
B	 Opposite phase mode of unsprung	 Rear tire 68% Front tire 28%
C	 Inphase mode of unsprung	 Rear tire 28% Front tire 70%

rear tire. Thus, the vibratory phenomenon appears to have been caused by friction at the tires, resulting in coupling of the sprung components' Mode A and a combination of the unsprung components' Modes B and C.

6. Conclusion

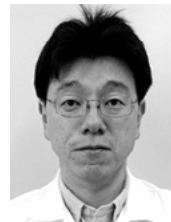
- (1) It was found that the motorcycle vertical-plane vibration observed in a motorcycle during high-speed turns was the self-excited vibration, in which the opposite phase mode of sprung components on the vehicle body and the mode of unsprung components on the rear wheel excited each other by friction at the tires' contact patches.
- (2) Actual running tests and simulation provided results which revealed the mechanisms of the vibration and enabled reproduction of the vibratory phenomenon in a simulation.

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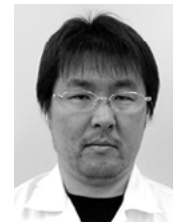
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■ Authors ■



Yoshitaka TEZUKA



Shinichi KOKUBU



Yoshinobu SHIOMI



Satoru KIYOTA