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The Chatter of Racing Motorcycles

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SUMMARY

The chatter of motorcycles appears during braking and consists of a vibration of the rear and front unsprung masses at a frequency in the range of 17-22 Hz depending on the motorcycle. This vibration could be very strong and acceleration of the unsprung masses can reach 5-10 g. The chatter is an auto-excited vibration and this fact explains why it appears suddenly when the mechanism of auto-excitation is generated. This paper presents the chatter phenomenon both from experimental and numerical point of view. First, the chatter is defined on the basis of some experimental data from racing motorcycles and from the comments of some racing teams technicians. Then, chatter is analyzed in different motion conditions and for different braking styles by means of linear and non-linear simulations of the motorcycle dynamics. A physical interpretation of the phenomenon is also proposed.

Keywords: motorcycle, chatter, experiments, vibration, multibody

1. INTRODUCTION

Over the past ten years a vibratory phenomenon appears nearly exclusively in racing motorcycles and only on some tracks and in some kinds of maneuvers. Such vibration consists of vertical oscillations of the unsprung masses with frequency around 17-22 Hz. The suspensions' shock absorbers are not able to dampen these vibrations; in the presence of chattering, riding the motorcycle near limit conditions becomes more difficult. Trying to reduce these negative effects, the technicians adopt remedies based on their experience and on the information brought back from the riders. Regarding the chattering phenomenon the racing technicians have suggested various evocative interpretations and several possible empirical solutions. Some more scientific definitions are the following:

"The Chattering is a recurrent problem in the last years exasperated from the high power available. The riders perceive the chattering when they close the throttle. The chattering starts when the braking phase is off, with closed gas and before the phase of acceleration. It is possible to attenuate the chattering vibrations rendering the system less sensible to the typical frequencies of the Chattering of 15-20 Hz, by means of the variation of the unsprung masses and/or the stiffnesses of the structural components of the vehicle" [1].

"Chatter is a phenomenon well known, and well feared, amongst the racing fraternity. When chatter occurs the rider feels a vibration, which reduces his sense of feedback from the tires. This usually results in a drop in confidence and a poor lap time. The vibration is actually "wheel hop", a resonance of one or both of the unsprung masses. So what can be done about this problem, from a tire perspective? It helps to know what might be causing or driving the vibration. It is often observed that the wheel rotation frequency is close to the chatter frequency, which suggests a tire non-uniformity; force variation or run-out, or maybe imbalance. This would normally occur in the mid-corner. The solution on race day may be as simple as changing the tire for a new one, but the general solution for the tire manufacturer is to strive for ever better tire uniformity. It is also often observed that chatter occurs when the tire(s) is operating near its grip limit. The driving mechanism here seems to be the stick-slip sliding, which will load and unload the suspension; a bump in the track might have a similar effect. In this instance a change in grip level (up or down) can often fix the problem, which can be achieved by something as simple as changing tread compound. Other parameters which can affect the situation are tire mass and stiffness, which fundamentally change the characteristics of the spring mass damper system, the idea being to shift the wheel hop frequency out of the problem range. This requires significant changes in mass/stiffness, which in turn require changes in bike setup to maintain performance. This is a difficult approach more suited to a development environment than the quick fix situation" [2].

"Although tire or wheel chatter is not an "every day problem" in modern GP racing, it is something we need to keep always in mind or with other words, it is a situation we need to be concerned about every day. When it happens, it affects rider's feel (feedback) for the bike as well as tire grip and even tire endurance may go down. Typically chatter happens at oscillating frequencies around 20 Hz (normally a little below 20 Hz). In our observation this frequency goes very well together with the rotational frequencies of the corresponding wheels. Even experienced GP riders do feel chatter most of the time first in the front, while it usually starts in the rear (I would say perhaps more than 90% of all cases) and then very quickly transfers to the front (due to geometry and stiffness balance of those motorcycles). Typical areas to experience chatter are braking (not maximum straight brake, but lean angle braking), corner entry and the rolling phase through the corner apex. On acceleration then the wheels stop chattering quite quickly. Therefore as far as lap time performance in motorcycle racing is concerned, excessive chatter damages mainly the phase from corner entry (including "lean angle braking"), corner speed itself and the initial acceleration or delays the point where the rider can start to accelerate" [3].

This short review demonstrates the difficulty in understanding the chattering phenomenon, even among the technicians of the world-wide motorcycle racing. On this topic few scientific papers have been published perhaps because of the difficulty of acquiring experimental results and perhaps because the phenomenon interests only racing vehicles. In 1996 the authors presented a study on the auto-excited vibrations of motorcycles which attempted to interpret a vibration phenomenon observed in the rear suspension of some racing vehicles [4] (in Italian). The cause of the vibrations was attributed to the fluctuation of the tire longitudinal force due to the vertical oscillations of the rear wheel. The fluctuating force component can be, in some conditions, in phase with the fluctuating velocity of the contact point and therefore increases the energy of the oscillations.

An extended series of experimental results, collected during each of the world championship races in year 2000 for the 125cc class, have been presented in reference [5] (in Italian). The analysis of the data shows that the chattering frequencies are around 20 Hz near to the resonance frequencies of both the rear and front unsprung masses. The chattering mainly appears during the braking maneuver and fades during the phase of acceleration. This work identifies the cause of the chattering in the combined fluctuations of the tire driving force, the vertical tire load, and the swingarm attitude.

In 2004 Tezuka et al. [6] analyzed the in-plane vibrations of a motorcycle during high speed turning using both actual tests and simulations. As a results, he found that the self-excitation is due to the rear sprung and unsprung mass resonance coupled with the rear tire frictional force variation.

This paper is organized as follows. In section 2 some examples of chatter phenomenon are briefly described; section 3 illustrates the mathematical model used for chatter simulations; section 4 shows the stability analysis via eigenvalues of a 125cc racing motorcycle during braking, whereas section 5 presents non-linear simulations of chatter.

2. EXPERIMENTAL CHATTERING EXAMPLES

In Figure 1 and Figure 2 some telemetric data are presented of a racing motorcycle braking from 50 to 25 m/s with a deceleration of about 7 m/s^2 (Figure 1a). These data show a chatter phenomenon at about 19 Hz which leads to rear wheel vertical acceleration up to 9 g (Figure 1c). Moreover the chatter starts at the rear wheel and then moves to the front wheel (Figure 1c-d). Figure 1b shows the compression of the front fork (up to 130 mm at 45.7 s) and the extension of the rear suspension (up to 0 mm at 45 s). The compression of the rear suspension after 46 s is mainly due to motorcycle entry in curve, i.e. to the in-plane component of centrifugal acceleration. The presence of chatter may also be noticed from analysis of suspensions travel, but vertical acceleration is a better gauge. Finally Figure 2 shows a typical feature of the chatter: the engine fluctuation (of a few hundred rpm's) is opposite in phase to rear wheel oscillation.

[fig1]
[fig2]

3. MATHEMATICAL MODEL

The mathematical model is an updated version of the eleven degree of freedom model described in [7]. It consists of a system of seven bodies: rear assembly (chassis and rider), front assembly (handlebar and front sprung mass), rear and front unsprung mass, rear and front wheel and a body including the engine-transmission inertia properties, which is connected to the rear wheel by means of a spring-damper system taking into account the transmission compliance. Tire dynamics is modeled with an advanced model which takes into account both the carcass compliance and geometry (for more detail see reference [8]).

[fig3]

The rear tire longitudinal slip is defined as the ratio between the horizontal component of the Eulerian velocity of the tire contact point P and the horizontal component of its absolute velocity (i.e. horizontal velocity of the rear wheel center):

$$\kappa = -\frac{VP_{eul}}{VP_{\mathrm{abs}}}$$

which may be rewritten as

$$\kappa = -1 - \frac{(R+\zeta_R)\left(\omega + \left(\frac{\partial}{\partial t}\xi\right) + \left(\frac{\partial}{\partial t}\theta\right)\right)}{\sin(\Phi+\Phi_0)\left(\frac{\partial}{\partial t}\Phi\right)L_{sw} + \left(\frac{\partial}{\partial t}x\right) + V} \tag{1}$$

where (angle and spin rate are positive if counter-clockwise, as shown in Figure 3)

R	[m]	is the rear tire steady state rolling radius
ζ_R	[m]	is the rear tire radial deformation (negative in compression)
$\frac{\partial}{\partial t}\xi$	[rad/s]	is the rear tire tangential rate of deformation

ω	[rad/s]	is the rear tire steady state spin rate
$\frac{\partial}{\partial t}\theta$	[rad/s]	is the rear wheel spin rate fluctuation
Φ	[rad]	is the swingarm angle fluctuation with respect to ground
Φ_0	[rad]	is the swingarm steady state angle with respect to ground
L_{sw}	[m]	is the swingarm length
$\frac{\partial}{\partial t}x$	[m/s]	is the fluctuation of swingarm pivot longitudinal velocity
V	[m/s]	is the steady state swingarm pivot longitudinal velocity

Note that the effective rolling radius R_e corresponds to the loaded radius in steady state conditions R plus the infinitesimal fluctuation ξ_R of the radial deformation.
The linearization of the equation (1) around the zero-slip condition with respect the fluctuating variables makes it possible to underline the four terms generating the rear tire longitudinal slip fluctuations:

$$\kappa_L = \frac{\left(\frac{\partial}{\partial t}\theta\right) + \left(\frac{\partial}{\partial t}\xi\right)}{\omega} + \frac{\sin(\Phi_0)\, L_{sw}\left(\frac{\partial}{\partial t}\Phi\right)}{\omega R} + \frac{\frac{\partial}{\partial t}x}{\omega R} + \frac{\zeta_R}{R} \tag{2}$$

Moreover, the rear wheel spin rate fluctuation depends on the chain transmission kinematics (see Figure 3) by means of the following relationship:

$$\frac{\partial}{\partial t}\theta = -\left(\frac{\partial}{\partial t}\Delta\right) + \frac{\left(\frac{\partial}{\partial t}\beta\right) rp - \left(\frac{\partial}{\partial t}L_{ch}\right)}{rc} \tag{3}$$

where

$\frac{\partial}{\partial t}\Delta$	[rad/s]	is the deflection rate of the sprocket isolator, which is usually positioned between chain sprocket and rear wheel
$\frac{\partial}{\partial t}\beta$	[rad/s]	is the drive sprocket spin rate
r_c	[m]	is the radius of the rear wheel sprocket
r_p	[m]	is the radius of the drive sprocket
L_{ch}	[m]	is the chain 'free-length' as shown in Figure 3

$$L_{ch} = \sqrt{L_{sw}^2 + R_z^2 + R_x^2 + 2\,R_z \sin(\upsilon)\,L_{sw} + 2\,R_x \cos(\upsilon)\,L_{sw} - (rc - rp)^2}$$

$$\frac{\partial}{\partial t} L_{ch} = -\frac{\left(\frac{\partial}{\partial t}\upsilon\right) L_{sw}\,(R_x \sin(\upsilon) - R_z \cos(\upsilon))}{L_{ch}} \tag{4}$$

where

$\frac{\partial}{\partial t}\upsilon$	[rad/s]	is the relative angular velocity between chassis and swingarm
R_x, R_z	[m]	are the coordinates of the drive sprocket relative to the chassis

Therefore the slip is a function of seven independent variables ($x, \upsilon, \Phi, \beta, \Delta, \xi, \zeta_R$), whereas the system may be described by eight dependent variables (vertical displacement of the swingarm pinion has to be added to the seven mentioned variables) and one constraint (tire contact point must lie on the ground). The effect of each of the previously introduced variables are divided into four groups (see equation (2)) and depicted in Figure 4. Figure 4a shows the effect of rear tire radial compliance ζ_R and Figure 4b the effect of swingarm motion Φ. From the physical point of view these two effects are inherently connected, and usually a tire radial fluctuation involves some swingarm oscillation. Figure 4c shows the effect of longitudinal swingarm pivot velocity $\dot{x}$ and Figure 4d the effect of the drive sprocket rotation β, plus sprocket isolator deflection Δ, plus effects due to suspension kinematics $L_{ch}(\upsilon)$ and tire tangential compliance ξ. Every vibration around the equilibrium position leads to a slip fluctuation.

[fig4]

Although each rider has his/her own braking style, from a dynamic point of view there are four different braking strategies: rear brake only, engine brake only, front brake only, and combined braking. In the case of rear brake only (Figure 5a), the upper side of the chain is tight because it is decelerating the engine. On the contrary, in the case of engine braking (Figure 5b), the lower portion of the chain is tight because it is transferring the braking torque to the wheel. In the case of front brake only (Figure 5c) the longitudinal force on the rear tire is propulsive. This force generates a counter-clockwise torque which decelerates the wheel and the transmission through the upper side of the chain. Finally, in the combined braking style (Figure 5d), the upper side or the lower side of chain may be tight depending if the transmission inertia or the engine braking prevails respectively.

[fig5]

4. LINEAR STABILITY ANALYSIS DURING BRAKING

The eigenvalues analysis of a racing motorcycle in straight-running motion, for speeds from 25 to 50 m/s with no longitudinal deceleration is presented in Figure 6. The well-known out-of-plane vibration modes wobble and weave, as well as the in-plane modes bounce and pitch are present in this root-locus plot.

[fig6]

There are three additional in-plane modes having frequencies around 16-20 Hz: “rear hop”, “front hop” and “transmission mode”. Figure 7 shows the eigenvectors in terms of chain force, rear/front suspension forces, rear/front tire vertical loads and longitudinal forces, at a speed of 35 m/s in straight-running motion without braking. The rear hop mode (Figure 7a) has a frequency of 19.0 Hz and mainly involves the motion of the rear unsprung mass, therefore the vertical and longitudinal tire forces as well as the rear suspension force fluctuations are greater than the corresponding front fluctuations.

[fig7]

On the other hand, the front hop mode (Figure 7b) has a frequency of 17.9 Hz and primarily concerns motion of the front unsprung mass, therefore the front fluctuations are greater than the corresponding rear fluctuations. Figure 7c depicts the eigenvector of the transmission mode (19.2 Hz). The chain force fluctuation is the greatest component, the rear suspension force, rear tire vertical and longitudinal force are also noticeable. The mode looks like a rear hop with a greater chain force fluctuation, i.e. a great transmission vibration. It should be noted that all three modes have a similar frequency and are strongly related to each other. Therefore, it should be expected that external forcing will excite the modes simultaneously and that there will be some energy exchange among them.

Figure 8 shows the frequency and real part of these modes for speeds varying from 50 to 25 m/s, at a deceleration of 5.7 m/s^2, for different braking styles. The rear wheel frequency is also represented because spin motion may be a forcing source. In all cases front and rear hop modes are stable and have a similar frequencies around 19-20 Hz. The transmission mode has a similar frequency, but it becomes unstable in some cases. Indeed, when using only the rear brake (Figure 8a), the transmission mode is unstable between 40 and 25 m/s. With 100% engine braking (Figure 8b), the transmission mode is unstable in the whole speed range. The case of 100% front brake (Figure 8c) is the only one which has a transmission mode stable in the whole speed range. In the combined braking style with 50% front brake and 50% engine brake (Figure 8d), the transmission mode is again unstable for speeds lower than 45 m/s.
It should be noted that the rear wheel frequency is close to the vibration modes' frequencies in the speed range 40-35 m/s, i.e. 144-126 km/h. It is therefore expected that rear wheel imbalance may excite these vibration modes in this speed range. Indeed, the chatter phenomenon is often observed in the same speed range.

[fig8]

The modal components of the modes at a speed of 35 m/s are represented in Figure 9 (rear hop), Figure 10 (front hop) and Figure 11 (transmission mode), for the four braking styles explained previously.
Rear Hop. The presence of braking forces does not significantly modify the original mode of Figure 7a. However, the chain force fluctuation has an appreciable increment when the rear brake or the engine brake are used. The 100% front brake style has a remarkable fluctuation of the rear tire vertical load, which may lead to a significant reduction of the rear tire adherence.
Front Hop. The original mode of Figure 7b is significantly changed in presence of front braking (Figure 10c and Figure 10d). In these cases there is a great increment of the front tire longitudinal force fluctuation.
Transmission Mode. The more significant differences may be found for combined braking. In this case the fluctuation of the rear tire vertical load increases, i.e. the rear tire adherence may be compromised.
It may be observed that the presence of a braking force causes an increment of the ratio between tire longitudinal force and vertical load.

[fig9]
[fig10]
[fig11]

Further investigations (not reported for brevity) showed that the transmission mode becomes less stable as the deceleration increases and in particular for high values of the engine braking torque.

5. NON LINEAR SIMULATIONS

Since the eigenvalues analysis of the previous section required a linearized model of the motorcycle, additional braking simulations in the time domain are presented in order to highlight the non-linear behavior of the vehicle.

Figure 12 presents a straight-running braking maneuver with a deceleration of 5.7 m/s^2, for speeds from 50 to 25 m/s and with 50% front brake and 50% engine brake. The presence of chatter is evident in Figure 12a where the accelerations of the front unsprung mass reaches 5 g and the accelerations of the rear unsprung mass reaches as much as 20 g. The phenomenon starts at about 37 m/s at the rear wheel and then moves toward the front wheel. The vibration amplitude of the rear suspension is always greater than that of the front one (Figure 12b), confirming linear results (Figure 11) and the experimental data (Figure 1). Figure 12c underlines that the contact forces fluctuate at 19 Hz, once again as expected from the linear analysis (Figure 8d). Finally Figure 12d shows that the engine spin rate is opposite in phase with respect to the rear wheel spin rate, according to experimental data (Figure 2).

[fig12]

As found in the linear analysis of section 4 the chatter vibration disappears if the braking maneuver is performed with 100% front brake (Figure 13): of course this is a limit case since there is always some engine brake unless rider operates the clutch lever, which is not an ordinary behavior. As in the previous section the simulations have been carried out in straight-running. On the track chatter appears during the braking maneuver which typically starts in straight-running and trails into the curve. As reported (section 1) and shown in the simulations (Figure 12) chatter does not initiate instantaneously but commences some time after initial braking, hence when the rider feels chatter he is already in the curve. Furthermore structural modes of the frame may take part in chatter vibration having frequencies not so far from 20 Hz [9], and wheels and swingarm with high modal damping may reduce the phenomenon.
Finally two things should be noted:

1) the motorcycle model makes use of a perfectly flat road, i.e. the chatter phenomenon may appear even without road roughness, even if it is expected that properly spaced road uneveness may further excite the vibration [10]

2) the motorcycle model makes use of perfectly balanced wheels, i.e. the chatter phenomenon may appear even without imbalanced wheels, even if it is expected that such imbalance may further excite the vibration.

[fig13]

6. CHATTERING PHYSICAL INTERPRETATION

How does energy enter into motorcycle during chattering? A braking maneuver may be seen as the composition of two motions: the non-vibrating braking gross motion, i.e. the sequence of equilibrium positions at chosen deceleration and different speeds, and the vibrating motion, i.e. the motorcycle oscillation around these equilibrium positions. During braking, the kinetic energy of the gross motion is lost, but if there is un unstable mode part of this energy is transferred to the in-plane dynamics, i.e. chatter appears. When riders start braking there is a transient, so the vehicle may start vibrating around the equilibrium position. These oscillations lead to a fluctuation of the rear longitudinal slip (see section 3), i.e. to a variation of the longitudinal force which may drive energy into the system, depending on the phase lag between the rear tire longitudinal force and fluctuations of the contact point position. Indeed, Figure 14 shows a two chatter cycles of the simulation presented in section 5 where the rear wheel spin rate oscillation and braking force generate positive work. More in detail, in A (Figure 14) the actual rear wheel angular velocity is lower (60 rad/s) than that of the non-vibrating motion (75 rad/s) and therefore the vibration is a counter-clockwise rotation. In B the actual angular velocity is equal to the non-vibrating angular velocity and so there is no vibration. In C the actual angular velocity (90 rad/s) is greater than the non-vibrating angular velocity: therefore there is a clockwise rotation. Also a rearward longitudinal force is present and so energy enters into the system. In D once again the fluctuating component of the spin velocity becomes zero and changes sign. Even with a longitudinal force still present no energy enters into the system from here on. On the contrary, the power of self-excitation P changes sign and becomes negative in the ensuing short phase. Finally, in E the chatter cycle ends and the condition is the same as in A.
The situation at the end of the braking maneuver (when chatter is fading off) is depicted in Figure 15: in this case is still present a fluctuation in both the longitudinal force and spin rate, but this is a normal situation without any self-exited phenomenon. Indeed, the thrust force (BCD) corresponds to a spin velocity greater than that of the non-vibrating motion (hence the power P is positive) and the braking force (DE) to a spin velocity lower than that of the non-vibrating motion (hence the power P is negative).
[fig14]
[fig15]

CONCLUSIONS

This paper presents the chatter phenomenon both from experimental (section 2) and numerical (section 4-5) points of view. The sources of the rear longitudinal force fluctuation are analytically isolated (section 3) and a physical interpretation of the phenomenon is proposed (section 6). The chatter appears during braking, and its presence is due to an unstable vibration mode, whose modal components are strongly coupled with the motion of the front and rear unsprung masses. The instability may appear, according to simulations presented here, also on a perfectly flat road and with balanced wheels, but it is expected that properly spaced road unevenness and/or wheel imbalances may further excite the vibration. Chatter instability grows at increasing deceleration and increasing engine and/or rear brake proportion in the braking maneuver. Moreover the phenomenon may be reduced by structural components with high modal damping. Numeric simulations aiming towards chatter reduction are currently in progress and will be presented in a future work.

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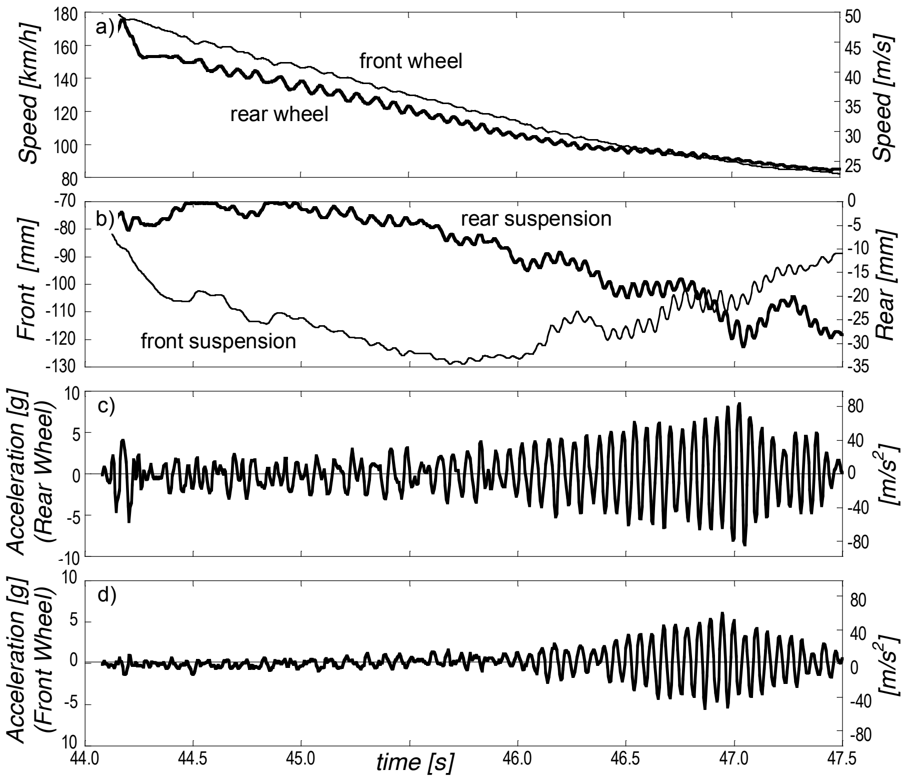


Figure 1 Forward velocity (a), front and rear suspensions travels (b), front (c) and rear (d) unsprung mass accelerations during a braking maneuver of a racing motorcycle.

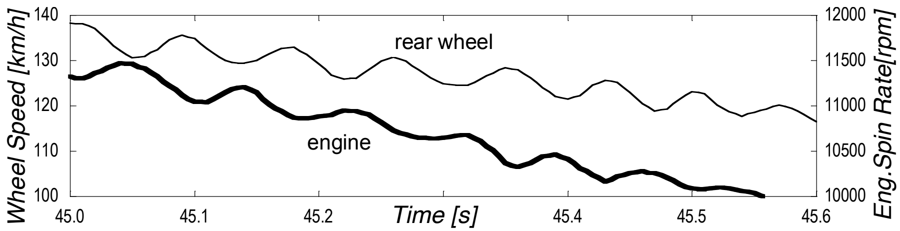


Figure 2 Rear wheel velocity and engine rate during a chattering phase of a racing motorcycle.

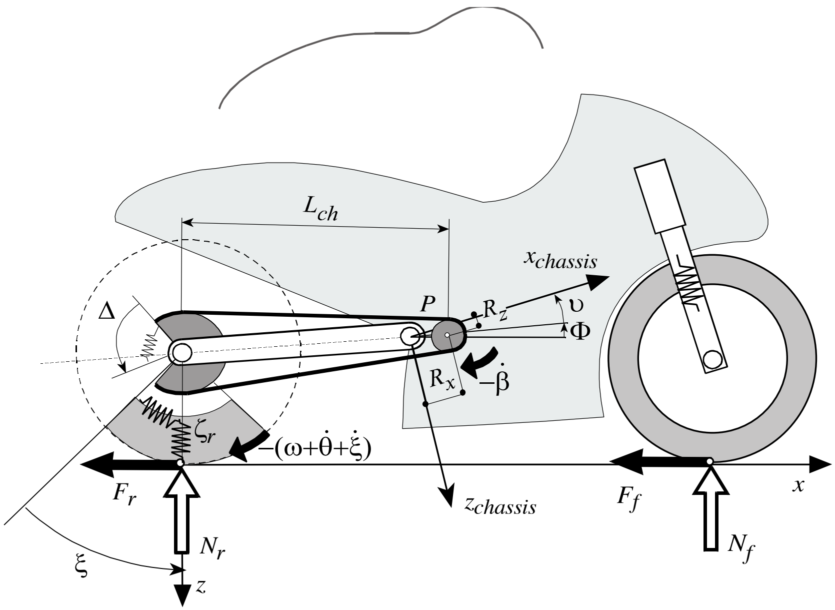


Figure 3 Motorcycle model.

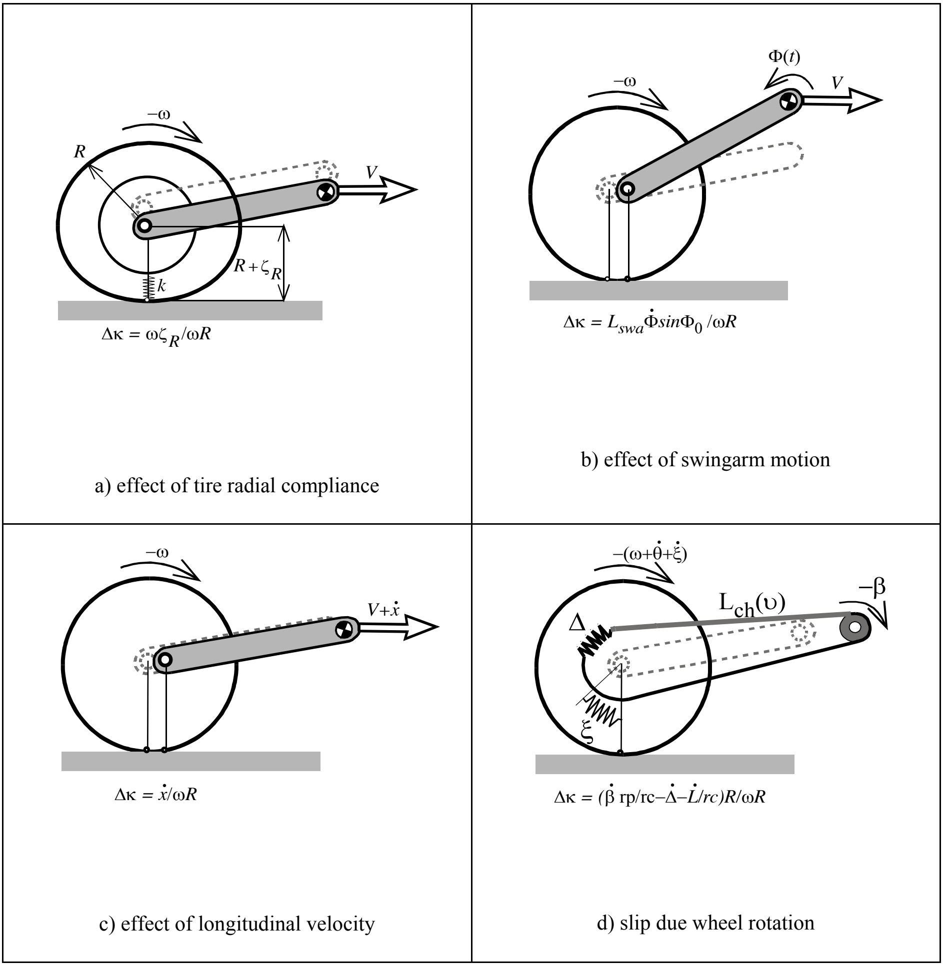


Figure 4 Mechanism generating longitudinal fluctuating slips.

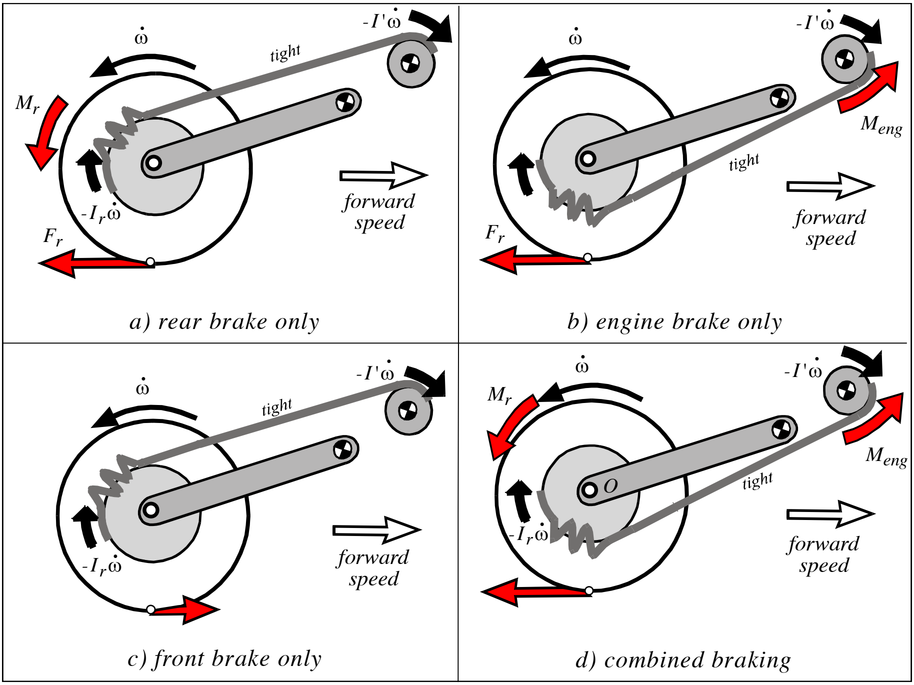


Figure 5 Different kind of braking styles.

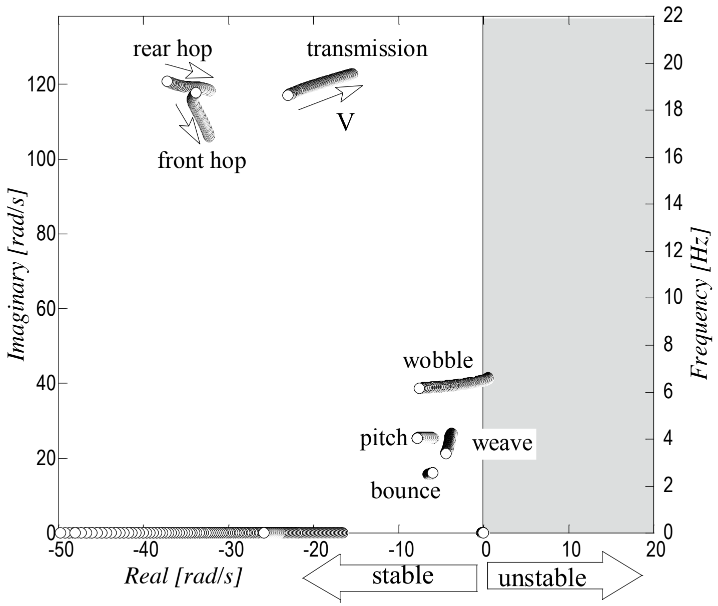


Figure 6 Eigenvalues of motorcycle (straight running, speed 25-50 m/s, deceleration = 0)

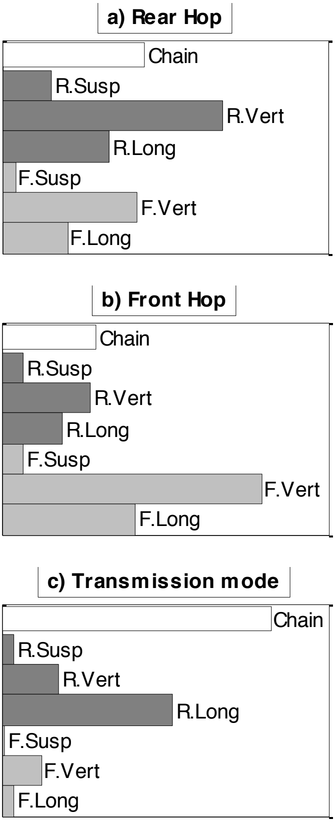
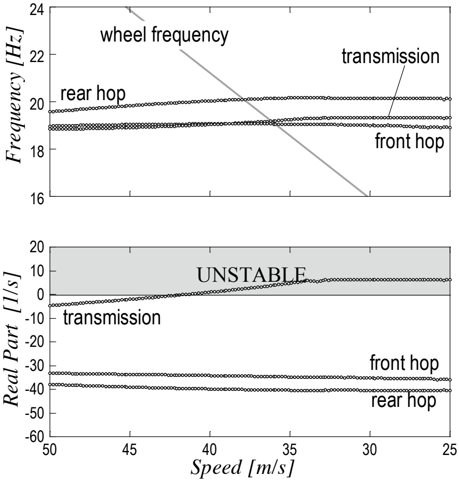
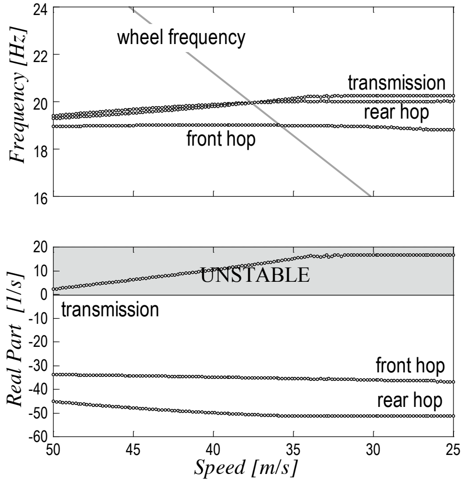


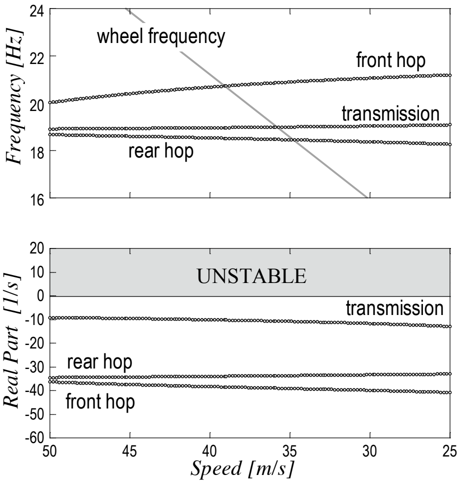
Figure 7 Modal component of rear hop, front hop and transmission mode in terms of force, at 35 m/s and with zero longitudinal acceleration



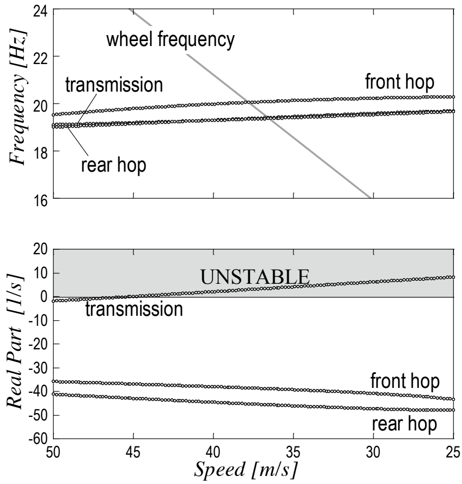
a) 100% rear brake.



b) 100% engine brake.



c) 100% front brake.



d) 50% front brake, 50% engine brake.

Figure 8 - Eigenvalues of motorcycle (straight-running, speed = 50-25 m/s, deceleration = 5.7 m/s^2)

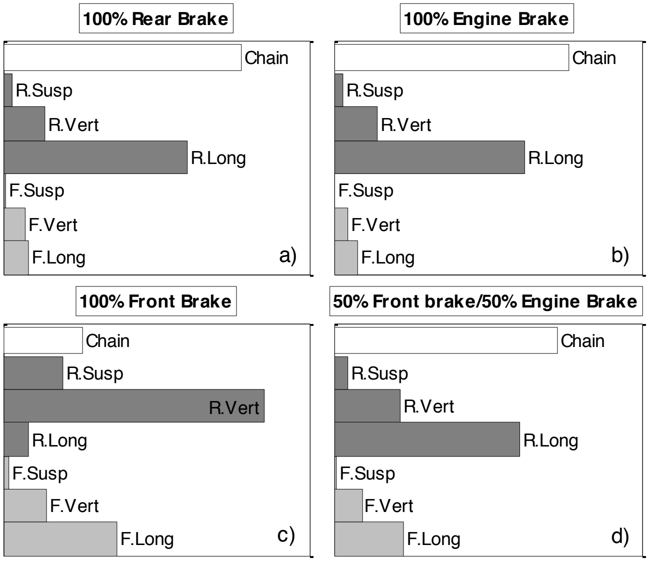


Figure 9 Rear Hop mode shape (speed = 35 m/s, deceleration = 5.7 m/s^2)

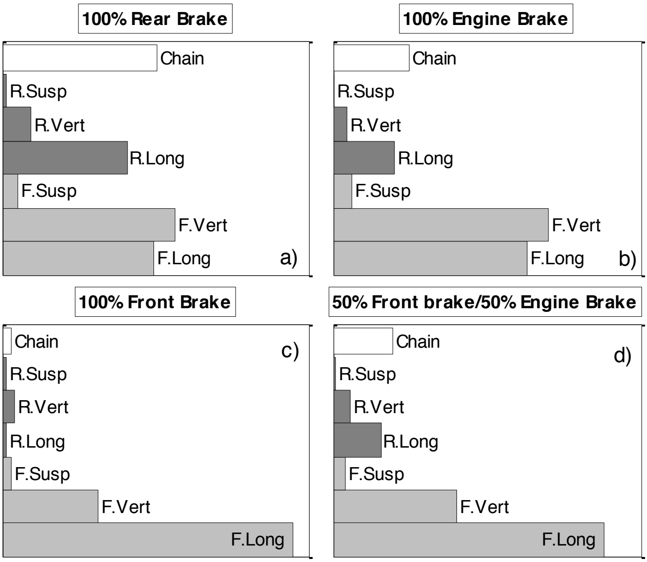


Figure 10 Front Hop mode shape (speed = 35 m/s, deceleration = 5.7 m/s^2)

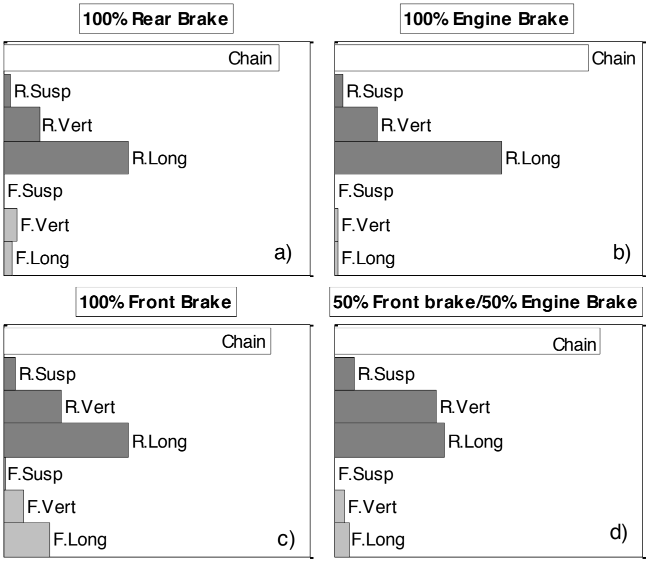


Figure 11 Transmission mode shape (speed = 35 m/s, deceleration = 5.7 m/s^2)

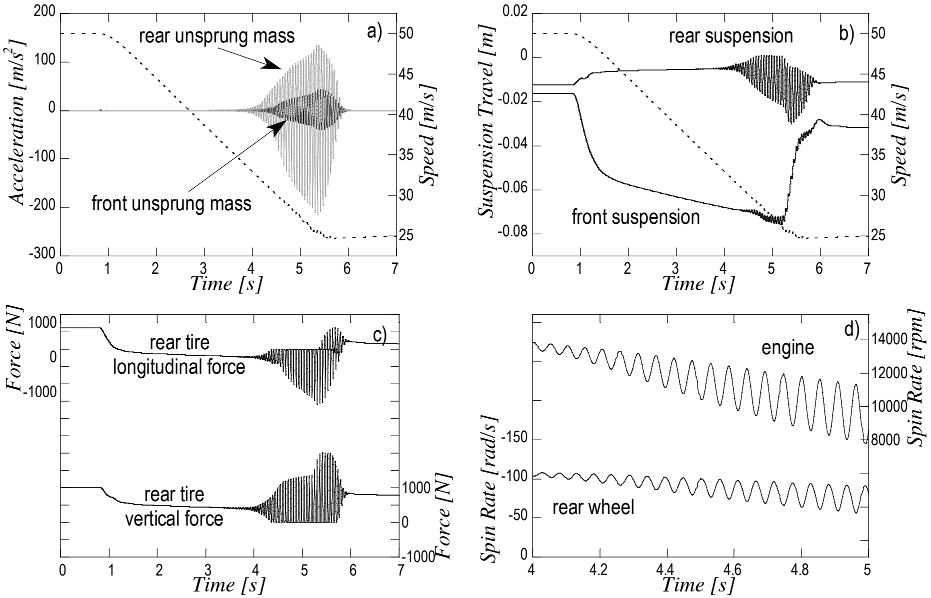


Figure 12 Braking from 50 to 25 m/s with a deceleration of 5.7 m/s^2 (50% front brake/50% engine brake).

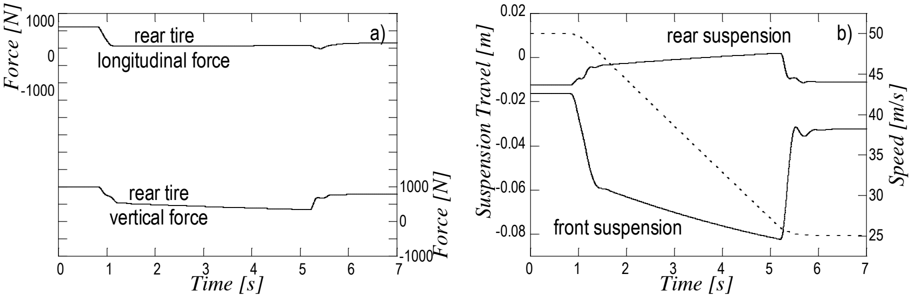


Figure 13 Braking from 50 to 25 m/s with a deceleration of 5.7 m/s^2 (100% front brake).

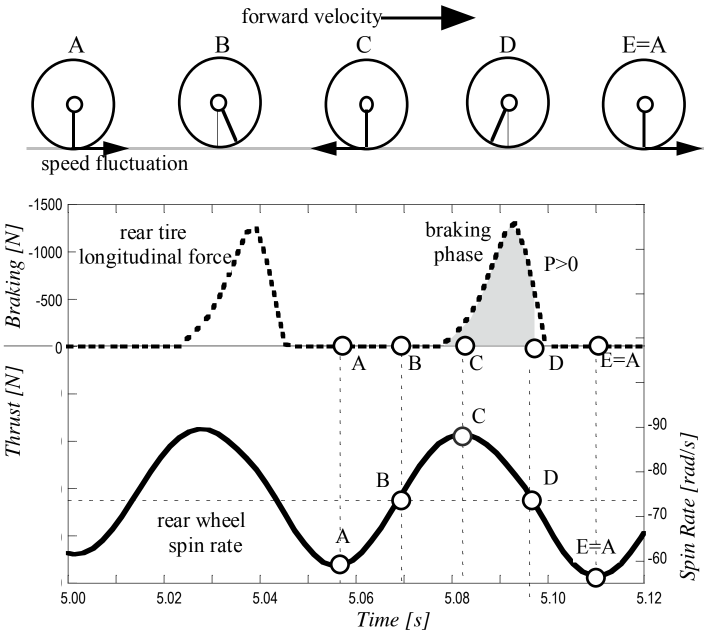


Figure 14 Braking with a deceleration of 5.7 m/s^2 (50% front brake, 50% engine brake, speed 27 m/s)

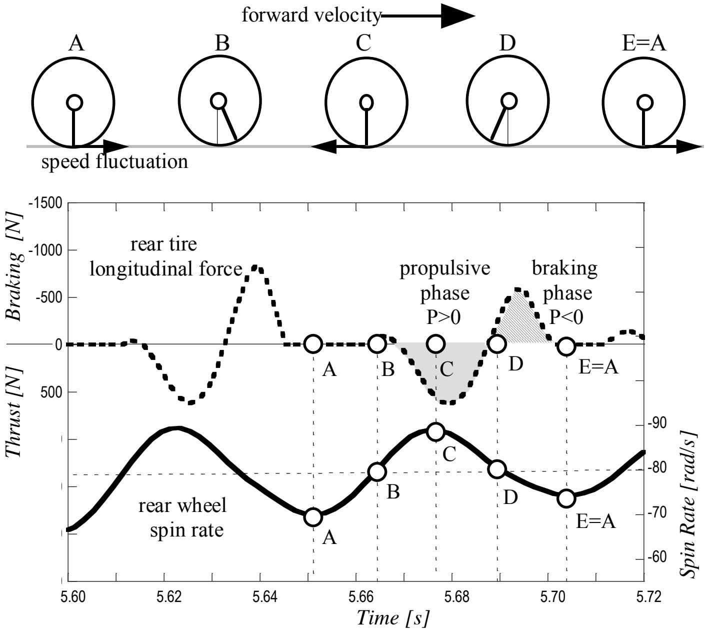


Figure 15 End of the braking maneuver (null deceleration, speed 25m/s)