

Analysis of Alternative Front Suspension Systems for Motorcycles

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Virtual prototyping is a common practice in the automotive industry but still a developing one for motorcycle companies. Among other advantages such techniques can greatly reduce the development time necessary to validate novel ideas. In that fashion, a motorcycle is modelled as a highly detailed multibody system in order to provide a test-rig for a number of alternative front suspension systems. These systems are compared in terms of kinematics and dynamics to provide improved insight into the aspects that need to be considered when deciding upon the use of an alternative system instead of the conventional forks. Apart from the motorcycle model, a driver model has been developed to allow for the systems to be validated through subjection to a variety of driving manoeuvres during simulations. Some indicative results are shown to prove the performance potential of such systems as compared with the default solution of conventional forks and bring into attention differences between each design philosophy.

Keywords: Multi Body Systems; Vehicle Dynamics; Motorcycle; Front Suspensions

1. Introduction

The traditional choice in motorcycle front suspension systems is the telescopic fork. Although after many decades of development the performance of such systems is often already adequate, the design inherits disadvantages are still present. A number of manufacturers have devoted considerable effort in alternative solutions that might replace the obvious choice and deal with the aforementioned issue. Still, examples of production bikes featuring an alternative front suspension system are scarce and the domination of telescopic forks in production as well as in racing is prominent.

Telescopic forks are mechanically prismatic joints, thus the static friction between the sliding parts cannot be eliminated. This can usually be experienced as poor response to small road excitations, especially under conditions that are critical, e.g. negotiating a bump while leaning. The state of the art up-side-down forks take advantage of advanced materials coating such as TiN or carbon matrix composites but the intrinsic disadvantage is still present. Furthermore, their intrinsic design necessitates high stiffness to avoid bending under extreme loads, and also transfers the aforementioned loads at a high point into the frame, resulting in further limitations regarding the frame design. The first aspect is quite crucial as any flexure of the fork in the lateral direction leads into misalignment between the front and rear wheel and between the front tire contact patch and the steering axis. Most riders have experienced phenomena as minor wobble oscillations that may be nothing more than an unpleasant shake of the handle bars but can possibly develop into a tank slapping. To minimize front wheel misalignment, engineers need to achieve high rigidity and, consequently, due to the significant amount of leverage, the front system design is burdened with high values of unsprung mass, albeit not so in up-side-down forks, and high moments of inertia. Also due to the aforementioned leverage, the road forces transmitted through the tire contact patch and via the front system to the frame are magnified, necessitating a large and heavy frame, in order to be adequately stiff. The design layout does not allow for significant variations in suspension geometry parameters such as caster angle, trail and wheelbase. Moreover it is bound to nose dive under braking calling for high damping values if the pitch rate is to be restrained.

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Many interesting proposals have occurred in the pursuit of finer handling qualities for motorcycles. Indeed, ideas such as active control of the suspension geometry have been considered, e.g. [1], while other suggestions include the use of active dampers for motorcycle front suspension, e.g. [2]. Most motorcycle manufacturers and suspension companies have experimented with alternative designs for the front wheel suspension. However, a systematic analysis is still missing. The main objective of this paper is to validate the behaviour of alternative front suspension systems through simulations, both in terms of kinematics (alternation of geometry parameters) and dynamics (resulting frame loads and effects on the motorcycle behaviour). Production models equipped with alternative front suspension systems such as the Bimota Tesi, see Fig. 1a, and the BMW K1200s, see Figs. 1b and 1c, were used as templates for the simulation models. These models were operated by a driver model designed to replicate the human driver in his approach and subjected to various established and indicative testing procedures in order to validate the investigated systems.

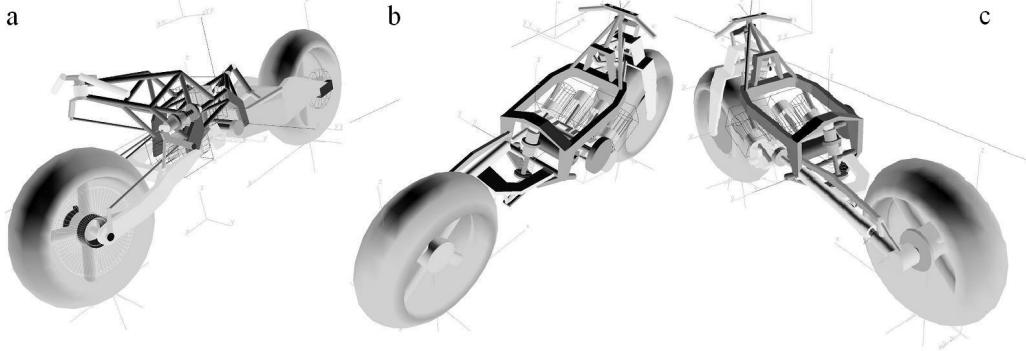


Figure 1: Simulation models (a Bimota Tesi, b/c BMW K1200s)

2. Front suspension system design

Alternative front suspensions exist in a number of variations. The main differences among these designs are related to the use of hub-steering and direct or indirect connection between the handlebars and the steering wheel. Another design detail that also affects the dynamic behaviour of the system defining the way the road loads are transmitted to the frame is the dive percentage under braking. Quantities such as the resulting stiffness to necessary mass ratio also vary among these systems but are easier to rate. Four design variations are validated, namely the Hossack system as used by the BMW K1200s and Britten, see Fig. 2a, the hub-steering suspension as used in the Bimota Tesi, see Fig. 2b, a variation of the James Parker layout known as Radd and applied in the Yamaha GTS and the racing prototype RaTZ, see Fig. 2c, and a hub-steering layout featuring a direct connection to the handlebars used in a slightly different design by the French racing team Atomo, competing in the Battle of the Twins series, see Fig. 2d. Dimensions and physical properties of the models were either acquired or computed and the final geometry of each system was tuned to comply with the established values for caster angle and trail to allow for realistic comparisons between the front systems.

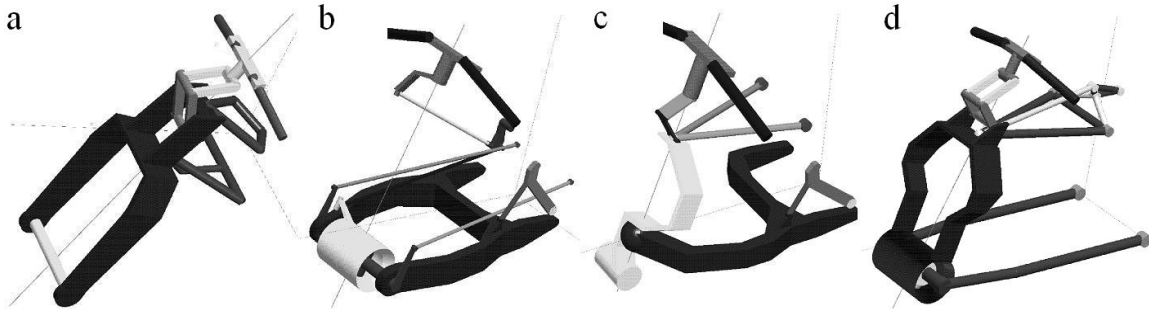


Figure 2: Alternative front suspension systems

3. Motorcycle model

For a reliable dynamical analysis of the front suspension systems the motorcycle has to be modelled as a whole. The model used in the simulations is detailed to a high grade while parts that are considered to be of minor influence to the dynamic behaviour are omitted. A short description of the motorcycle and the subsystems that have been modelled will emphasize that.

3.1 Frame and rear suspension

Modern sports and race bikes benefit from very stiff frames. The highest feasible stiffness is desirable to ensure accurate function of the suspensions. However an overly stiff chassis, especially in the lateral direction, is less so because of the resulting high eigenfrequencies. These indicate a system that requires a driver of very high skill and, possibly, reaction time that is beyond the human abilities. Nevertheless, the frame is considered to be rigid with dimensions taken as typical for the class [3]. The rear suspension elements have also been modelled as rigid bodies. Some mass and inertia values are available in the related literature, albeit to a limited extend and not for all the design layouts that have been considered for the systems. As with the front suspension, the components have been designed in a commercial CAD program and in various layouts, see Fig. 3, so that the required values could be derived. A later research stage will also include elastic components in order to validate the effect the deformations have on the motorcycle's overall dynamic behaviour, especially when tackling road excitations in lean angles higher than $\pi/4$ rad.

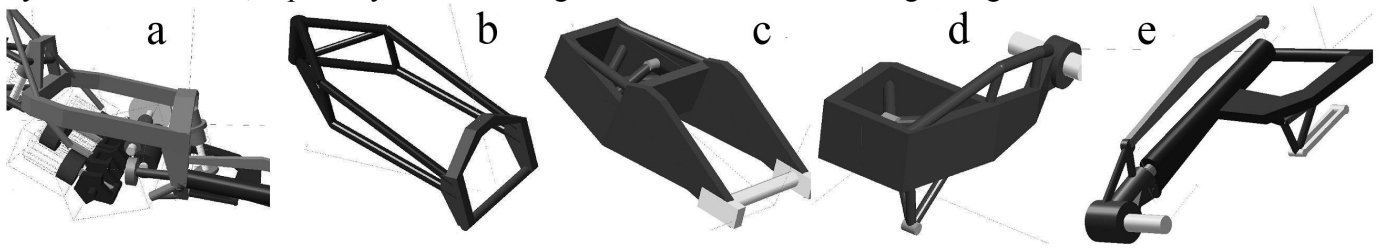


Figure 3: Frames and rear suspension systems

A swingarm supports the rear wheel and is suspended to the frame, while the linkage for the spring-damper element is of full-floater type, designed to provide progressive rise in the suspension ratio. As with the frame, it is intended to model the swingarm as an elastic body in order to compare single- and double-sided swingarms as well as investigating the effect of the lateral stiffness in high lean angles.

3.2 Tires

The development of tire models is not a primary goal of this research project. As the state of the art models are quite advanced, and thus sufficient, a commercial model based on the Pacejka Magic Formula [4] combined with data for certain motorcycle tire models, as supplied by TNO, is used.

3.3 Power-train

The effect of gyroscopic phenomena due to crankshaft revolution is also investigated. This led to a counter rotating crankshaft for the model used to validate the alternative front systems, resulting to instant turn in (higher roll rate) at the entry of the corners, and the V4 format due to its short crankshaft. Power is transmitted to the rear wheel via a shaft transmission, a system similar to the latest Paralever design used by BMW, which almost eliminates the effects of torque reaction to the function of the rear suspension.

4. Driver model

To perform transient response simulations of the models in realistic dynamic performance tests a detailed driver model is necessary. To properly validate the dynamic qualities of the tested vehicle, the driver model must mimic the abilities of an experienced rider, capable of accessing the limits of the bike but also be constrained by human limits such as reaction time. The motorcycle model is operated by a control system designed to replicate the human driver in his approach, e.g., by using full width of the road instead of following a predefined trajectory, and physical terms, e.g., by moving the upper body to ensure better dynamic behaviour of the vehicle. The required torque on the handlebars provided by the driver is calculated via a PD controller with speed dependent gain values, a necessary dependency for the preview length as well. The movement of the driver's upper body is controlled in similar manner, albeit with constant gain values.

4.1 Driver movement

To properly simulate the behaviour of a motorcycle the rider cannot be considered as rigidly attached to the bike's frame. An experienced driver will move his body longitudinally to enhance traction of the front/rear wheel under braking/acceleration and laterally to lower the bike-rider system's centre of gravity height. The amount of movement is dictated by the rider's feeling, as sensed by the longitudinal and lateral acceleration. It is assumed that the rider's upper body is attached to the frame at a certain point close to the bike's Centre of Gravity via a revolute joint allowing rotation around the bike's longitudinal axis and that the rider's CoG (placed ca. 0.25m above this point) moves along an axis parallel to the bike's longitudinal axis. So the rider's movement relative to the bike frame is defined by the coordinates X_p and Φ_r . As the rider does not have a definite measure of the longitudinal acceleration, his body movement is dictated by two auxiliary variables for which he has definite sense: the thrust and brake torque he applies for the movement along the x axis.

Let X_p be the rider body CoG position on the axis of translation, X_{lim} the maximum allowed displacement (set by the system dimensions) and X_{ratio} the coefficient to define the displacement "gain". Let also M_{tb} be the torque required for thrust and brake, calculated by the longitudinal velocity control law output in Nm. Then the rider position in the longitudinal axis is given by

$$X_p = \min[X_{lim}, \max(-X_{lim}, X_{ratio}M_{tb})] \quad (1)$$

Typical values for the coefficients used in this equation are $X_{ratio} \approx 0.075$ and $X_{lim} = \min(2X_{ratio}, 0.125)\text{m}$.

The rider to bike lean angle gain is defined as $Y_{ratio} = \Phi_r / \Phi_b$. Using the driver sensor and setting the yaw angular velocity due to road curvature equal to $\Omega_{zp} = d\Psi_r / dt$ ($1/R$ at the previewed length), the rider lean angle is given by the following equation

$$\Phi_r = \min \left[\text{Lean}_{\max}, \max \left(-\text{Lean}_{\max}, -Y_{ratio} \frac{\Omega_{zp} V^2}{g} \right) \right] \quad (2)$$

with typical values $\text{Lean}_{\max} \approx 0.5\text{rad}$ and $Y_{ratio} \approx 0.9$. It should be noted that these are not definite limits, rather good approximations of the actual rider movement based on experience of bike driving and in results of numerous simulations. The goal of these equations, therefore, is not to define an optimal behaviour of the driver through an optimization function in order to achieve higher speed or agility but to consider a typical behaviour in dynamic simulation. Indeed, the objective in his body movement is not to apply control force and

torque, namely the pressure on the handlebars and rear-sets necessary for his movement, but to better position the CoG of the system.

4.2 Steering torque control law

The control law for the steering torque applied on the handlebars by the rider is of the PD form, namely

$$M = P(\Phi_a - \Phi_d) + D(\Omega_a - \Omega_d) \quad (3)$$

where Φ_a and Φ_d denote actual and desired bike roll angle respectively, Ω_a and Ω_d the actual and desired bike roll angular velocities. In case of experimental data, the application is quite simple as both the desired roll angle and roll angular velocity are known and the actual values are simulation measurements. In this case the control law can be easily enhanced by use of an integral term. The definition of the PD (or PID) coefficients is done by satisfying the condition that the motorcycle eigenmodes are stable. Application on a typical sport/racing motorcycle (weight ca. 175kg, even load distribution on wheels, caster angle ca. 24degree, wheelbase ca. 1400mm, trail ca. 100mm) results in the values $P \approx 625$, $D \approx 50$

Taking these values as a starting point, one can accurately define the respective values for the system under consideration. Depending on the motorcycle specification the values used in our up to date simulations varied through the velocity range between 575 to 675 for the proportional and 25 to 75 for the derivative gain coefficient. However, the route taken in case of no available measurements is slightly different. In this case the desired roll angle and angular velocity have to be user defined/computed.

The driver steering torque is based (rightfully) on the assumption that the motorcycle is a stable system. Thus, regardless of the course, the optimum operation is of a system in equilibrium and this is the objective of the driver. The torque applied by the driver shall be the minimum required to bring the motorcycle from one state of equilibrium (straight) to the desired one (necessary roll angle for the given radius of turn and velocity) as the system can absorb small perturbations due to its stable albeit lightly damped nature. Consequently, the desired angular velocity on the longitudinal axis, i.e. the roll angular velocity Ω_d , should be zero for a smooth transition from one state of equilibrium to the other.

The desired roll angle Φ_d is a function of vehicle and road parameters. Consider Fig.5, representing the vehicle's position on the selected trajectory. The values of Y (lateral position), H (preview length) and L (lateral deviation from the desired vehicle position) and the vehicle's longitudinal velocity V are derived by a typical vehicle-driver sensor. Suppose that a bow connecting the intermediate points δ, ε with point γ taken as bow-centre and a radius $R = \delta\gamma$ is drawn. This radius is an adequate estimate of the turn radius at the position of the vehicle and for small values of Y (so that points α and δ almost coincide) the drawn bow is a good approximation of the respective curve connecting α and ε . Thus, the aforementioned radius is equal to

$$R = \delta\gamma \approx \varepsilon\gamma \text{ or } R = \frac{H^2 - L^2}{2L} \quad (4)$$

used in the following equation describing the condition required in order to ensure moment equilibrium along the longitudinal axis

$$\Phi_d = a \tan\left(\frac{gV^2}{R}\right), \text{ where } g \text{ denotes the gravity acceleration.} \quad (5)$$

Of course this is the most elementary feasible expression. To take into account road camber (banked track), repositioning of the centre of gravity due to rider lateral movement, track width and actual values of Y , a number of parameters has to be introduced to the formula. So, let us first assume that there are two preview

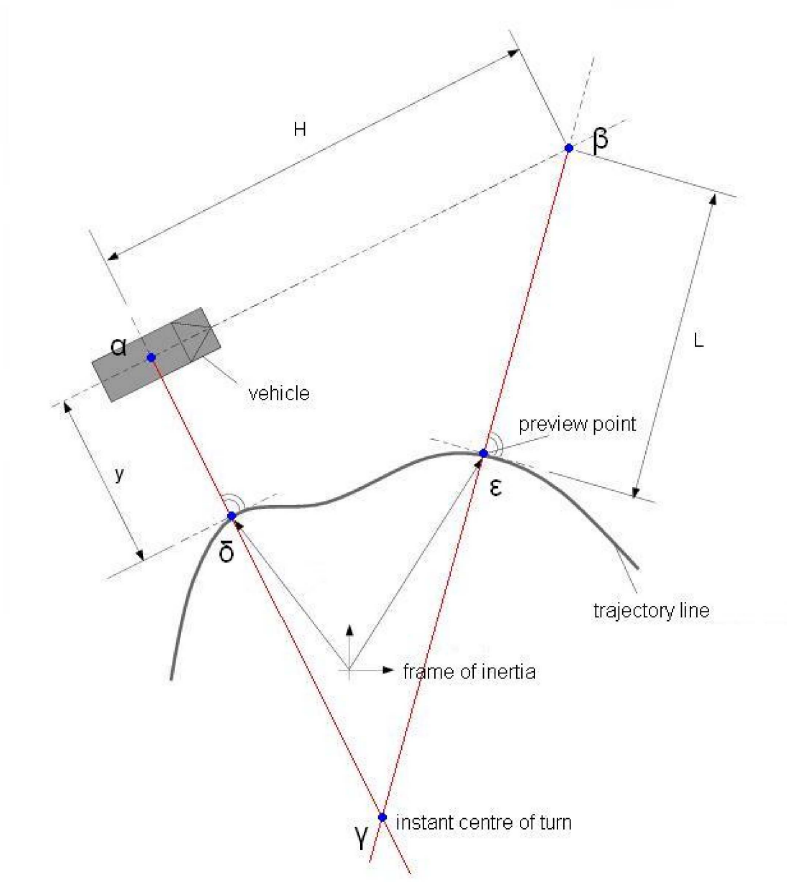


Figure 4: Driver model preview

points resulting in two values of L and H : L_1, L_2 and H_1, H_2 respectively. Let Φ_r be the road camber angle, RW the track width and RW_r the ratio of the road track width we desire to be used to the total road width. Values for RW_r that proved to be valid during simulations range between 0.5 and 1. The lower limit corresponds to a conservative riding style whereas the upper limit to a more aggressive riding style with the driver making use of the full road width. A coefficient to compensate for the repositioning of the CoG is C_{cog} . Let H_b and H_r be the height of the bike and rider upper body CoGs respectively. Let also M_b and M_r be the respective masses and Y_{ratio} be the rider's lateral movement gain coefficient or ratio of the rider's to the bike's lean angle, already defined as $Y_{ratio} = \Phi_r / \Phi_b$. Then, C_{cog} is set equal to

$$C_{cog} = \frac{(M_r + M_b)H_b}{M_r H_r (1 + Y_{ratio}) + (M_r + M_b)H_b} \quad (6)$$

As typical can be considered the following values $M_r \approx 50\text{kg}$, $H_r \approx 250\text{mm}$, $H_b \approx 625\text{mm}$. These result in values of 0.9 to 0.925 for the C_{cog} ratio. One last vehicle measurement is required, the bike's yaw angular velocity $\Omega_z = d\Psi_b / dt$. The final formula for the required roll angle is then

$$\Phi_d = a \tan \left\{ \text{abs}(\cos \Phi_r) \frac{V^2}{0.5g(H_1 + H_2)} \sin \left[\begin{aligned} & \left[a \tan \left(\frac{C_{\text{cog}} L_1 + RW_r (\Omega_z \frac{RW}{2} - Y)}{H_1} \right) + \right. \\ & \left. a \tan \left(\frac{C_{\text{cog}} L_2 + RW_r (\Omega_z \frac{RW}{2} - Y)}{H_2} \right) \right] \end{aligned} \right] \right\} \quad (7)$$

It is a simple task to derive the final expression but the proper use of the coefficients for the CoG correction due to the driver's body lean and the use of road is rather complicated. This is the final expression, certified by a number of simulations with bike models of totally different specification.

5. Front systems analysis

A technical analysis of the examined alternative front suspension design layouts is necessary in order to provide a substantial understanding of the operation and the advantages and disadvantages of each case. The main difference among the various design proposals is the use or not of the so called "hub centre". This term refers to the use of a complex wheel centre that allows the wheel to be steered on the supporting axis rather than be rigidly attached and steered with the supporting axis. The other class of alternative front suspension systems employs a double link layout in various formations. There has also been the case of the BMW Telelever and the similar SAXON system that use one wishbone in cooperation with a prismatic joint, namely a slider mechanism quite similar to the Mc Pherson struts used in cars, but these designs have been omitted in the present comparisons. Although differentiating the functions of steering and suspension, these designs do not have the potential of the other cases as they do not avoid all the shortcomings of the telescopic forks, thus it was not considered necessary to examine their properties in depth.

5.1 Hub-centre systems

In this category belong front suspension systems such as the one used in the Bimota Tesi as well as applications such as the Neracar and the system designed by the British engineer Joe Difazio, both preceding the Italian company. The examined layouts correspond to the models depicted in Figs. 2b and 2d.

Usually the design consists of a hub attached to a king-pin so that it is allowed to rotate in the direction of the vertical axis. A hub of larger diameter is attached to the king-pin mounted hub and is allowed to rotate in the direction of the lateral axis. Rigidly mounted on the outer hub is the front wheel while the king-pin is usually connected to the frame via a front swingarm. The inner hub is allowed to steer but not to rotate and to restrain the king-pin in this direction of rotation a number of solutions can be applied, e.g. to rigidly mount the king-pin on the front swingarm or use torque arms to connect it to the frame. The vertical axis of the king-pin defines the steering axis, thus steering geometry parameters such as caster angle, trail, or wheelbase, can be easily altered. Furthermore, if the inner hub is not rigidly attached to the front swingarm the steering geometry values can be tuned to be stable or increasing/decreasing in relation to the wheel vertical movement. The proximity of the steering axis, as defined by the king-pin and the axis of rotation greatly reduces the possible flexures and the resulting front wheel misalignments.

The system used in the Bimota Tesi, see Fig. 2b, has the king pin attached to a front swingarm, mounted on the frame. It applies a pair of torque arms to prevent the king-pin from rotating in the lateral axis direction and also transfer part of the braking force to the frame. They are adjustable in length, thus defining the steering

axis castor angle and rate of caster gain/loss under braking. The steered hub is connected to the handle bars via linkage bars, designed to ensure the highest feasible linearity.

5.2 Dual-link systems

In this category belong systems such the Hossack system used in the BMW K1200s, see Fig. 2a, the Elf-Honda raced in the late eighties and early nineties, the Radd system used in the Yamaha GTS1000, see Fig. 2c, as well as systems designed by Tony Foale and others. In this case the wheel spindle is rigidly attached to an upright which is successively connected to the frame via pivoted arms and/or wishbones. The differences among these designs are determined by the arm positions and the means of steering the upright.

The links can be positioned either inside or outside the wheel circumference. If they are inside, the link has to be single sided and curved outward to allow for tyre clearance on full steering lock, a shape far from ideal to handle the resulting loads. In case of both links laying in this manner, as per the Elf design, the upright length is limited resulting in low weight and high stiffness. On the opposite end lies the Hossack design employing short links over the wheel, supporting a long upright. This solution allows for short, thus light and stiff, links but due to the increased leverage the loads on the joints are far greater as are the bending loads on the upright. Furthermore, it necessitates the use of a frame similar to the one used for conventional forks due to the location of the link joints thus rendering the structural use of the engine as the main frame impossible. Another design solution combining elements from the two extremes is the Radd system, locating the lower link inside the wheel circumference and the upper link above the wheel. Steering of the upright is achieved via direct or indirect connection. In case of the Elf system a complex linkage was used whereas the Hossack and Radd systems employed direct connection via levers and a telescopic torsion tube respectively.

6. Simulation results

To validate the performance of the aforementioned systems, simulations of situations close to real driving conditions are necessary. In order to achieve this goal, the models are subjected to various testing procedures such as a double lane change, a u-turn, braking/acceleration simulation. Apart from these widely accepted tests, very indicative of the tested suspension systems performance proved to be the "s-bend" test. This test consists of a short straight leading to two successive turns of constant radius with a short straight in between and another after the second turn. The corners are tackled in non-constant speed, predefined so that the models operate at the limit of tire adhesion as defined for commercial tires fitted to sport bikes. Thus, through a test distance of 750m the model is subjected to validation of its overall performance. The simulations have been carried out using the method of multibody systems [5] and the commercial software SIMPACK® and respective animations are available at the institute web page (<http://www.itm.uni-stuttgart.de/research/motorbikes>). The reference model corresponding to the diagrams consists of a spaceframe construction and a conventional swingarm, see Figs. 3b and 3c, due to their higher stiffness to mass ratios.

The significance of anti-dive geometry is underlined in Fig. 5a, the variation in the motorcycle's pitch angle is much greater in the model equipped with a conventional fork (thick line) whereas in the other models, see Fig. 2, it depends on the 'anti-dive' percentage of the suspension design but is always lower. Furthermore, the resulting bending torque on the frame's lateral axis due to road forces transmitted to the frame further underlines the superiority of the alternative systems, see Fig. 5b, as compared to a conventional fork system. The final decision among these systems is made by consideration of many parameters, not excluding the engineering packaging limitations. Judging in terms of kinematics and dynamics, the design layout depicted in Fig. 2c seems to combine desirable characteristics to a higher degree than the other designs that have been validated. Each design shows advantages and disadvantages and the design engineer must decide in which

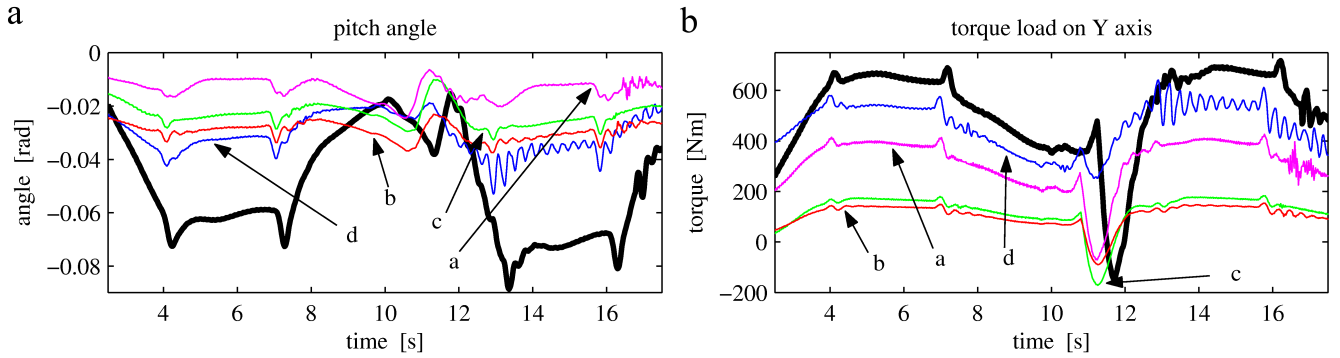


Figure 5: Simulation results

area he shall benefit. To understand this, consider Fig. 6. The sum of all forces in the suspension joints is shown and in this case it is clear that the ball-joint connecting the lower swingarm to the Hossack fork (line a) is heavily stressed. The respective weakness of hub-steering designs results from the construction of the king-pin and its bearings as flexure or high tolerance results in wheel lateral displacement, a very critical parameter of stability. Another point that requires overall evaluation of the simulation results is the perturbations apparent during the braking phase in some vehicles. A thorough investigation on the resulting tire forces certified that they were due to excessive anti dive geometry, leading to high frequency oscillations of the whole model on the tires. It may seem contradicting as model c also employs a rather high percentage of anti dive, yet seems immune to oscillatory behaviour under braking. This advantage results from the use of significantly longer wishbones than those used in model a, allowing improved steering geometry consistency.

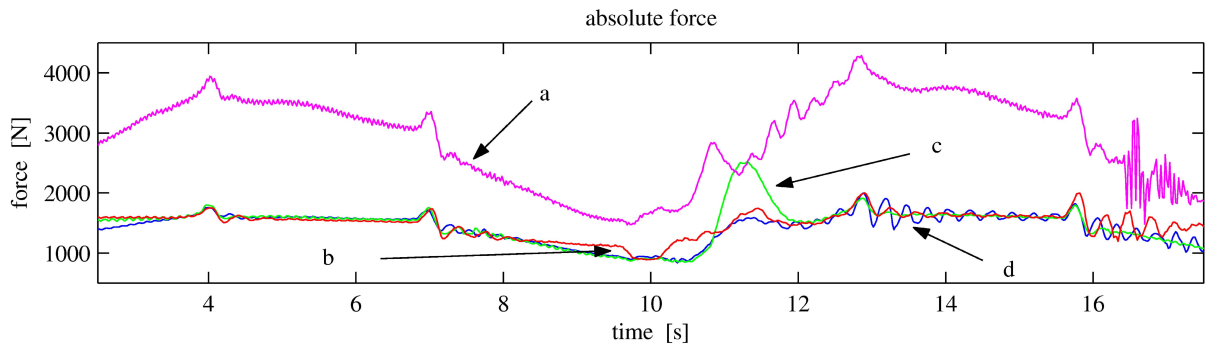


Figure 6: Simulation results

Simulations provide a valuable mean of evaluating the overall dynamic behaviour of the vehicle and for this reason the results depicted in Figs. 7a and 7b are shown. First consider Fig. 7a. The vehicle roll velocity is used to judge the motorcycle's reluctance to lean into a curve. The values extracted by the simulation results show that the models are willing to roll into the curve, as expected by the geometry settings. The prominent perturbations in the roll velocity of models a and d, and to a lesser extend model b, are indicative of unstable behaviour. The main difference among these designs is the point on the simulation track that the models show this undesirable behaviour. Models a and b present a slight disturbance while exiting the first and second corner respectively, while model d presents seriously critical behaviour at the end of the braking phase, while starting to lean in order to tackle the second corner. To properly interpret a simulation plot, the engineer needs to combine the results with those of the other graphs in order to extract reliable assumptions. In Fig. 7b it is clearly shown that all aforementioned models present a higher yaw rate with respect to the roll rate. Models a and b present abrupt transitions in various points in the roll rate to yaw rate graph, a result that can be due to driver input whereas model d shows a higher coverage of the graph area. Thus, the instability can be attributed

to the setup of the vehicle being keener to oversteer rather than deficiencies of the front suspension system. Whether the engineer might opt for higher response accepting decreased stability is another difficult decision.

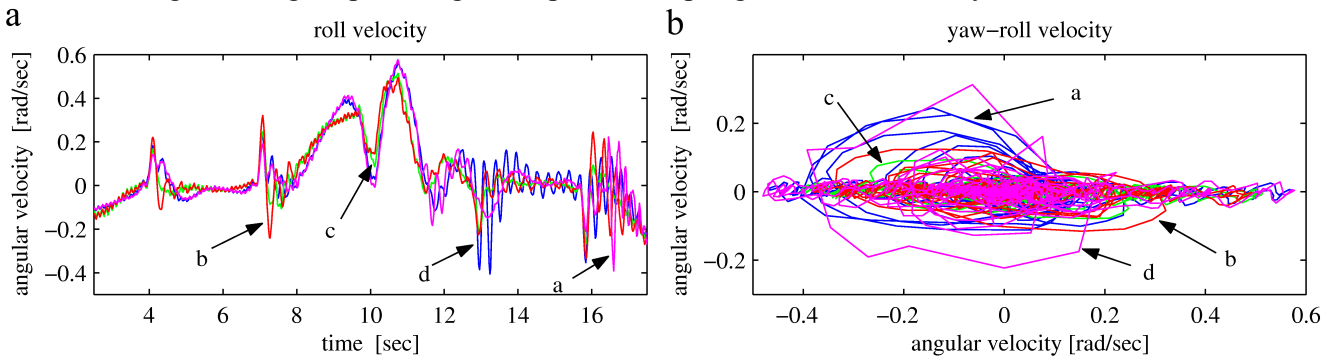


Figure 7: Simulation results

7. Future directions

The next step in this research project is the verification of the simulation results with experimental data and the incorporation of elastic bodies in the model, as it is of major interest to investigate how the component flexures might affect the dynamic behaviour of the vehicle. Furthermore, modelling the frame as an elastic body shall allow the investigation of the lateral stiffness effect, a critical parameter for motorcycle dynamic behaviour under lean angles higher than 45 degrees where the motorcycle suspension efficiency deteriorates. The effect of using different types of frames such as beam frames or spaceframes, see Fig. 3a and b, single-sided swingarms instead of conventional ones, see Figs. 3c and 3d, axle transmissions, see 3e, instead of chain transmission and of various other details will be investigated. A driver model upgrade such as the optimal manoeuvre method [6] is also considered and other forms of handling test procedures [7] will be applied too. Apart from the aforementioned model upgrades, new directions on motorcycle dynamics are constantly investigated. In that frameset is the application of two counter rotating Control Moment Gyros to act as a stabilising device. The frequency of operation is considerably higher than 15Hz so that the rider will not be disturbed by the intervention. For further information and animations, again refer to the institute webpage.

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