Program and Book of Abstracts

Bicycle and Motorcycle Dynamics 2010

Symposium on the Dynamics and Control of Single Track Vehicles

A.L. Schwab and J.P. Meijaard (eds.)

Delft University of Technology
Delft, The Netherlands
20 - 22 October 2010

BMD2010

Front cover:
“Levende Etalage Dag”, A view of the wall at Pluympot, Delft, April 2010.
A Bicycle in Mosaic, a photo made by Shirley Elprama.

Contents

Preface

This booklet contains the programme and abstracts of the Bicycle and Motorcycle Dynamics Symposium 2010, held in Delft, 20 - 22 October 2010. The idea for this symposium came up about a year ago when we met in Delft. It appeared that sessions about the dynamics and control of single-track vehicles had been part of more general conferences, such as the biennial IAVSD symposia and the SAE congresses, most notably the SAE congress in Detroit, February 1978, but that dedicated scientific meetings had not been held. Regarding the increased research activity in recent years, we saw a dedicated symposium as timely and needed in order to have a podium for the international exchange of results and ideas.

The organization of this symposium would not have been possible without the help of many. In the first place, we should like to thank the authors, who warmly received the idea and responded to our call beyond expectation.
The help of the other members of the scientific committee contributed to the quality of the scientific content. The local assistance of both Jodi Kooijman and Yani Sutjiadi was essential. We thank Tom Scarpas for suggesting and allowing us to make use of Yani's organizational expertise.
After the colloquium was already announced, the International Association for Vehicle System Dynamics (IAVSD) offered us to collaborate and give us the opportunity to publish some papers in their organ, Vehicle System Dynamics.
The contributions of the sponsors kept the cost for the participant low. These sponsors are the Royal Netherlands Academy of Arts and Sciences, Delft University of Technology, TNO, Batavus, Yamaha, KTM and Harley-Davidson.

Arend Schwab and Jaap Meijaard,
October 2010

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Program

Bicycle and Motorcycle Dynamics 2010
Symposium on the Dynamics and Control
of Single Track Vehicles

WEDNESDAY, 20 OCTOBER 2010 at ROOM: A

8:30 - 9:00 Registration
9:00 - 9:30 Opening Ceremony by: *Prof.drs. Marco Waas, Dean faculty 3ME, TU Delft.*

Session 1.1

9:30 -10:00 Rider Control of a Motorcycle near to its Cornering Limits
R. S. Sharp, University of Surrey, United Kingdom

10:00 -10:30 Analysis of the Biomechanical Interaction between Rider and Motorcycle by Means of an Active Rider Model
Valentin Keppler, Biomotion Solutions, Germany

10:30 - 11:00 Coffee Break

Session 1.2

11:00 - 11:30 Modeling Manually Controlled Bicycle Maneuvers
Ronald Hess, Jason K. Moore, Mont Hubbard and Dale L. Peterson, University of California, Davis, USA

11:30 - 12:00 A Dynamic Inversion Approach to Motorcycle Trajectory Exploration
A. Saccon, Instituto Superior Técnico, Portugal, J. Hauser, University of Colorado, USA, and A. Beghi, University of Padova, Italy

12:00 - 12:30 Speed-Adaptive Path-Following Control of a Riderless Bicycle via Road Preview
C.K. Chen and T.K. Dao, Dayeh University, Taiwan

12:30 - 13:30 Lunch

Session 1.3

13:30 - 14:00 Proposal of Personal Mobility Vehicle Based on Stabilization Control of Two-Wheel Steering and Two-Wheel Driving
Chihiro Nakagawa, Osaka Prefecture University, Japan, Kimihiko Nakano, Yoshihiro Suda and Yuki Hirayama, University of Tokyo, Japan

14:00 - 14:30 Simulation and Control of the Anaconda
P. Kabeya and O. Verlinden, University of Mons, Belgium

14:30 - 15:00 Intrinsic Stability of a Classical Monocycle and a Generalized Monocycle.
M. J. Coleman, University of Vermont, USA, and J. M. Papadopoulos, University of Wisconsin-Stout, USA

15:00 - 15:30 Coffee Break

WEDNESDAY, 20 OCTOBER 2010 at ROOM: A

Session 1.4

15:30 - 16:00 Research Activity of the mOve Research Group in the Field of Electronic Motorcycle Control: Past and Future
S.M. Savaresi, Politecnico di Milano, Italy

16:00 - 16:30 Motorcycle State Estimation for Lateral Dynamics
A.P. Teerhuis and S.T.H. Jansen, TNO Automotive, The Netherlands

16:30 - 17:00 On Linear-Parameter-Varying Roll Angle Controller Design for Two-Wheeled Vehicles
M. Corno, Delft University of Technology, The Netherlands, M. Massaro, R. Lot, University of Padova, Italy, and S.M. Savaresi Politecnico di Milano, Italy

17:00 - 17:30 Dynamic Model of a Bicycle with a Balancer and Its Control
Lychek Keo and Masaki Yamakita, Tokyo Institute of Technology, Japan

18:00 - 22:00 Dinner and Bicycle Exhibition/Test at Sports Center of TU Delft

THURSDAY, 21 OCTOBER 2010 at ROOM: A

08:30 - 09:00 Registration

Session 2.1

9:00 - 9:30 The Balance of Walking and Bicycling
Andy Ruina, Cornell University, USA

9:30 - 10:00 Critique of Assumptions Underlying Bicycle Handling Research
Jim M. Papadopoulos, University of Wisconsin - Stout, USA

10:00 - 10:30 The Basic Human Input to Bike Control
A.J.R. Doyle, University of Sheffield, United Kingdom

10:30 - 11:00 Coffee Break

Session 2.2

11:00 - 11:30 Design of a Novel Aerodynamically Efficient Motorcycle
D. J. N. Limebeer and A. Sharma, University of Oxford, United Kingdom

11:30 - 12:00 Motorcycle Control by Variable Geometry Rear Suspension
S. A. Evangelou, Imperial College London, United Kingdom

12:00 - 12:30 Sensitivity Analysis of Multibody Systems : Evaluation of Mountain Bike Dynamical Performances
A. Poncelet, N. Docquier and P. Fisette, Université Catholique de Louvain, Belgium

12:30 - 14:00 Thematic Lunch "The Future of Cycling: where Industry meets University"
Moderator: Han Goes, Q Square, The Netherlands

Session 2.3

14:00 - 14:30 Experimental Analysis of Rider Motion in Weave Conditions
A. Doria, M. Formentini and M. Tognazzo, University of Padova, Italy

14:30 - 15:00 Motion Analysis of a Motorcycle Taking Account of Rider's Effects
S. Zhu, S. Murakami and H. Nishimura, Keio University, Japan

15:00 - 15:30 The Influence of a Passive Rider on the Open Loop Dynamics of a Bicycle.
A. L. Schwab, J. D. G. Kooijman, Delft University of Technology, The Netherlands, J. P. Meijaard, University of Twente, The Netherlands

15:30 - 16:30 Coffee Break & Poster Session

Poster 1 IntelliBike: Condition Monitoring of Our Cycling Infrastructure
M. Taylor and C. A. Fairfield, Edinburgh Napier University, United Kingdom

Poster 2 Accurate Measurement of Bicycle Physical Parameters
Jason K. Moore, Mont Hubbard, Dale L. Peterson, University of California, Davis, USA, A. L. Schwab and J. D. G. Kooijman, Delft University of Technology, The Netherlands

Poster 3 OBD: Open Bicycle Dynamics
Dale L. Peterson and Mont Hubbard, University of California Davis, USA

THURSDAY, 21 OCTOBER 2010 at ROOM: A

Poster 4	Hypotheses Formulation in Multibody Modeling: Application to Bicycle Transmission Dynamics *N. Docquier, A. Poncelet, P. Fisette and J.C. Samin, Université Catholique de Louvain, Belgium*
Poster 5	An intelligent Frontal Collision Warning system for Motorcycles *F. Biral, University of Trento, Italy, R. Lot, R. Sartori, University of Padova, Italy, A. Borin Yamaha Motor R&D Europe, Italy, and B. Roessler, SICK AG, Germany*
Poster 6	Modeling of a Motorcycle using Multi-Body Dynamics and Its Stabilization Control *S. Murakami, S. Zhu and H.Nishimura, Keio University, Japan*
Poster 7	Experimental investigation on the shimmy phenomenon *D. De Falco, G. Di Massa and S. Pagano, Seconda Università di Napoli, Italy*
Poster 8	An eleven degrees of freedom dynamic model of a motorcycle *W. Ooms, I.J.M. Besselink and H. Nijmeijer, Eindhoven University of Technology, The Netherlands*
Poster 9	Automatic Generation of Linearised Equations of Motion for Moving Vehicles *B.P. Minaker, University of Windsor, Canada*
Poster 10	Assessing Slip of a Rolling Disc and the Implementation of a Tyre Model in the Benchmark Bicycle *E.J.H. de Vries and J.F.A den Brok, Delft University of Technology, The Netherlands*
Poster 11	Three Structural Component Linkage Front Suspension and Directly Connected Suspension for Motorcycles *Robert Rae, RaerDesign, Ireland*

Session 2.4

16:30 - 17:00	Some Investigations on the Wobble Mode of a Bicycle *Chr. Ott, B. Angrosch, J. Edelmann and M. Plöchl, Vienna University of Technology, Austria*
17:00 - 17:30	Comparison of a Bicycle Steady-state Turning Model to Experimental Data *S. M. Cain and N. C. Perkins, University of Michigan, USA*
17:30 - 18:00	Steering Characteristics of Motorcycles *Shigeru Fujii, Souichi Shiozawa, Akinori Shinagawa and Tomoaki Kishi, Yamaha Motor Corporation Limited, Japan*
18:15 - 22:00	Delft Walking City Tour and Conference Banquet at de Waag Restaurant

FRIDAY, 22 OCTOBER 2010 at ROOM: C

8:30 - 9:00 Registration

Session 3.1

9:00 - 9:30 On the Validation of a Motorcycle Riding Simulator
V. Cossalter, R. Lot and S. Rota, University of Padova, Italy

9:30 - 10:00 A Virtual Rider for Reproducing Experimental Manoeuvres
M. Massaro, V. Cossalter and R. Lot, University of Padova, Italy

10:00 - 10:30 A Bicycle Model for Education in Machine Dynamics and Real-time Interactive Simulation
J.L. Escalona and A.M. Recuero, University of Seville, Spain

10:30 - 11:00 Coffee Break

Session 3.2

11:00 - 11:30 Build It Wrong, But Build It – A Bicycle Trek
R. E. Klein, University of Illinois at Urbana-Champaign, USA

11:30 - 12:00 Modeling Mechanical Optimization in Competitive Cycling
P. Cangley, University of Brighton, United Kingdom

12:00 - 12:30 Design Sensitivity Analysis of Bicycle Maneuverability and Experimental Validation
Kwangyeol Baek, Chongsung Won and Taeoh Tak, Kangwon National University, Korea

12:30 - 13:30 Lunch

Session 3.3

13:30 - 14:00 Study on Characteristics of Motorcycle Behavior during Braking
Ichiro Kageyama and Yukiyo Kuriyagawa, Nihon University, Japan

14:00 - 14:30 Influence of the Front Suspension on the transient Dynamics of Motorcycles on Braking
Sebastian Risse, KTM Sportmotorcycle AG, Austria

14:30 - 15:00 Motorcycles Dynamic Stability Monitoring During Standard Riding Conditions
F. Cheli, M. Pezzola, E. Leo, A. Saita and T. Ibrahim, Politecnico di Milano, Italy

15:00 - 15:30 Coffee Break

Session 3.4

15:30 - 16:00 Application of the Rigid Ring Model for Simulating the Dynamics of Motorcycle Tyres on Uneven Roads
A.J.C. Schmeitz, S.T.H. Jansen, TNO Science and Industry, The Netherlands, Yoshitaka Tezuka, Makoto Hasegawa and Syunichi Miyagishi, Honda R&D Co., Japan

16:00 - 16:30 Measuring Dynamic Properties of Bicycle Tires
A. E. Dressel and A. Rahman, University of Wisconsin-Milwaukee, USA

FRIDAY, 22 OCTOBER 2010 at ROOM: C

16:30 - 17:00 Closing Ceremony

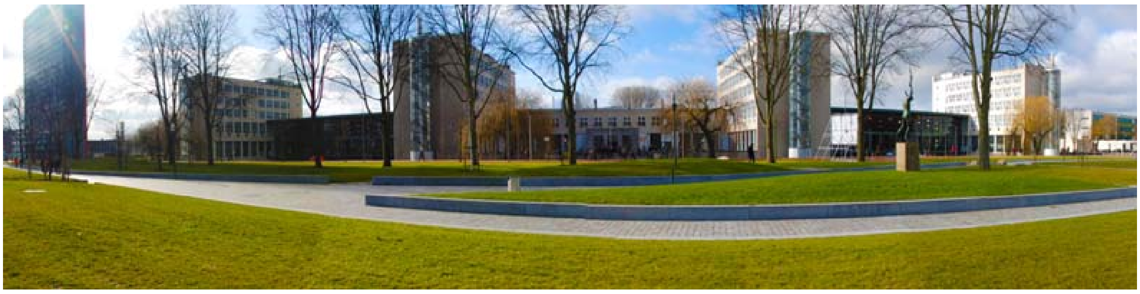
19:00 - 22:00 Boat Trip and Dinner on the Maas in Rotterdam

Venue

The Symposium will take place at Delft University of Technology at the faculty of Mechanical Engineering (3mE):

Faculty Mechanical, Maritime and Materials Engineering (3mE)
Delft University of Technology
Mekelweg 2, 2628 CD Delft – The Netherlands

- Wednesday and Thursday, 20 – 21 October 2010: Room A
- Friday, 22 October 2010: Room C
- Lunches at Room: "Lagerhuys"



Contact

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- Jodi Kooijman 06 5234 0939

Wireless Network

During the symposium you have free wireless network access on the TUDelft congress network.
Wireless network: TUDelft-congress
Network key: awe46

Dinner addresses

- Wednesday, 20 October 2010
  Sport Centrum TU Delft,
  Mekelweg 8 – 10, 2628 CD Delft (Phone: 015 – 2782442).

- Thursday, 21 October 2010
  Proeverij De Waag
  Markt 11, 2611 GP Delft (Phone: 015 – 214 46 00).

- Friday, 21 October 2010
  Spido B.V.
  Willemsplein 85, 3016 DR Rotterdam (Phone: 010 – 275 99 88).

Abstracts

Bicycle and Motorcycle Dynamics 2010
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Bicycle and Motorcycle Dynamics 2010
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20–22 October 2010, Delft, The Netherlands

Rider control of a motorcycle near to its cornering limits

R. S. Sharp
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Abstract

In previous work [1, 2, 3, 4, 5, 6, 7, 8], skilled riding of single-track vehicles has been depicted as involving optimal control with full preview of the path to be followed. Linear optimal preview control theory has been developed and applied to the separate representations of steering and speed control and it has been argued that the full range of feasible vehicle motions can be covered by adaptation of the controls to the running conditions.

General well-ordered vehicle motions are seen as entailing small perturbations from trim (dynamic equilibrium) states. At any particular time, an appropriate trim state is chosen as a reference state for the motions occurring and the linear optimal preview control scheme belonging to that trim state is employed temporarily. Gain scheduling by interpolation allows the controls to adapt to the changing running state.

In the present case, the separate representations of steering and speed control for a high-fidelity and well-documented motorcycle [9, 10] are combined into simultaneous path and speed (x, y, t) control. Optimal throttle, steering and rider-lean controls are generated for trim states with variations in speed and lateral acceleration, and they are used to illustrate how the controls change as the running conditions alter. In particular, as the lateral acceleration increases towards the limit of physical possibilities, the control authority decreases towards zero and these influences are reflected in the optimal controls. Exemplary preview control gains, with sampling interval 0.05 s and 10 s preview time, are shown in figure 1 as functions of lateral acceleration for a fixed speed of 30 m/s. Similar figures show throttle position to x-error gains and cross-coupling effects when the motorcycle is cornering.

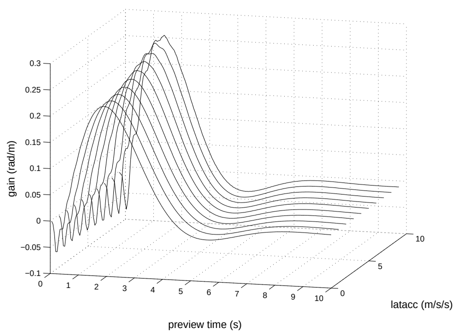

Figure 1. Optimal preview y-error to steer torque gains as functions of lateral acceleration for a speed of 30 m/s, for 10 s preview.

Other new results available comprise frequency-responses of rider-controlled systems, indicating tracking capabilities, and path-tracking simulations. These simulations include a lane-change manoeuvre with speed variation and control adaptation over speed and lateral acceleration by bilinear interpolation, figure 2, and constant speed clothoid manoeuvres up to the lateral acceleration limits. Results to be shown in the paper will be selected from this library.

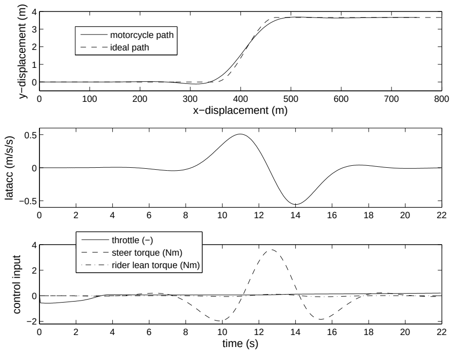

Figure 2. Motions occurring in a lane-change with bilinear gain scheduling by speed and lateral acceleration. The speed starts at 50 m/s, decreases to 30 m/s after 3.5 s and builds up to 47 m/s again from 13 s on.

References

[1] R. S. Sharp, "Optimal linear time-invariant preview steering control for motorcycles", The Dynamics of Vehicles on Roads and on Tracks (S. Bruni and G. Mastinu eds), supplement to VSD 44, Taylor and Francis (London), 2006, 329-340.

[2] R. S. Sharp, "Optimal stabilisation and path-following controls for a bicycle", *Proceedings of the IMechE, Part C, Journal of Mechanical Engineering Science* **221**(C4), 2007, 415-428.

[3] R. S. Sharp, "Optimal preview speed-tracking control for motorcycles", in C. L. Bottasso, P. Masarati and L. Trainelli (eds), *Proceedings, Multibody Dynamics 2007, ECCOMAS Thematic Conference*, Milano, Italy, 25–28 June 2007, Politecnico di Milano, Milano, 16 pp.

[4] R. S. Sharp, "Motorcycle steering control by road preview", *Trans. ASME, Journal of Dynamic Systems, Measurement and Control* **129**(4), 2007, 373-381.

[5] R. S. Sharp, "Dynamics of Motorcycles: Stability and Control", in W. Schiehlen (ed.) *Dynamical Analysis of Vehicle Systems: Theoretical Foundations and Advanced Applications*, Springer, Wien/New York, 2007, 183 - 230.

[6] R. S. Sharp, "Optimal preview speed-tracking control for motorcycles", *Multibody System Dynamics* **18**(3), 2007, 397-411.

[7] R. S. Sharp, "On the stability and control of the bicycle", *Trans. ASME, Applied Mechanics Reviews* **61**, paper 060802, October 2008.

[8] R. S. Sharp, "On the stability and control of unicycles", *Proc. Roy. Soc., Series A*, published on-line 20/01/2010.

[9] R. S. Sharp, S. Evangelou and D. J. N. Limebeer, "Advances in the modelling of motorcycle dynamics", *Multibody System Dynamics*, **12**(3), 2004, 251-283.

[10] R. S. Sharp, S. Evangelou and D. J. N. Limebeer, "Multibody aspects of motorcycle modelling with special reference to Autosim", in J. G. Ambrósio (ed.), *Advances in Computational Multibody Systems*, Springer-Verlag, Dordrecht, The Netherlands, 2005, 45-68.

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Analysis of the Biomechanical Interaction between Rider and Motorcycle by Means of an Active Rider Model

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Abstract

Compared with other vehicles like cars, the mechanical interaction between rider and his motorcycle is much closer. Dynamics of motorcycles is well understood [1] and even complex multibody models of real bikes have been realized [2]. But models which consider the influence of the rider on the dynamics of the man-machine system are restricted to upper body lean and artificial steering torques. To enable the analysis of issues like motorcycle ride comfort, safety and instable ride modes Biomotion Solutions has developed a biomechanical rider model for motor-cycle simulations which is capable of steering the motorcycle by moving the handlebars.

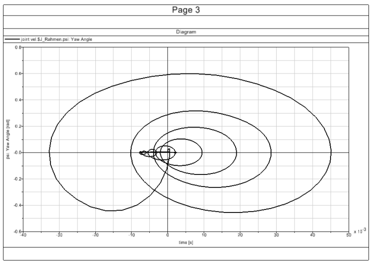
The Human Body Model: Based on anthropometrical data [3] we have implemented a 17 segment full body model. The model consists of 2 legs (foot, shank and thigh), a 3-parted trunk, neck head and two arms (hand, fore-arm, upper arm). Furthermore, each of these rigid bodies is coupled with a so-called wobbling mass which takes into account that human body tissue is not a rigid material. The consideration of wobbling masses [4, 5] is crucial - especially for ride comfort simulations.

Enabling Active Movement: To enable active steering movements we have implemented passive and active actuators into the model. The lower extremities, the trunk and head-neck are stabilized by passive impedances whose parameters are chosen according to literature values [6]. To enable the rider model to control the handlebars, shoulders and elbows are actuated via muscle moment generators. Their input is provided by our driver model controller which uses a look ahead measurement of the track (e.g. [2]) as feedback for the path control. Muscle moments are generated by PID-Controllers in the joints, which take the desired joint space configuration as set value. The heavier a motorcycle is, the more dominant the so-called countersteering with the handlebars is. Consequently we take the cornering via controlling the steering angle into account at first. Appropriate roll angles can be computed by PID-Controllers using as input the lateral track error of the bike's position and the yaw-angle, which describes the difference between the bike's direction and the curvature of the road trajectory. Both strategies can be superposed; PID-parameters have to be chosen carefully to allow for stable operation.

The Motorcycle Model: The motorcycle model is built as a 3d multibody system using SIMPACK as simulation platform. The bike consists of a fork mounted frontwheel connected to the steering axis by a 1 DOF (*degree of fredom*) prismatic joint. The steering is connected by a 1 DOF hinge joint to the frame. The swingarm mounted rear wheel has also 1 DOF of rotation. Parameters for inertia and masses were chosen in agreement with literature. As this configuration came out as too ideal to generate wobbling modes, we added some further DOF e.g. to describe the elastic properties and joint clearance.

Analyzing Safety Issues: Motorcycle and rider are a coupled system whose dynamics emerge from the interaction of both. Experienced riders are reporting possibilities to provoke or to damp down highly dangerous instable ride modes like weave or wobble. We used our rider model to analyze the rider-bike-system near weave mode. Riding with 60 meters per second a transient disturbance moment has been applied at the steering. The bike then showed a short latency time in which a negative damped oscillation showed up which finally led to exponential rising amplitude in yaw angle and to uncontrollable crashing. Furthermore we have analyzed the sensitivity of the weave

phenomenon to the seat cushion parameters ot to total muscular tension of the rider's body. The results predict strong influence of biomechanical factors on ride dynamics.



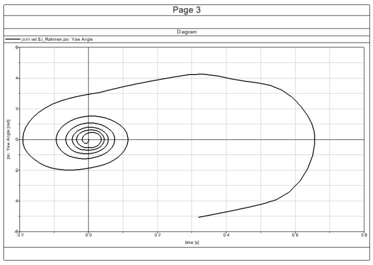
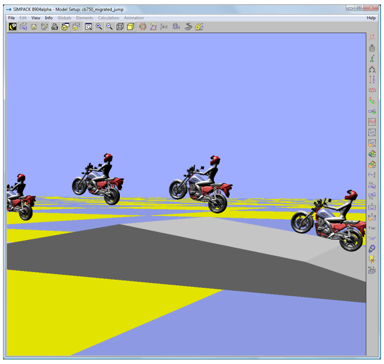


Figure 1. Phase space trajectories of the yaw. The man-machine system becomes unstable if the rider tries to hold the handlebars too tight (right figure).



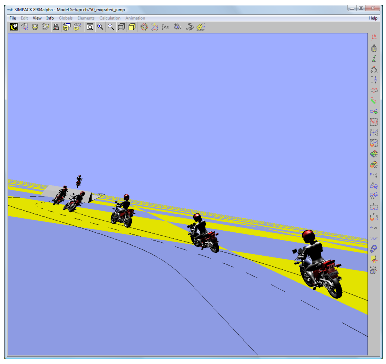


Figure 2. The biomechanical rider is capable of handling tasks like straight running, cornering and even jumping (some animations can be found under www.biomotion-solutions.com/drivermodel).

References

[1] Limebeer, DJN. and Sharp, RS., "Bicycles, Motorcycles, and Models", *IEEE Control Systems Magazine, - imperial.ac.uk* (2006).

[2] Berritta, R., Biral, F. and Garbin, S., "Evaluation of Motorcycle Handling with Multibody Modelling and Simulation", *Proc. 6th Int. Conf. on High Tech. Engines - dinamoto.it* (2000).

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[6] Cossalter, V., Doria, A., Fabris, D. and Maso, M., "Measurement and Identification of the Vibration Characteristics of Motorcycle Riders", *PROCEEDINGS OF ISMA 2006 - dinamoto.it* **1793** (2006).

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Modeling Manually Controlled Bicycle Maneuvers

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Abstract

An ability to predict the handling qualities of bicycles with different physical characteristics remains an important research issue in the study of manual control of single-track vehicles. Numerous factors affect bicycle handling qualities as perceived by the rider, e.g. the dynamics of the bicycle itself, the dynamics of the rider, the control characteristics of the rider, and the manner in which the rider quantifies his/her opinion of the bicycle's handling qualities. The work to be described follows in the footsteps of [6] and [7], and utilizes a human operator model discussed in [2] as applied to piloted aircraft control.

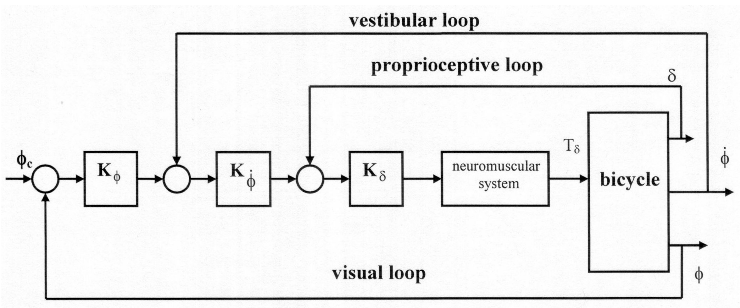
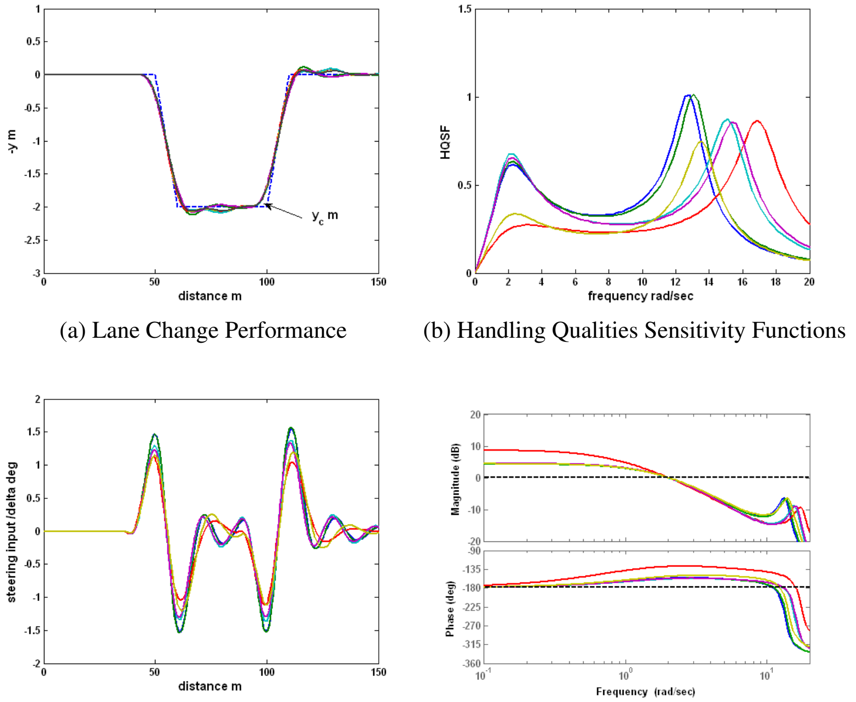


Figure 1: Block Diagram of Rider/Bicycle System

Figure 1 is a block diagram representation of the rider model for roll control of a bicycle, with appropriate sensory modalities noted. As presented in Fig. 1, only three gains and a simple neuromuscular system model parameterize the rider. By way of exposition in this brief abstract, six bicycle models were chosen, five from existing bicycles as parameterized by the second author in [5], and the sixth being the "benchmark" bicycle from [4]. All models are linear and valid for a forward velocity of 5 m/sec. A simple two meter lane change and lane return maneuver was selected as a simple task. Figure 2a shows the maneuver paths of the six rider/bicycles, where the bicycle coordinate shown is associated with the rear wheel contact point. The complete rider model includes two additional outer-loop closures not shown in Fig. 1 (heading and lateral deviation). In addition, a simple model of rider preview is included [1]. Figure 2c shows the rider steering inputs for the task. Figure 2b shows the "Handling Qualities Sensitivity Functions" (HQSFs) for each vehicle. The magnitude of each of these functions has been shown to correlate well with handling qualities ratings for aircraft flight control [2]. As an example here, the two functions with the lowest magnitudes in the frequency range of importance for manual control ($0 < \omega < 10$ rad/sec) are associated with the two bicycle models that are either stable or marginally unstable (one root just into the right-half plane). Finally, Fig. 2d shows Bode diagrams for the rider/bicycle open-loop transfer function for each application, showing that the rider model follows the dictates of the well-known "crossover model" of the human operator [3], appropriate for the dynamics of the bicycles in question. The final paper will include nine bicycle models and maneuvers and a thorough discussion of applying the HQSF to the prediction of bicycle handling qualities.



(a) Lane Change Performance

(b) Handling Qualities Sensitivity Functions

(c) Rider Steering Inputs

(d) Bode Diagrams of Rider/Bicycle Open-Loop Transfer Functions

Figure 2: Linear simulation results and comparative metrics for six bicycles.

References

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A dynamic inversion approach to motorcycle trajectory exploration

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Abstract

The dynamics of two-wheeled vehicles is extremely rich. It exhibits, to cite the most important features, unstable dynamics, underactuation, mechanical symmetries, countersteering ("steer left to turn right"), a wide range of operating conditions, and speed dependent transitions from instability to stability regions. From a control theory point of view, the maneuvering control problem for these mechanical systems still poses an interesting number of challenges. Our interest, in this context, is the the development of control strategies for the exploration of two-wheeled vehicle dynamics in virtual prototyping studies for high performance motorcycles [1, 2].

Theoretical investigation on simplified motorcycle models [3, 4] suggests that it seems possible to *uniquely* parameterize any *upright roll* motorcycle trajectory of *infinite time extent* (i.e., $t \in \{-\infty, +\infty\}$) by means of the corresponding flatland trajectory traversed by the rear wheel contact point on the ground. Inspired by this idea, in [5] it was shown that the trajectories of a nonholonomic motorcycle can be parametrized by the trajectories of a nonholonomic car. Given a time indexed plane curve (including appropriate derivatives), a state-control trajectory of a nonholonomic motorcycle that (approximately) implements a desired plane trajectory was obtained.

In this paper, we introduce a *rigid motorcycle model* which captures many important aspects of real motorcycle dynamics including sliding and load transfer. This model is used to demonstrate a dynamic inversion procedure which maps a desired *flatland* trajectory into a corresponding (state-control) trajectory for the rigid motorcycle model. We assume that the rider is firmly attached to the main body of the vehicle, leaving the discussion on the important control and configuration effects offered by rider motion for future investigation.

In [5], the dynamic inversion was obtained *embedding* the motorcycle dynamics into an extended control system containing a sufficient number of additional *artificial controls* to make the system trivially controllable. We follow the same approach here, obtaining a fully actuated mechanical system starting from the underactuated rigid motorcycle model.

Due to the trivial controllability of the extended control system, we can can make it follow any desired velocity-curvature profile. Subsequently, the effect of artificial controls can be optimized away, obtaining a suitable motorcycle trajectory consistent with the requirements.

To solve the optimal control problem which is required to remove the effect of the artificial controls, we use the *projection operator approach* detailed in [6]. This optimization approach exhibits second order convergence rate in a neighborhood of the solution, being a generalization of Newton method in the infinite dimensional setting. We remark that the dynamic inversion procedure we propose in this work is suitable to be extended to more detailed motorcycle models, where, e.g., steering torque input, nontrivial steering geometry, and rider motion are considered.

In the paper we first introduce the *sliding plane motorcycle* (SPM) model, a rigid motorcycle model whose dynamics will be inverted using optimization. Then, the mechanical symmetries of the SPM model are described, and its dynamics are formulated in body coordinates. The *equilibrium manifold* for the SPM model, whose existence derives from the presence of mechanical symmetries, is analysed. We then present the (approximate) dynamic inversion strategy, that we term *lifting*, that "lifts" a flatland trajectory to a full state trajectory for the SPM model, using an optimization strategy. Effectiveness of the method is shown by means of numerical simulations. The paper will also include some discussion on the use of the lifted trajectory to design a maneuver regulation controller (virtual rider) to make a multibody motorcycle model follow a desired curvature-velocity profile [2].

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Speed-Adaptive Path-Following Control of a Riderless Bicycle via Road Preview

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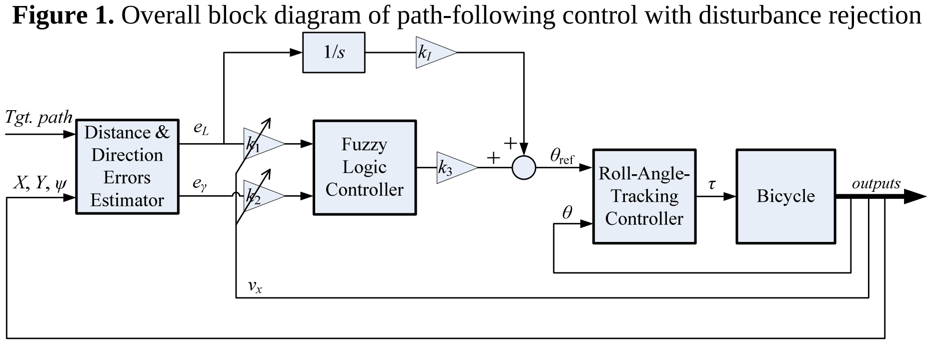
Abstract

In this study, a genetic-fuzzy control system is used to control a riderless bicycle where control parameters can adapt to the speed change of the bicycle, with the control structure shown in Figure 1. This controller consists of two loops: the inner is a roll-angle-tracking controller which generates steering torque, and the outer is a path-following controller which generates the reference roll angle for the inner loop. The equations of motion of a bicycle with constraints of rolling-without-slipping contact condition between wheels and ground are developed. The inner loop is controlled by a sliding-mode controller (SMC) on the basis of a linear model obtained from the non-linear one via system identification. By defining a stable sliding surface of error dynamics and an approriate Lyapunov function, the bicycle can reach the roll-angle reference in a finite time and follow that reference without chattering. Three control parameters for SMC can be tuned to adjust the performance of this controller.

Figure 1. Overall block diagram of path-following control with disturbance rejection



The outer loop determines the proper roll-angle reference by using a fuzzy-logic controller (FLC). For path-following control, the preview distance error e_L and the preview direction error e_γ are considered simultaneously. The preview errors are determined following the scheme in Figure 2. First, the preview point P is determined at a preview distance ahead from the reference point of the bicycle, where the preview distance is given by $L_{\text{pre}} = v_x \times T_{\text{pre}}$, with v_x is the forward speed and T_{pre} is a constant preview time. Once the preview point is determined, the preview distance error e_L will be the signed shortest distance from the preview point to the defined path. The preview direction error e_γ is the angle between the axis of the bicycle and the direction vector (or tangent vector) of the path at the nearest point H on the path to the preview point. This scheme is adopted from a study of Sharp [1] with a slightly difference in determining the target point, to avoid the situation in which the target point might be undefined.

From the preview distance and direction errors, the FLC generates the reference roll angle. To make the controller speed-adaptive, the controller parameters including the scaling factors and the deforming coefficients of the fuzzy membership functions are tuned by using genetic algorithms [2] for different speeds in the operational speed range, and the gain scheduling technique is used. Figure 3 shows the results of a simulation in which the bicycle is controlled to follow a sinusoidal path defined by the equation $Y = 2.5\sin(2\pi X / 50)$ from an initial lateral error of 2m. At the same time, the speed is controlled to increase from 5 at the initial position to 30km/h at a longitudinal position of 50m, then decrease back to 5km/h at the longitudinal position of 100m. It can be observed that the bicycle approaches the target path in the first 10m. After that, the absolute value of the tracking error is kept under 0.07m in this simulation.

From the riding experience, lateral wind force and disturbance usually affect the maneuverability of the bicycle and cause stability problems. This problem can be solved by adding an integrator and an adjustable gain k_I between the preview distance error e_L and the reference roll angle. A simulation with this controller at a speed of 10km/h is shown in Figure 4, where the value of k_I is 15. As soon as the disturbance is applied, the tracking error increases. However, by the effect of the integrator, an additional term is added to the reference roll angle. This can help the bicycle be pulled back to the direction making the tracking error reduced. When the disturbance releases and the equilibrium point changes, a similar process also happens, but in the inverse direction. The overshoot is about 0.3m, and the transient longitudinal distance is about 7m. The tracking error the simulation without the integrator is also plotted for comparison, showing that it approaches a steady-stead error of nearly 0.8m during the appearance of the disturbance.

Figure 2. Error estimation for path-tracking control with preview

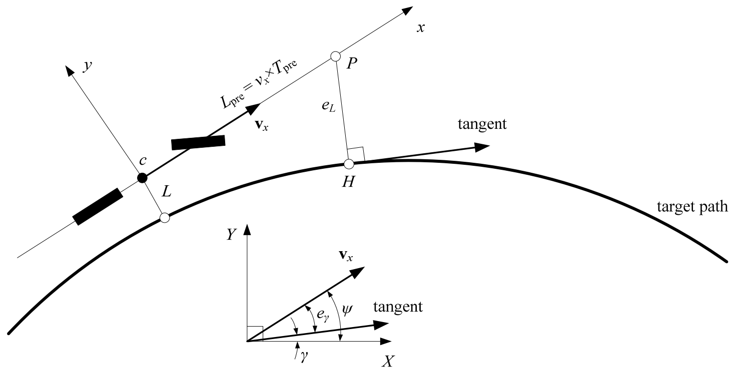


Figure 3. Path-tracking control at varying speed

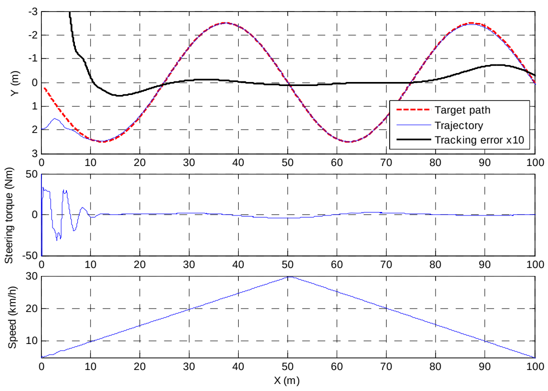
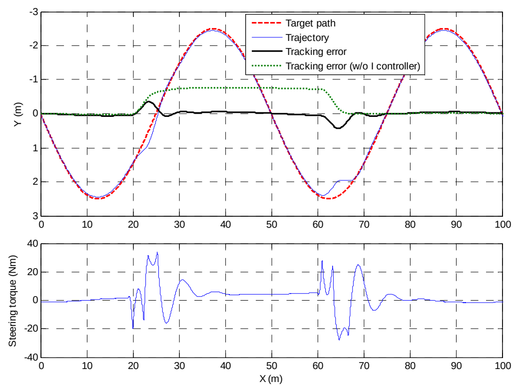


Figure 4. Path-tracking control with disturbance rejection at a speed of 10km/h



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Proposal of personal mobility vehicle based on stabilization control of Two-Wheel Steering and Two-Wheel Driving

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Abstract

Mobility in a city is an important part of our life. For the sustainable development, there should be a mobility which is friendly for human and environment[1]. Recently, as a new mean, a personal mobility vehicle (PMV) which is compact and convenient attracts attention[2, 3]. As a PMV, the following features should be considered. 1. To make short range transport be efficient and comfort by using low-impact actuator, 2. To be used safely in non-exclusive space for pedestrians, 3. To be enough compact to achieve seam-less transit with existing public transportation.

A bicycle is the one of PMV, however, it becomes unstable at low speed[4]. The smaller the tire diameter becomes, less stable it becomes. The authors propose a stabilized vehicle with two-wheel steering and two-wheel driving(2WS/2WD) that solves the problem. The stability of the conventional bicycle has been discussed in a lot of papers, however, the study about the stability and control of the 2WS/2WD bicycle has not been investigated so much. In this paper, first the stabilization of 2WS/2WD bicycle is shown and then the new bicycle based on the simulation is suggested.

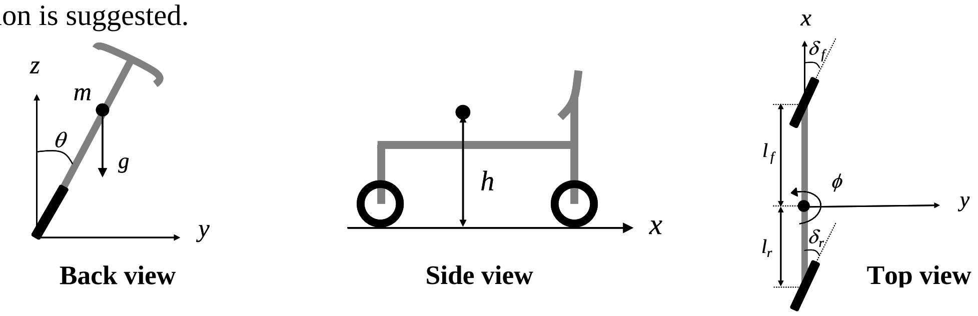

Figure 1. Model of bicycle

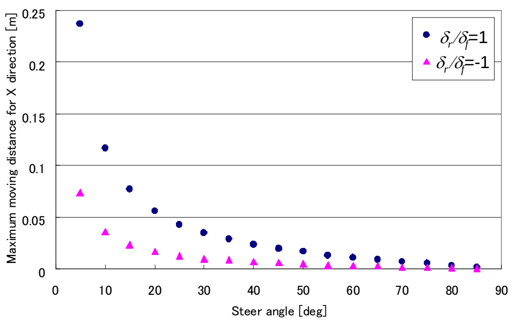

Figure 2. Maximum moving distance

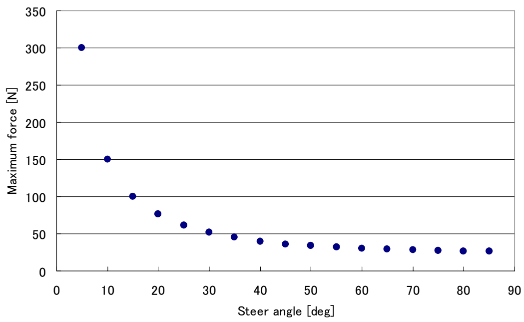

Figure 3. Maximum driving force

The model of the bicycle is shown in Figure 1. The authors propose the stabilization of the bicycle using driving forces and design a controller using linear-quadratic control theory. The prior analysis made clear that the same steer angle between the front wheel and rear wheel makes the bicycle the most stable at low speed.

Figure 2 shows the plot of the maximum moving distance for *x* direction shown in Figure 1 and Figure 3 shows the maximum driving force. It is shown that by increasing the front and rear steering angle, the required moving distance for *x* direction for stabilization becomes smaller. The condition of the steering angle 90 degree corresponds to the parallel two-wheel vehicle[2].

Bicycle mode **Parallel two-wheel mode**

Figure 4. Concept of PMV

Finally, a new PMV for this result is proposed. The concept of PMV that authors propose consists of two modes, the bicycle mode and the parallel two-wheel mode shown in Figure 4. These two modes are convertible each other. It will be an effective mobility with low energy consumption by switching between two modes.

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Simulation and Control of the *Anaconda*

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Abstract

The *Anaconda* project consists in designing an articulated in-line polycycle propelled by man power and able to follow any winding road.

The Anaconda (Figure 1) consists of a Head Module (HM), which resembles the traditional bicycle, followed by several Pedal Modules (PM) connected to each other by spherical joints. As the purpose is to allow a maximum number of modules, it appears that each module must be fitted with a handlebar in order to control itself its equilibrium. On the contrary, the motricity and the brake system should be centralized. This aspect will not be considered in this paper which will focus on the stability of the Anaconda and try to demonstrate that its handling is possible.

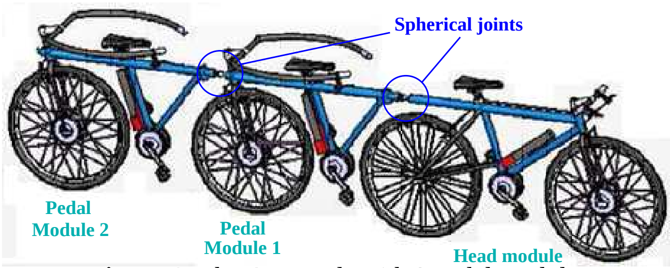

Figure 1. The *Anaconda* with 2 pedal modules

The dynamics of the Anaconda has been modeled according to the multibody theory, with the help of the EasyDyn framework [1], based on the minimal coordinates approach. Three models have been developed: the HM alone (5 bodies and 10 dof), the PM alone (4 bodies and 6 dof) with the attachment point constrained to follow a straight line, and finally an Anaconda with a HM and 2 PM (13 bodies and 22 dof). In all models, the rider body lean is frozen and tires are considered through contact forces calculated according to the model of the University of Arizona [2].

To stabilize the first two systems, the effect of the biker is introduced through a controller whose output is a torque applied on the handlebar as usually considered in bicycle models [3]. To design the controllers, systems are linearized about a stationary state where the module is ridden in straight line, at the velocity of 20 km/h on a flat level surface. Only the out-of-plane behavior is retained, the number of concerned dof coming down to 4 for the HM and 3 for the PM respectively: the lateral displacement (for HM), the yaw and the roll angles of the module's frame and the steering angle. Optimal state feedback controllers have been designed according to a LQG approach. So far, no observer has been included in the model and all state variables are assumed to be available.

The optimal controllers have then been successively tested on the complete nonlinear models of the HM and PM alone. It is observed that the torques necessary to control the vehicle fit to the human possibilities. Moreover, in order to demonstrate the manoeuvrability of the complete polycycle, the controllers established on the independent modules have been put on the complete model and lead to a stable behaviour. For the purpose of illustration, Figure 2 shows the results obtained during a lane-change. This result is encouraging as it shows that the equilibrium skills acquired during the learning process on a single module contribute to the global equilibrium. On another side, the rideability index describes by Seffen et al in [4] was evaluated for some Anaconda models with the amount of the PM up to 9 ; and as we can except this index increases with the PM amount.

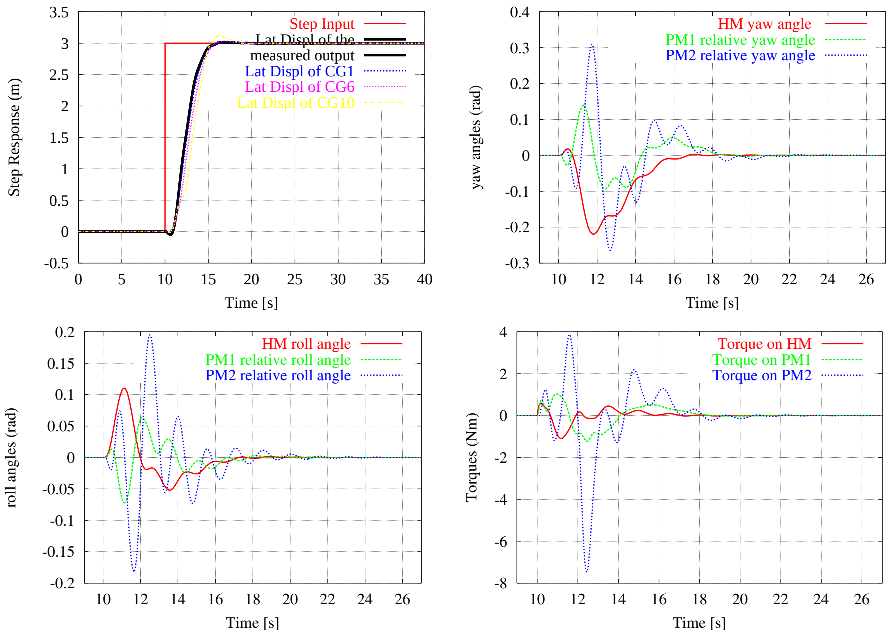

Figure 2. Time histories of modules motion (lateral displacement, yaw and roll angles) and steering torques

In parallel, a prototype with a HM and 2 PM has been built (Figure 3) and it appears that after some training, the bikers are able to follow usual trajectories.

Figure 3. An *Anaconda* prototype

In this work, a multibody model of the Anaconda has been established, from which its manoeuvrability can be hopefully considered as realizable by standard human people. First tests on a prototype with 2 PM modules have confirmed this statement. The future prospectives are to validate the obtained controllers with respect to the human behaviour, so as to use the model to optimize the mechanical design of the polycycle in terms of stability and manoeuvrability.

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Intrinsic stability of a classical monocycle and a generalized monocycle.

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Abstract

In the real world, a fast-rolling coin or child's hoop exhibits asymptotic stability in lean and in heading rate. That is, after being perturbed, it returns to straight upright rolling, with exponentially decaying deviations (as long as the forward rolling rate remains high).

However, for an idealized model of a dissipation-free rolling wheel, such stability is not predicted by the linearized dynamic equations. As shown in many textbooks (*e.g.,* Greenwood [5]), above a critical speed, the perturbation eigenvalues all have zero real parts, indicating a constant (*i.e.,* undamped) oscillation superimposed on a steady (*i.e.,* unstraightening) circular trajectory. In fact, such neutral stability is the most that can be achieved by non-dissipative systems with fore/aft symmetry: true stability is prohibited, since the time reversibility of any trajectory means that decaying motions imply the simultaneous existence of growing motions (as pointed out by Meijaard, *et al.* [7] and proved in Bloch, *et al.* [1]). The observed stability of real-world disks or hoops is evidently a result of dissipation, such as tire spin friction, that is not included in most rigid-body dynamics analyses.

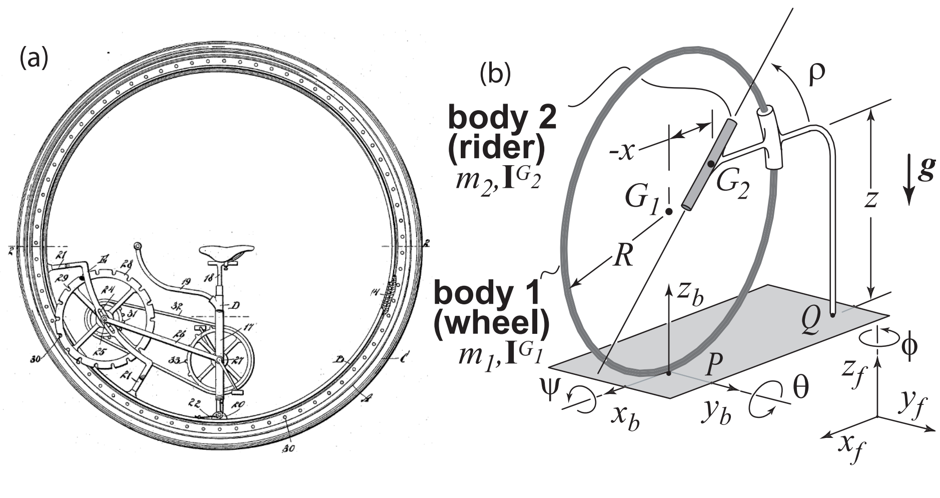

Figure 1. (a) An early classical monocycle from patent No. 611,534 by V. D. Venable, 1898. (b) An un-driven generalized monocycle model showing parameters and configuration variables; it reduces to a classical monocycle for $x = 0$ and $z < R$. Note that, in this view, $x < 0$. The subscripts 'f' and 'b' refer to the fixed and bank frames, respectively. The massless rigid supporting structure of Body 2 is in white.

We focus our attention on the *monocycle* (see Fig. 1(a)), a laterally symmetric wheel augmented by a mass distribution representing a person within it, which is steered by lateral body shifting. We wish to investigate whether an intentional departure from fore-aft symmetry of a laterally symmetric rider could provide the kind of strong dynamic stability *not dependent on dissipation* that is observed for a typical uncontrolled bicycle [7]). Here, in order to further investigate the effects of asymmetric mass distribution, we extend the analysis to accommodate fore/aft displacements of the rider center of mass, which requires a second support point (such as a caster wheel) in order to maintain the rider's position. This we call the 'generalized monocycle' model.

We show that a generalized uncontrolled, conservative model of a monocycle consisting of a rigid body rider plus a wheel that reduces to a classical monocycle can have asymptotically stable steady, vertical, straight-line motions. The model predicts enhanced stability with increasing amounts of fore/aft asymmetry in the mass distribution. We obtain the results using both linearized stability analyses and numerical solutions of the full nonlinear governing equations. The model's arbitrary rider center of mass position within the wheel permits greater stability than for the classical design. The results corroborate and extend those that can be extracted from the century-old analysis by Carvallo [2].

Three main features combine to produce stability: (1) nonholonomy, (2) a fast enough rolling wheel with non-negligible mass (nonholonomy and an adjustable rider mass distribution are not enough to produce stability), and (3) proper mass distribution. For comparison, some examples of other mechanical systems that are also constrained to motions on a flat plane (or small downhill slope), are statically unstable in all configurations, yet can be dynamically stable at a local potential energy maximum without control are: tops; a riderless bicycle with massless rigid wheels [7]; a skateboard with a properly positioned rigid rider [6]); a rigid regular polygon rolling downhill [3];and, some simple rigid-body models of walking [4]). These systems gain their stability by having some mixture of fast-spinning parts, dissipation, nonholonomic kinematic constraints, intermittent contact, special mass distributions, and linkages that connect internal degrees of freedom. An important aspect of the smooth systems is that Hamiltonian (conservative, holonomic systems) *cannot be asymptotically stable* while conservative nonholonomic systems *can* [8].

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Research activity of the mOve research group in the field of electronic motorcycle control: past and future

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Abstract

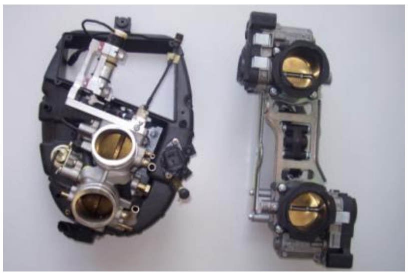
This contribution has the goal of presenting and briefly discussing the research activity developed by the mOve research group at the Dipartimento di Elettronica, Politecnico di Milano (TU Milan) in the field of electronic control systems for motorcycles, in the last five years.

The interest in electronic control in motorcycles boosted in the last few years, both in academic and in industrial research centers, since it is particularly challenging from many point of views:

- motorcycle dynamics are (much) more complicated than four-wheel vehicles dynamics
- the sensitivity of the average motorcycle driver to dynamic performances of the vehicle is very high
- severe constraints of cost, space and weight
- severe instability issues

Motorcycle control hence is far from being a simple re-casting of “traditional” control systems developed for automotive applications.

Figure 1. Examples of full-by-wire body throttles for motorcycles.



The research activity in this field has been developed along many streams:

- suspensions electronic control (in particular semi-active control)
- steer-damping electronic control
- traction control (with and without ride-by-wire systems)
- braking electronic control
- stability control (still the most challenging and unexplored sub-system)
- electronic control of gear-shift
- virtual dashboard (for data logging and controller tuning)

Another interesting research stream is related to light electric vehicles: light vehicles (light motorcycles) are the best candidates to be fully electrified, for short and mid-range personal mobility.

In the presentation all the research activity developed by the research group in the last years along such research stream will be briefly presented and outlined. Also, future direction of research will be discussed.

Figure 2. A motorcycle equipped with electronic suspension on a test-rig.

Figure 3. A laser sensor for lean-angle estimation.

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Motorcycle State Estimation for Lateral Dynamics

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Abstract

The motorcycle lean (or roll) angle development is one of the main characteristics of motorcycle lateral dynamics. Control of motorcycle motions require an accurate assessment of this quantity and for safety applications also the risk of sliding needs to be considered. Direct measurement of the roll angle and tyre slip is not available, therefore a method of model-based estimation is developed to estimate the state of a motorcycle. This paper investigates the feasibility of such a Motorcycle State Estimator (MCSE). A simplified analytic dynamic model of a motorcycle is developed by comparison to an extended multi-body model of the motorcycle, designed in Matlab/SimMechanics. The analytic model is used inside an Extended Kalman Filter (EKF). Experimental results on an instrumented Yamaha FJR1300 motorcycle show that the MCSE is a feasible concept for obtaining signals related to the lateral dynamics of the motorcycle.

Model development

In this feasibility study, it is chosen to use a model-based estimation algorithm: the Extended Kalman filter. The basis of such algorithms is formed by an analytic representation of the observed system, in this case a motorcycle. The objective is to achieve a real-time application and the resulting model has been a challenging compromise between accuracy and computational demand. In this respect, the internal estimation model has been set up to describe the lateral dynamics at constant velocity only. To find out which degrees of freedom should be modelled, first an extensive multi-body representation of the actual motorcycle has been set up in SimMechanics. The tyres have been modelled using Delft-Tyre [1], such that realistic tyre behaviour (e.g. tyre cross-section, combined slip) is accounted for. This multi-body model is validated using experimental results obtained from test drives with an instrumented Yamaha FJR1300 motorcycle. Several model configurations, which could potentially be used inside an EKF, have been compared with the multi-body model in order to select the concept for implementation in the EKF. The selected motorcycle model has 7 degrees of freedom being the forward velocity (v_x), lateral velocity (v_y), yaw rate (r), roll angle (ϕ), steering angle (δ), front wheel angular velocity (ω_f) and rear wheel angular velocity (ω_r). Linear tyre slip characteristics are used. For this model, the equations of motion are derived using the Lagrange energy equations. The model is similar to the motorcycle model in [2], however, the motorcycle model in this paper will not be linearised for the roll and steering angle. The result is a non-linear model, which is valid for all roll and steering angles:

$$\begin{array}{rcl} \dot{x} & = & f(x,u) \\ z & = & h(x) \end{array} \quad \text{with:} \quad \begin{array}{rcl} u & = & \left[\begin{array}{ccc} \delta & \omega_f & \omega_r \end{array} \right]^T \\ x & = & \left[\begin{array}{ccccc} v_x & v_y & r & \dot{\phi} & \phi \end{array} \right]^T \\ z & = & \left[\begin{array}{ccc} a_x^s & a_y^s & \dot{\psi}^s \end{array} \right]^T \end{array} \tag{1}$$

Here, the sensor equation z contains all sensors which are used by the EKF: longitudinal a_x^s and lateral acceleration a_y^s and yaw rate $\dot{\psi}^s$. Note that due to the model concept the state estimator is valid only for relatively constant forward velocities. Furthermore, as linear tyre behaviour is implemented without, extreme sliding situations are out of the validity region as well.

State Estimator setup and results

Because the internal model is non-linear, the Motorcycle State Estimator (MCSE) is set up using an Extended Kalman Filter method [3]. TNO has adapted this method earlier in its Vehicle State Estimator (VSE) for passenger cars in order to account for non-linear tyre behaviour and as a result, the VSE gives reliable lateral slip estimation over the full operating range [4, 5]. The MCSE algorithm is written in C-code and compiled onto a real-time platform, dSPACE MicroAutobox, which is located in the top case of the experimental motorcycle. The MCSE runs with a sample frequency of $100\,Hz$. As an example, the experimental results for a slalom manoeuvre are presented in Fig.1. From the results in Fig.1, it can be seen that the roll angle is estimated well.

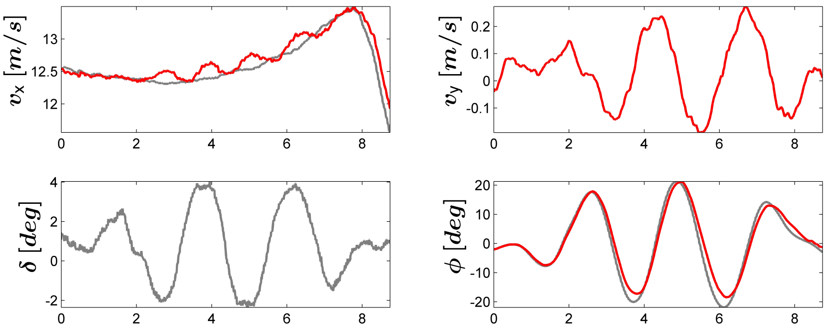

Figure 1. Experimental results of the MCSE: forward velocity v_x, lateral velocity v_y, steering angle δ and roll angle ϕ. Estimated signals are red, reference measurements are grey.

Conclusion

An Extended Kalman Filter has proven to be a feasible way for creating a Motorcycle State Estimator for lateral motions with sufficient accuracy and practical computation demands. The accuracy of the MCSE is related to the internal model description of the motorcycle and its validity range. The MCSE concept has been evaluated for roll angles up to 40 degrees. To improve the performance for a wider application range, future development should focus on extending the internal motorcycle model for longitudinal behaviour and modelling of more realistic tyre behaviour.

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On Linear-Parameter-Varying Roll Angle Controller Design for Two-Wheeled Vehicles

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Abstract

In the past several years, strong efforts have been put forward to derive accurate mathematical models of two-wheeled vehicles [1, 2]. Accurate simulation of two-wheeled vehicle dynamics is often not enough; a model of the driver is also required to study the response on the vehicle during demanding maneuvers.

The literature on two-wheeled vehicle rider models is recent [4, 7, 8]. In most cases a two-layer controller is adopted: an *external* control law computes the control input that would track the reference ground trajectory, and an *inner controller* stabilizes the dynamics. In [5, 6], an extra step is added where an optimal trajectory is computed. The *Optimal Maneuver Method* is used to compute the reference ground path and speed to be followed. The reference optimal trajectory is then stabilized using two independent loops for controlling speed and lateral deviation. Currently the optimization phase, because of computational limitations, can be carried out only on a simplified model. To simulate the maneuver on the complete model, the optimal maneuver is computed on the simplified model and then the inner stabilizing loop is used to track the reference on the complete model. This approach is successfully applied in many conditions; but as the maneuvers become more extreme (with hard accelerations and high lean angles) the inner PID controllers cannot track the reference in a satisfactory way.

The scope of the present paper is that of improving the above internal controller by using a gain-scheduled roll-angle controller. Accordingly, it will be assumed to have an optimal roll angle reference.

By linearizing the complete nonlinear model around different values of forward velocity and lateral acceleration, it can be noted that the transfer function from steering torque to roll angle is subject to a strong variability. The velocity mainly affects the weave and wobble modes and the lateral acceleration affects the low frequency gain. A fixed structure controller has to compromise the achievable performance in order to cope with the time-variability of the system. Given these premises, a Linear-Parameter-Varying (LPV) approach [3] represents a good solution as it provides tools to design scheduled controllers which are guaranteed to retain stability and performances.

Two controllers are designed: a fixed structure $\mathcal{H}_\infty$ controller and an LPV controller. The use of

the $\mathcal{H}_\infty$ design tools enables a fairer comparison as the two controllers optimize the same cost function. The LPV controller is designed using a grid-based technique; in particular a dual grid synthesis is employed in order to improve the numerical properties of the final controller without increasing the computational complexity.

Figure 1 plots the responses of the two controllers to a roll angle reference that simulates high speed cornering. The motorcycle enters the corner at constant speed, when the apex of the corner is reached the motorcycle is accelerated to full throttle. A maximum lean angle of 60° is reached. The LPV controller successfully negotiates the maneuver whereas the fixed structure controller cannot stabilize the system during the acceleration phase.

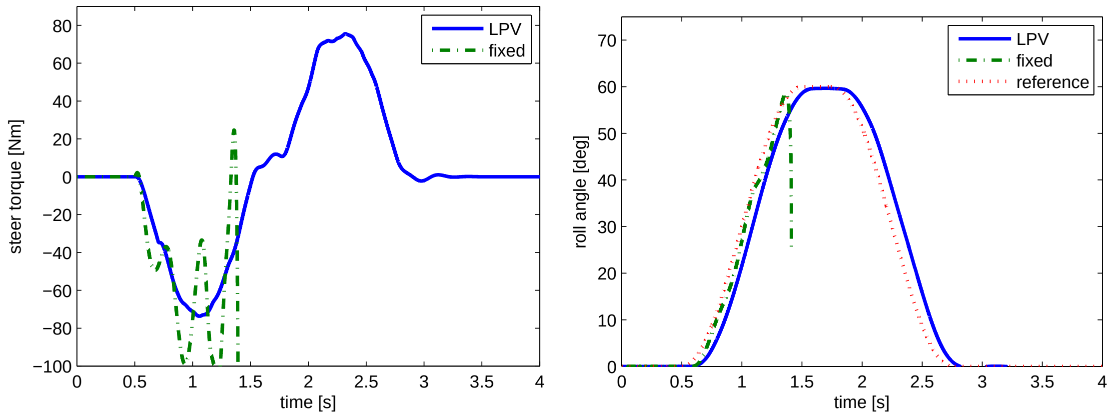

Figure 1. Simulation results of a cornering maneuver carried out with the two discussed controllers. Applied steer torque (left) and roll angle (right).

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Dynamic Model of a Bicycle with a Balancer and Its Control

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Abstract

Research on the stabilization of bicycles has been gained momentum over the last decade in a number of robotic laboratories around the world. Modeling and control of bicycles became a popular topic for researchers in the latter half of the last century. The bicycle literatures are comprehensively reviewed from a control theory perspective in [1], which also describe interesting bicycle-related experiments. A. L. Schwab et al. [2] developed the linearized equations of motion for a bicycle as a benchmark and it is suitable for research or application. It is well known that the control of bicycle with steering at zero or slow linear velocity is very difficult. Thus, we are interested to stabilize the bicycle with the balancer that it can allow us to control the bicycle at zero or slow linear velocity. In this paper, we present the development of a bicycle with a balancer and the balancing control in our laboratory. The first generation of the bicycle with balancer was developed by M. Yamakita in 2005 [3] by using Lagrange dynamic equations and the balancing control used an output function which is defined by an angular momentum and the new state function is controlled to zero. We reported the experimental study of automatic control of bicycle with a balancer in 2006 [4] and in 2008 [5]. In order to control the bicycle in narrow place, we introduced acrobatic turn via wheelie motion that it can allow the bicycle turn on the back wheel at zero linear velocity [6]. This system is still in the development of experiment and we will report the results in the future.
In addition to extending those results to balancing the bicycle, we propose a new balancer configuration for stabilizing of an unmanned bicycle that it shows in Fig. 1 [7]. The balancer can be

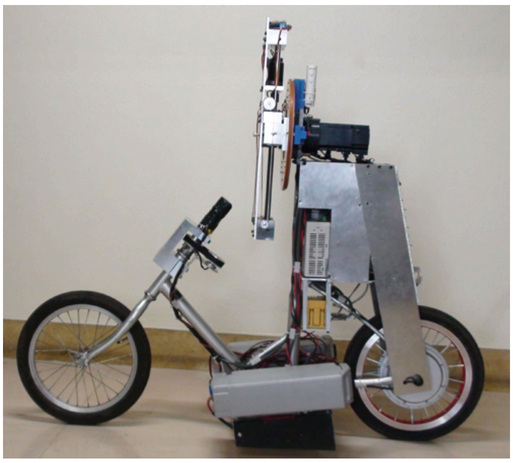
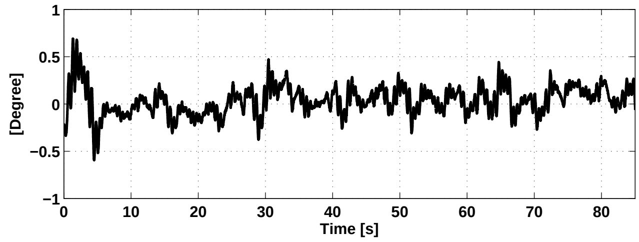


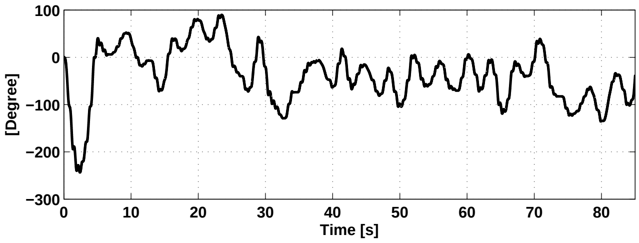
Figure 1. Bicycle with flywheel balancer hardware.

configured as a flywheel mode or a balancer mode by shifting the center of gravity of the balancer. This balancer configuration is changed according to the situation of the bicycle system, which corresponds to the change of the dimension of the system. The balancer is configured as a flywheel, when disturbances to the system are large, and it will switch to the balancer when the position of the center of the gravity should be shifted. Stabilizing bicycle with the flywheel has better performance than the balancer but it cannot control to shift the bicycle angle to track the desired value,

unlike the balancer which can do this motion. The simplify dynamic model of the bicycle with the flywheel balancer model is derived based on an inverted pendulum model. We consider the bicycle as a point mass with two wheels contacting with the ground, and we consider the bicycle and the balancer as a two link system where the first link is the bicycle body with steering and the second link is the balancer. The details of a bicycle with flywheel balancer was presented in [7]. We will report the experimental development in this paper. Figure 2 shows the experimental results of bicycle stabilization with the flywheel balancer.



(a) Roll angle α



(b) Balancer angleβ

Figure 2. Bicycle stabilization with the flywheel balancer

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The balance of walking and bicycling

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Abstract

Our research on human walking and on walking robots has been largely inspired by the passive stability property of many bicycles. Similarly, perhaps the comparison with walking might shed light on the control of bicycles. What are the relations between the both the passive stability and controlled stability of walking and bicycling?

A bicycle is a complex mechanism that can balance itself. A human skeleton is a complex mechanism, maybe it can balance itself. And so, at least to some extent, it seems to be. In 1988 Tad McGeer discovered that walking robots need not be controlled (e.g. [1]), at least in two dimensions. Inspired by McGeer's results, and also by the analogy with bicycles, we have further explored the passive aspects of the stability of walking[**?**, **?**, **?**, **?**, **?**], extending the results to three dimensions. Similar research on walking has also been done here in Delft[].

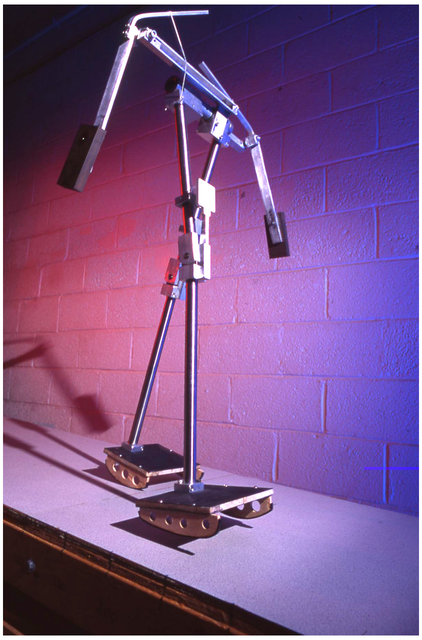

Figure 1. A passive walking robot made at Cornell. It walks and balances itself. No computers, no sensors, no motors. Power is from gravity. The inspiration came largely from the self-stability of bicycles. Videos of this and other passive walking machines will be shown. Photo by Hank Morgan.

That bicycles can be both conservative (neglecting air, rolling and bearing friction) and have asymptotic stability comes from the non-holonomic rolling constraint. Walking is also non-holonomic, but in an intermittent sense[**?**]. It is still not clear whether the asymptotic stability of passive walking robots depends on this non-holonomicity or on collisional dissipation, or both.

How do people balance bicycles? A plausible partial answer, based on the passive stability bicycle results, is that people learn to let the bicycle balance itself. And so it might be with human walkings. In one way of thinking a person is flesh that is riding on the bones. Perhaps, based on the passive robot results, people, at least in part, learn to let the skeleton balance itself.

But certainly their is an active aspect of balance, both for walking and bicycling. People balance bicycles that have no passive stability. And passive robots (and simulations) are not nearly robust enough, to explain why people don't fall very often. A typical passive robot will fall down if someone in the room merely imagines a disturbance.

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Critique of Assumptions Underlying Bicycle Handling Research

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Abstract

Stabilization / maneuvering may be harder to understand for bicycles than for other vehicles with roll freedom, such as airplanes or motorcycles. Since rider mass far exceeds bicycle mass, difficult-to-model corporeal flexibility dramatically reduces inertial reactions; and lagging postural control becomes an important part of the dynamics. The added degrees of freedom of rider deformations, and associated internal-variable postural control systems, not only increase the complexity of the system, but provide multiple avenues by which a skilled rider can control a bicycle (or an unskilled rider may disturb it). Furthermore, powerful pedaling efforts impose large forces on the steering mechanism. All the additional systems and qualities involved in bicycle control are inevitably subject to human inconsistency, adaptation, and fatigue.

The purpose of this paper is to call attention to some important ways in which ridden bicycles may differ from the models so far used to understand them. It is primarily ‘philosophical’, involving consideration of what makes an appropriate model, and what are meaningful questions to pose, rather than matching an assumed model to data. Unfortunately, some of the tentative conclusions offered here are quantitatively based on a simple rigid model [1] already known to be inadequate, so these will have to be revisited in future.

The perspective of this author is that bicycling is widely accepted only because it is normally a very low-bandwidth task. This could be in part because the steering adjustments necessary to keep the bicycle upright largely occur “instantaneously” through the intrinsic dynamics of the front assembly, rather than through rider sensing of roll and responsive control of steer angle (which are delayed by sensing and actuation lags.) To benefit from those intrinsic dynamics, the steer angle must be made a degree of freedom (the well-known “torque control” rather than “angle control”); and the uncontrolled (open loop) bicycle must possess near-stability. There may also be some involvement of upper-body leaning.

For a *rigid-rider model* in hands-free operation, at all but the lowest speeds, the system typically has several stable eigenvalues that rapidly damp any steering transients; plus a near-zero eigenvalue that allows the roll to maintain a (slowly changing) value with no steering input. Those hands-free eigenvalues mean that any open-loop steer torque (or impulse) quickly produces a ‘settled’ roll rate (or angle), absolving the rider of any need to monitor or control the roll angle in detail. And most cycling takes place on wide paths, requiring adjustments only occasionally (i.e., every few seconds). Then, typical path-following or stabilization actions are only needed at frequencies well below any estimate of maximum human-controller bandwidth [2, 3].

To convince oneself of this perspective, it may be helpful to explore an atypically *difficult* riding task – staying on a narrow (0.1 m) painted line – where monitoring / reacting must be both precise and rapid. This is fatiguing and quickly leads to a degradation of performance. It is suspected that similar difficulty will be experienced when riding a bicycle lacking helpful front-assembly dynamics (reduced mass, trail, and spin momentum, plus Coulomb friction), because it too will demand constant rider sensing and input to prevent falling.

Within the above context, the paper attempts to discuss the following issues:

1. Is precision of path following [4] a useful metric for evaluating the bicycle/rider system? We argue that it exercises a skill that is only rarely needed in the real world, and involves a riding regime that most would find intolerably wearying.

2. Is it appropriate to model a rider as an immediately-reacting (apart from known neuromuscular delays) continuous feedback control system? We hypothesize that ordinary tolerable riding might involve sensing roll angle error in relation to desired path curvature only *every few seconds*, followed by minimal *open loop* impulsive or steady steering inputs to keep the vehicle within the roadway edges – in essence a discrete control process, that relies on the rapid damping of roll transients after any steer torque input. We also explore a philosophy and possible method of approximate trajectory planning and following, based on the principle of minimizing rider sensing and control actions. In this scheme, continuous feedback control would be practiced only rarely, when path precision requirements become unusually demanding.

3. Bicycle + rider models are closely critiqued. (a) Appropriate rider flexibilities (roll, yaw, and lateral displacement of torso; lateral knee motion [5]) coupled with their internal-variable postural control dynamics (needed to stabilize rider configuration [6]) are clearly essential to the actual mechanics, and maybe also to the ease of riding, *but* appear prohibitive in complexity. We argue that rigid-rider models should be used exclusively until ideas about rider control are solidified; related experiments must rigidify the rider with a support frame. [7] (b) Models should also be upgraded to incorporate passive arm loadings on the steering assembly (inertia, geometric stiffness, and muscular impedance), and appropriate tire properties (pneumatic trail, camber thrust, spin torque) [8].

4. What benchmark maneuvers best reveal the roll authority afforded by a given bicycle, or the rider's limitations in high-frequency tracking? We propose that pure roll maneuvers in an open space eliminate subsequent data-obscuring roll moves (i.e., the heading and path-position corrections needed to stay on a path or treadmill). One test option could be tracking a commanded roll angle; another could be the response to a sudden step in roll torque.

5. The relation of steering impedance to "stability" will be discussed. High impedance does not directly imply greater dynamic stability, but it could have a role in reducing 'jitter' from muscular tremors or postural disturbances.

We outline methods for evaluating the foregoing hypotheses, and suggest what consequences to expect if they are validated.

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The Basic Human Input to Bike Control

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Abstract

This study, completed in 1987, aimed to establish the essential input from a human bicycle rider to maintain vertical stability. A human rider on a bicycle is an example of an inherently unstable system. With only two points of support and a high centre of gravity it is not possible to maintain static balance. The special case of a skilled rider with a fixed wheel is discounted as being not entirely static. The reason the system is so unstable is that as soon as the centre of mass is displaced from the vertical the weight forms a disturbing couple about the support points and the greater the angle of lean the greater the moment arm. This produces an accelerating angular velocity into the fall. The modern bicycle has two built in aids to stability. Up to some angle determined by the design details, the raked steering axis produces a steering couple in the direction of lean when the machine is set at an angle to the vertical. When the bicycle is moving forward this produces a force at the front wheel ground contact point in the direction of lean which in turn rotates the frame in the horizontal plane thus producing a similar force at the rear wheel ground contact point. These two forces combine to produce two relevant effects. On the one hand it puts the system onto a curved trajectory in the horizontal plane in the direction of the lean which changes the direction of travel, and at the same time it acts about the high centre of mass to produce a couple that opposes the disturbing couple in the vertical plane. The other automatic stability factor is due to the gyroscopic effect of the wheels. Angular velocity in the vertical plane is precessed through ninety degrees to become a rotation in the horizontal plane in the direction of the lean. At typical bicycle speed this effect is not strong but it is enough to add another couple about the steering head tending to turn the front wheel into the lean. The combination of these two steering couples produces a tendency for a lean to be converted into a turn in which the centripetal couple opposes the fall couple. Both stability features depend on speed for the production of force at the wheel ground contact point so their effect changes with riding speed. Obviously how a bicycle or bicycle rider combination behave will depend on the specific design details but in general terms at around twenty miles per hour most bicycles will respond to disturbances in the vertical plane by a rapid short period turn into the fall which restores the upright running with very little change in direction. This is the sort of performance expected from a motor cycle where the gyroscopic effect is much more powerful and the normal operating speed much higher. As the speed reduces the automatic stability features get weaker until they are no longer strong enough to restore upright running. A riderless bicycle left free to run at walking speed will slowly collapse along a circular path. The same would happen to a 'no-hands' rider who was foolish enough to insist on remaining in the saddle. A bicycle with the front steering locked dead ahead will fall much faster showing that the stability features are still having some effect even at low speed. Since humans are able to ride bicycles at very slow speeds without falling it is evident that they are doing something in addition to the automatic stability and it was the intention of this study to find out as much as possible about this input.

The first thing was to remove the automatic stability features to ensure that all the steering input came from the rider alone. The trail angle was removed by mounting the steering axis vertically. The gyroscopic effect was removed by mounting a second front wheel vertically above the other and driving it at the same speed but in the opposite direction. The result was a bicycle with zero built-in stability and various tests were carried out to confirm this. Launched on its own it fell over as quickly as a bike with the steering locked dead ahead. If the rider took his hands off the bar the bike fell over. Though potentially dangerous at high speed the machine was pleasant to ride and easy to control.

The records consisted of roll rate and handle bar position on a common time base. Weight transference has been suggested as a means of control and this would have been a contaminating factor as there was no record of rider position. A chapter in the thesis dealing with this subject in detail shows that although shifting the rider's weight either side of the plane of the bicycle's frame will have an effect on the configuration of the system it cannot on its own alter the balance between the two major couples in the vertical plane. Front wheel movement is necessary for control and since a riderless bike can be made to run upright at high speed it is also sufficient. Riding 'no hands' depends on front wheel movement via the built-in stability of the bike and is not possible with a zero-stability bicycle. In addition to this both subjects made every effort to reduce body movements to a minimum during the recording runs.

It was realized from the start that analysis of the records was going to be much easier if as little as possible of the control input was caused by extraneous events. Consequently visual input was seen as a potentially contaminating factor because it was not possible to tell whether features in the surrounding scene might be influencing the rider to make control inputs. It was soon found that riding a bicycle blindfold was no harder than doing so sighted so all the runs were made with zero visual input. It was then possible to identify the information available to the rider. The semicircular canals respond to roll and yaw accelerations and integration would produce velocity information. A further integration would also yield some

angle information. There would be inputs from the otolithic organs but the linear accelerations would be very low. There would possibly be some information from the pressure changes at the seat but these were bound to be somewhat ambiguous as in a sustained turn the loading down the body axis is much the same as when upright. Finally the rider would have positional and rate information about steering angle via the bar position.

Since all the stability augmentation at the front wheel had been removed, any movement of the steering bar can have come only from the rider, consequently, since the blindfold riders were able to prevent the bike falling, the movements of the steering bar were both necessary and sufficient to prevent a fall. This meant that the record of the roll and bar activity was a complete record of how the subjects were responding to the roll changes to prevent the machine falling over. The next task was to analyze the traces with the aim of constructing a model that would account for their relationship. To assist in working out what was going a computer simulation of both a stabilized and zero-stable bike was written.

The values recorded were handle bar angle and roll velocity. Because the vestibular system responds to roll acceleration rather than velocity the roll velocity was differentiated to convert the trace into roll acceleration and the initial comparisons were between this and the steering angle traces. The high correlation between these two traces combined with the lack of any autocorrelation in the bar trace showed that the bar was following the roll continuously rather than timing internally generated pulses. By finding the peak correlations the time delay between signal and response were established in the range 60-120 milliseconds. These were fairly constant within but different between the two subjects.

When this control technique was applied to the computer model it showed that responding to the acceleration in this way removed the roll acceleration and produced a gently oscillating value either side of zero. However this did not contain the velocity so with this control the simulation continued to fall at a steady rate. The riders on the other hand did not, so it was evident that they were responding to velocity as well. When some velocity as well as acceleration feedback was added to the simulated control it produced a similar oscillation around zero velocity. When both acceleration and velocity were combined correlations between handle bar and roll activity were even higher. There was still something missing because the simulation showed that with this control the bike entered a continuous turn whereas the riders although they were briefed to make no attempt to maintain a straight course tended to correct turns and always maintained roughly the same direction. So they must have been able to detect and respond to turn as well.

The simulation allowed the exploration of some possible alternatives. Adding a continuous component of direction produced traces that were a good deal more stable than the actual traces. A closer study showed that there were places where the two curves seemed to fit rather badly so the residuals once the ‘fit’ between the two curves had been accounted for were extracted. These took the form of short pulses of input every now and then. When these were put on the same time trace as lean angle they coincided with the places where the turn was corrected back up to upright running. It appeared that when the turn rate exceeded some low value the riders were producing a short stab of handle bar input to push the bike back up towards the upright. This control technique, when applied to the computer simulation, produce a very similar output to that recorded on the real bike. The result of the above showed that given an unstable bicycle two riders of quite different build and riding experience were using their existing riding skills to ride blindfold in an approximately straight line and that the control they were using could be accounted for with the model outlined above.

Some unrecorded experimental runs made on a bike with full stability showed that given sufficient speed lean control was automatic and the bike ran upright without human contribution. A short push on the bar destabilized the bike momentarily but the combination of trail and precession rapidly brought the bike back to the upright on a slightly different heading. Smooth directional control was achieved by gently applying an angle independent torque to the steering head. This is interesting, as at first sight it appears contradictory. To go left the torque is applied as though to turn the handle bar to the right. To put it another way, to go left push on the left bar. This push produces a strong roll to the left and the reason riders do not easily appreciate what they have done is because the automatic control immediately responds to this rapid roll by turning the bar to the left to control the fall with a torque value greater than the rider’s input. Consequently the bar goes left even though the rider is pushing it to the right. When questioned riders almost always claim they turn the bar into the turn. If the rider were to apply an angle input instead of an angle independent torque then there would be a dramatic fall in the opposite direction. As soon as the rider removes the torque the bike returns to upright running. In effect, by adding a steering torque to the right the rider resets the zero position of the automatic control from ‘steady upright’ to ‘steady lean’ and the automatic control responds by turning so as to contain the resulting roll. The automatic stability on a bike depends on speed so it becomes less effective as the speed falls. Since, however, riders can remain in control down to very low speeds, it is evident that they must be supplementing the automatic stability as speed decreases. This makes sense as learning always takes place at low speed and most children’s bikes have poor automatic stability characteristics. The reason that there is no conflict between the two systems at intermediate speeds and above is almost certainly due to the considerable difference in response times. The system delay in the human response measured here was in the range 60-120 milliseconds but the two mechanical control features have literally no delay at all, consequently at high speed the automatic control removes any roll error before the human sensory system can detect it. As the speed falls there comes a point where the automatic system fails to contain the vertical angle and the human system picks this up and adds the required additional torque to the steering head. It is emphasized that neither control system deals in terms of steering angle. In both cases the controlling input takes the form of an angle independent couple or torque.

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Design of a Novel Aerodynamically Efficient Motorcycle

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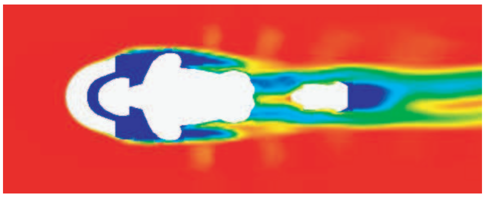
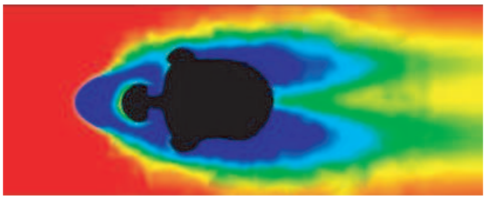
Abstract

Recently a novel aerodynamically efficient motorcycle known as the ECOSSE Spirit (ES1) was designed and patented [1]. Initial computational fluid dynamics (CFD) analysis estimates the ES1's drag-area coefficient at $0.16m^2$ - a 54% improvement over a conventional sports machine (such as the Yamaha R1), which translates into a 30% increase in top speed for a given fixed engine power. Or alternatively, 170bhp performance with a 78bhp engine; see Figure 1. Since the initial design of the ES1 was approached from an aerodynamic perspective, Imperial College London has been working with Spirit Motorcycle Technology on improving the overall dynamics and performance of the vehicle. This paper addresses the parameters that affect the lateral dynamics of the vehicle, which is one of the many aspects in motorcycle design.



(a) ES1 CFD calculation top view

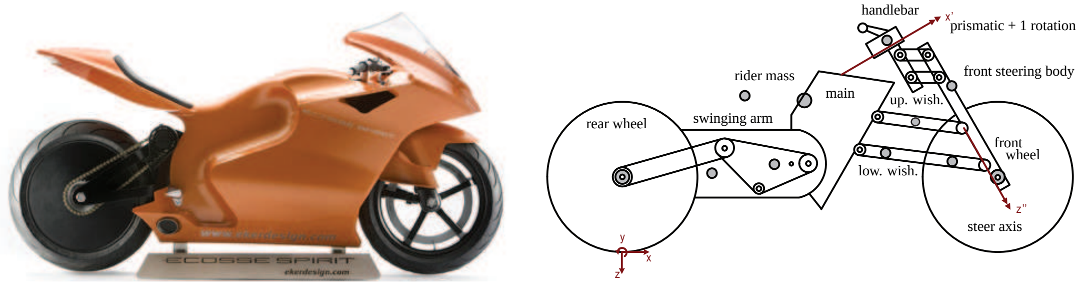


(b) Yamaha R1 CFD calculation top view

Figure 1. Plots comparing air velocity CFD calculations between the ES1 and the Yamaha R1. The ordering red-yellow-green-blue depicts the relative air velocity from low to high respectively. Plot (a) shows the air flow around the ES1-rider combination. The computed aerodynamic drag-area coefficient with the rider in the prone position is $C_dA = 0.16m^2$. Plot (b) shows the air flow around the Yamaha R1-rider combination. The computed aerodynamic drag-area coefficient with the rider in the prone position is $C_dA = 0.35m^2$. These results were obtained from Spirit Motorcycle Technology.

In order to capture the dynamics of the ES1, nonlinear and linearized mathematical models of the vehicle were computed using the multi-body package VehicleSim [2]. The rigid body model is shown in Figure 2 and further details are given in Chapter 6 of [3]. Root-loci and Nyquist diagrams of the linearized vehicle under constant speed and lean [4], as well as acceleration and braking [5], are used to determine the stability bounds on various design parameters.

The vehicle parameters considered are related to the tires, the lower wishbone flexibility, the rear swinging arm flexibility, a proposed introduction of a front lateral flexibility and changes in the aerodynamic properties. The effect of each component is individually analyzed by comparing stability changes due to variations of a single parameter in an otherwise rigid vehicle. The results



(a) ES1 side view (b) ES1 multi-body diagram

Figure 2. Diagrams comparing the side view of the ES1 with the multi-body model programmed in VehicleSim [2]. Picture (a) was obtained from Spirit Motorcycle Technology and shows the right side view of the ES1 motorcycle. Diagram (b) shows the multi-body arrangement of the ES1. Shaded circles represent the body masses, white circles represent ball joints, while concentric circles show hinge joints with rotational freedoms along an axis pointing out of the page. The motorcycle is fitted with Pacejka Magic Formulae tires [6].

show that high friction tires are required, a deliberate introduction of a front lateral flexibility is ill-advised, the rear swinging arm and the front lower wishbone flexibilities should be made as stiff as economically possible and a passive mechanical network involving an inerter [7] is needed to stabilize the accelerating machine. Finally, the effects on the lateral dynamics due to changes in aerodynamics and vehicle loading are investigated. The weave mode becomes more unstable when rear wheel loading increases, or a higher aerodynamic center of pressure, and the wobble mode destabilizes when front wheel loading increases. The observed trends are compared with several results in the literature [4, 8, 9], and similarities as well as differences are highlighted.

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Motorcycle control by variable geometry rear suspension

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Abstract

This work is about controlling the geometry of the rear suspension to reduce cornering weave oscillations associated with high performance motorcycles operating at high speeds. The main lateral oscillations in two-wheeled vehicles are "wobble" and "weave". The wobble mode is a steering oscillation whose frequency is normally in the range $6 - 9$ Hz and varies little with speed. The weave mode is a fishtailing motion that involves mainly yawing, rolling and steering of the vehicle. At low and intermediate speeds it is well damped but becomes less damped as the speed increases, in straight running or under moderate lean angle conditions. The natural frequency is lower in heavier motorcycles and rises from zero at very low speed to somewhere in the range $2 - 4$ Hz. In cornering, the above lateral modes and the in-plane tyre deflections and suspension modes become coupled, as was first shown by Koenen [1]. The motorcycle becomes prone to resonant forcing when regular road undulations produce displacement forcing that is tuned to lightly damped modal frequencies of the machine. Moderate roll angles appear to represent the worst case conditions [2].

Conventional motorcycle rear monoshock suspension systems are designed to provide varying but predetermined leverage ratios between spring damper unit and road wheel, as the suspension travels to its limits. These characteristics are motivated and designed by static equilibrium considerations. Further variations of the leverage ratio can be superposed by utilising variable geometry arrangements in which the mechanical path along the suspension kinematic loop is combined with an actuator. The actuator will essentially act as a displacement controller and will interact with the conventional passive force-producing elements, spring and damper, to provide active control. The degree of their interaction will primarily be determined by the geometric design, and it will prescribe the actuator force and power requirements. Variable geometry suspension ideas have already been explored in [3, 4, 5, 6, 7] for cars and in [8] for motorcycles. In the case of motorcycles the participation of the rear suspension in weave oscillations implies the potential for an active variable geometry suspension to add damping to the weave mode.

The broad purpose of this research is to investigate the practicality and performance of a variable geometry suspension scheme in improving the cornering weave behaviour of modern high performance motorcycles. The suspension under consideration aims to incorporate four major characteristics. Firstly, due to packaging constraints, the operating space of the variable geometry system should be close in size to that of the standard passive system. Secondly, in order to reduce costs it should be possible to use much of the existing passive technology with easy retrofitting of any new parts. Thirdly, variation of the geometry should be done with low actuation forces and power so that actuators of reasonable size can be used. Fourthly, fail safe operation should be easily achievable.

The large perturbation performance of the variable-geometry-controlled motorcycle is studied by simulation. An illustration of such simulation results is shown in Figure 1.

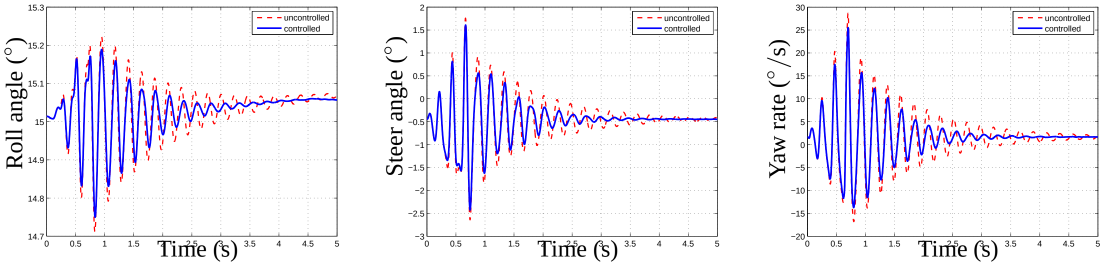

Figure 1. Transient behaviour of the roll and steer angles and the yaw rate for the uncontrolled (dashed line) and controlled (solid line) machine, in response to sinusoidal road forcing that begins at $t = 0$ and ends at $t = 0.623$ s, and has a peak amplitude of 5 mm. The forcing frequency is tuned to the weave mode. The forward speed is 75 m/s and the lean angle 15 deg.

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Sensitivity Analysis of Multibody Systems : Evaluation of Mountain Bike Dynamical Performances

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Abstract

In the field of vehicle dynamics, it recently appeared that the role or – more specifically – the precise impact of suspension parameters on *subjective* comfort evaluation and handling performances is needed to be better quantified. The general purpose of the present research is the development of a precise, robust and efficient method to quantify vehicle parameter sensitivity for given parameter ranges and specific excitations. In view of the problem complexity and of the difficulty to organize relevant experimentation programs with cars and professional pilots (for their expertise in vehicle dynamics), and with common drivers (for statistically collecting their own feeling), we propose to focus, from the experimental point of view, on a simpler set-up consisting of a bike and a biker on a home trainer, possibly shaked to take road irregularity into account.

Figure 1. Experimental set-up - left: Tacx® Virtual Reality trainer, right: ROBOTRAN multibody model.

Starting from a commercial trainer (see Fig. 1_{left}) (from the Tacx® Compagny), which places the biker who is pedaling on his own bicycle, in an agreeable environment, the system will be updated as follows:

- increase in degrees of freedom (roll and lateral displacement);
- vertical shaking of the roller (road irregularities simulation);
- instrumentation for sensitivity analysis :
    - input (e.g.): suspension pressure, tube pressure, frame rigidity, etc;
    - output (e.g.): driver head acceleration, suspension force, etc.

In addition to the previous quantitative output – which can be qualified as *objective* – that we will use to validate the multibody model (see Fig. $_{right}$), *subjective* output will be of the utmost importance in the present case. Indeed, the main goal of the study relates to the *subjective evaluation* of vehicle performances, which is far from being trivial since it refers to subtle human perception whose ”absolute” value is obviously an individualized feeling. Thus, instead of using absolute values (as for vehicle comfort via rms accelerations), we propose to measure human ”sensation variations” via experiments (as done in [1]) and to evaluate the associated cost function sensitivity, via suitable multibody simulations. The aim of the present research work is really to compare experimental and simulated sensitivities for a rather simple situation (namely the biker on rollers) to assert for the pertinence of the proposed model-based approach in vehicles dynamics.
An other part of the research will focus on the parameter sensitivity computation for large multibody systems, via the symbolic process underlying the ROBOTRAN program.

To compute this subjective evaluation numerically, we have to consider objective functions which are based on complete time integrations along a given trajectory, for instance to quantify the sensitivities of the passenger rms accelerations (comfort) with respect to suspension parameters. On one hand, the accuracy of the various partial derivatives computation can be greatly enhanced thanks to the symbolic capabilities of the ROBOTRAN multibody program [2]. On the other hand, the computational efficiency of the process also takes advantage of the recursive formulation of the multibody equations of motion. Indeed, the latter must be time integrated with respect to both the generalized coordinates and their partial derivatives in case of the so-called direct method underlying sensitivity analysis. In the multibody dynamic context, a typical objective function $\psi(p)$ for sensitivity analysis purpose can be written as:

$$\psi(p) = G^1(t^1, u^1, \dot{u}^1, p) + \int_{t^0}^{t^1} F(t, u, \dot{u}, \ddot{u}, p)\, dt \tag{1}$$

in which t^0 and t^1 are the initial and final simulation time, G^1 refers to the final state (ex. the configuration at t^1 of a given body of the system), F depends on the dynamic behavior of the system in the time interval $[t^0, t^1]$. For the general objective function (1), sensitivity analysis consists in computing $\frac{d\psi}{dp}$, that is:

$$\frac{d\psi}{dp} = \frac{\partial G^1}{\partial u^1} . \frac{du}{dp}\bigg|_{t^1} + \frac{\partial G^1}{\partial \dot{u}^1} . \frac{d\dot{u}}{dp}\bigg|_{t^1} + \frac{\partial G^1}{\partial p} + \int_{t^0}^{t^1} \left(\frac{\partial F}{\partial u} . \frac{du}{dp} + \frac{\partial F}{\partial \dot{u}} . \frac{d\dot{u}}{dp} + \frac{\partial F}{\partial \ddot{u}} . \frac{d\ddot{u}}{dp} + \frac{\partial F}{\partial p} \right) dt \tag{2}$$

In this paper, we will point out how a recursive symbolic generation [2] can be advantageously exploited to compute (2) in the most optimal way. Application to bike dynamics will then illustrate the proposed method and the envisaged protocol.

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Experimental Analysis of Rider Motion in Weave Conditions

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Abstract

Motorcycle dynamics is characterized by the presence of modes of vibration that may become unstable and lead to dangerous conditions. In particular weave instability shows large yaw and roll oscillations of the rear frame and oscillations of the front frame about the steering axis in opposition to yaw [1] [2]. The presence of the rider influences the modes of vibration [3], since the mass, stiffness and damping of limbs modify the dynamic properties of the system; moreover at low frequency the rider can control vibrations.

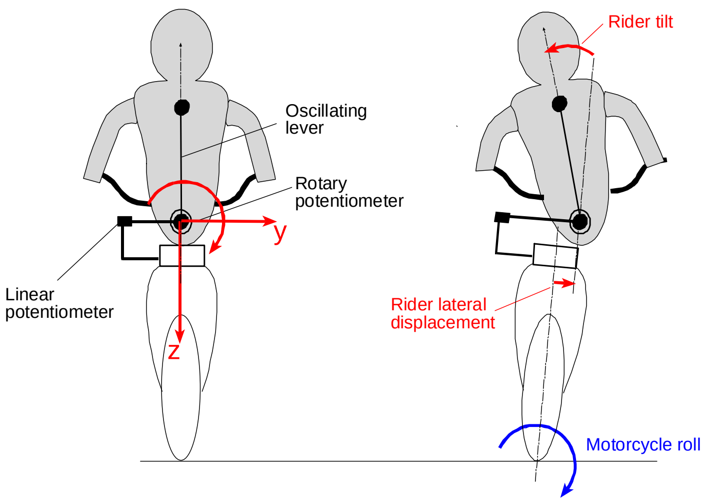

Figure 1. Measurement scheme.

Up to now very few experimental results dealing with rider's behavior in the presence of weave are available. This lack is due to the difficulty of carrying out measurements on the road and of reproducing the phenomena in laboratory.

This paper deals with a research program aimed to measure the oscillations of rider's body on a running motorcycle in the presence of weave vibrations. First experimental methods are dealt with. The motorcycle is equipped with many sensors (accelerometers, gyrometers, steering angle potentiometer) that monitor its motion. Special equipment has been developed to measure the relative motion between the rider and the motorcycle (figure 1). A linear potentiometer measures the lateral displacement of the lower body of the rider with respect to the motorcycle. A rotary potentiometer measures the rotation of an oscillating lever that connects the motorcycle with the upper body.

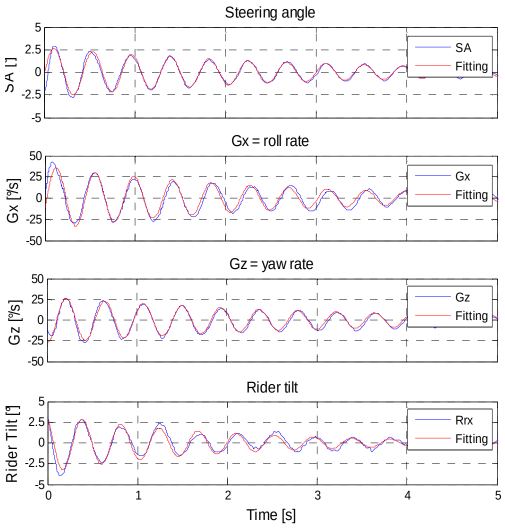

Figure 2. Measured motion parameters at 180km/h.

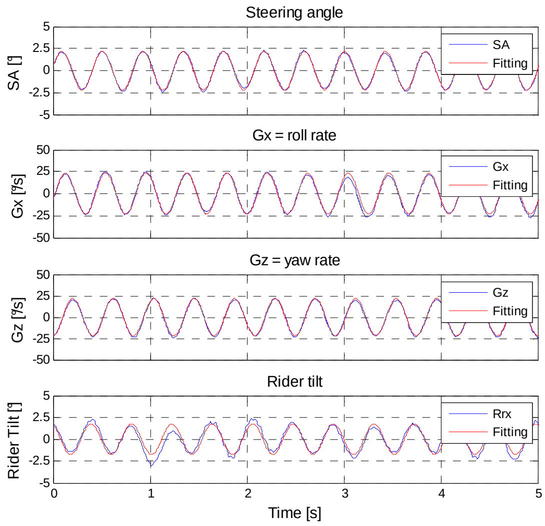

Figure 3. Measured motion parameters at 210km/h.

Then the results of road tests carried out at increasing speed (from 160 to 210 km/h) are described. Below the instability threshold the weave mode is excited by a steer input, at higher speed the vibration is self-sustained. Figure 2 shows an example of kinematic parameters of the motorcycle (steering angle, roll and yaw rates) and of the rider (rider tilt angle) measured below the instability threshold. Figure 3 shows the same parameters when weave is self-sustained.

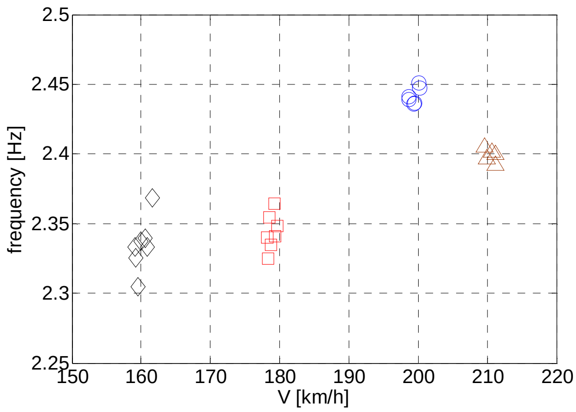

Figure 4. Identified weave frequency against speed.

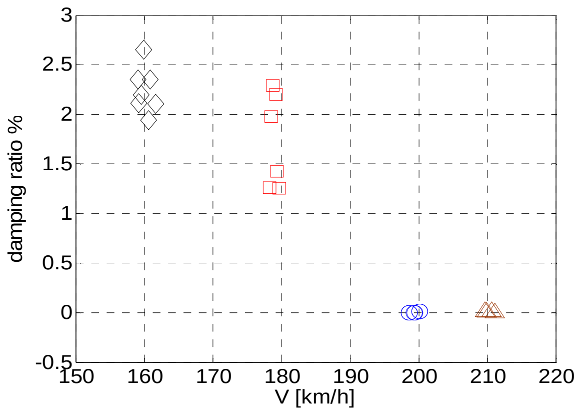

Figure 5. Identified weave damping ratio against speed.

The last section deals with the analysis of experimental results. Measured quantities are fitted in time domain, natural frequencies, damping ratios and relative phases are identified at the various speeds, see figures 4 and 5. The characteristics of the motion of the vehicle and of the rider are described in terms of amplitudes and relative phases. These results are useful for understanding rider's influence on motorcycle dynamics and for validating numerical codes.

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Motion Analysis of a Motorcycle Taking Account of Rider's Effects

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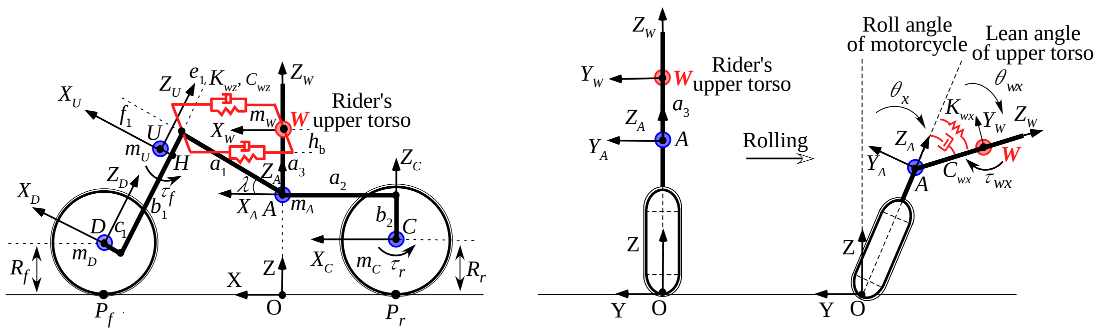
Abstract

In this paper, to analyze rider's effects on the motion of a motorcycle, we model a rider-motorcycle system taking account of the lean motion of the rider's upper torso [1], [2] and the rider's arm [3]. Figure 1 shows the dynamical model of the rider-motorcycle system, where the rider's upper torso can rotate around the roll axis and the rider's arm is connected to the handle from the upper torso with a spring and a damper. In addition, we introduce the nonlinearity of the tire force to the tire model with taking account of the cross-sectional shape, the elastic deformation and the tire-ground contact area[4], [5]. In Reference [6], we have already designed a front-steering assist control system [7], [8] to stabilize the motorcycle on the basis of the dynamical model including the lean motion of the rider's upper torso without the rider's arm.

In this study, by carrying out simulations, not only the effect of the lean motion of his/her upper torso but also that of the rider's arm in steady-state turning are analyzed. The constant steering torque from the rider is directly applied about the handle axis. In simulations, we have a rider lean in, lean with and lean out the motorcycle roll angle by applying the lean torque to the rider's upper torso. Figure 2 shows the simulation results of the steady-state turning taking account of rider's effects at 35 km/h. The friction coefficient of the road surface is originally 0.8 and suddenly decreases to 0.6 from 2 s to 7 s. The constant steering torque about the handle axis is - 9 Nm. The lean torque to the rider's upper torso is 20 Nm to have the rider lean in. In case of the rider model with arms, when the rider's upper torso leans in the same direction of the roll angle and keeps the lean angle of 5.6 deg at the steady-state, the roll angle of the motorcycle increases to about 26 deg. In case without arms, the lean angle, the roll angle and the steering angle become larger than those of the rider model with arms. Also the friction coefficient change severely affects the motorcycle motion and causes the roll angle vibration comparing to the rider model with arms. It is seen from Figure 2 that effects of the rider's arm and the rider's posture to motion analysis of motorcycles are large.



(a) Side view (b) Rear view

Figure 1. A dynamical model of the rider-motorcycle system

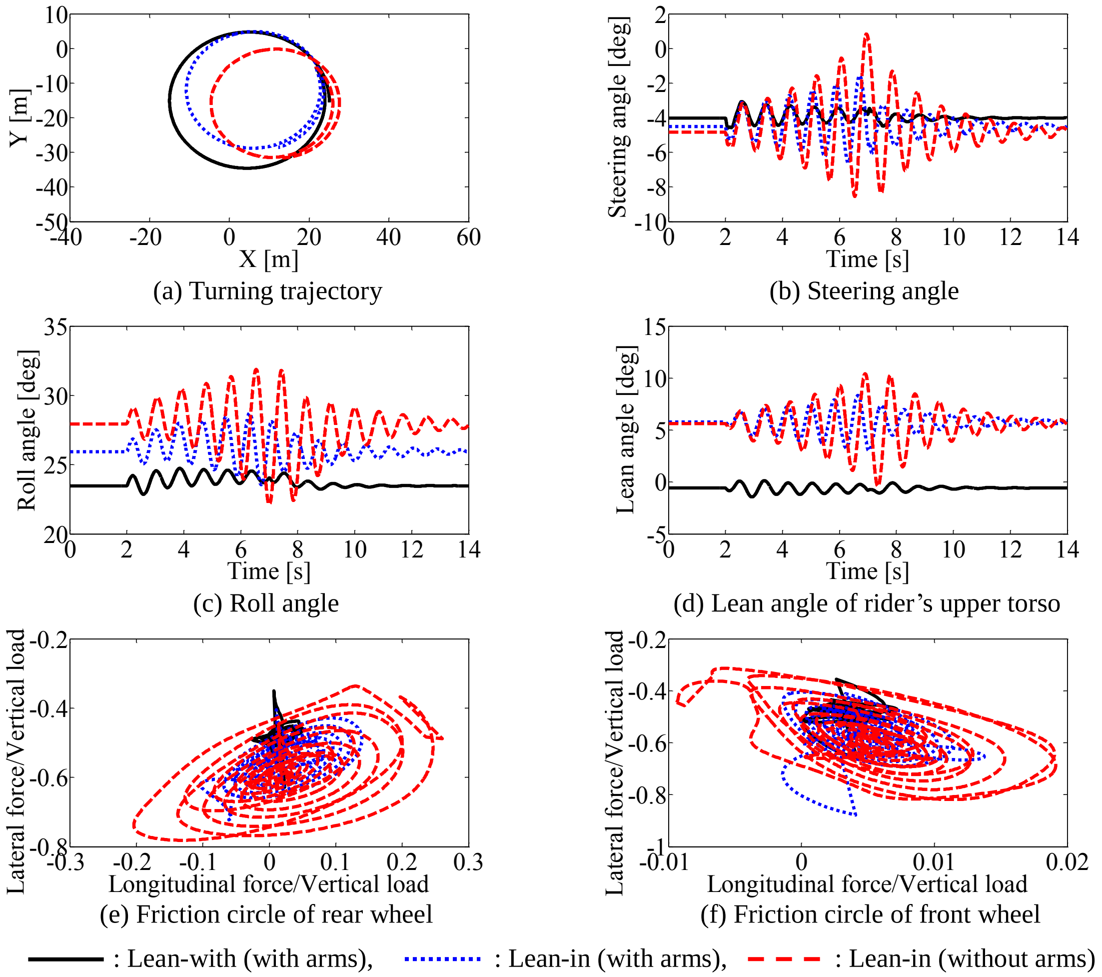


Figure 2. Simulation results of steady-state turning taking account of rider’s effects at 35 km/h (0 s - 2 s: μ=0.8, 2 s - 7 s: μ=0.6, 7 s - 14 s: μ=0.8)

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The Influence of a Passive Rider on the Open Loop Dynamics of a Bicycle

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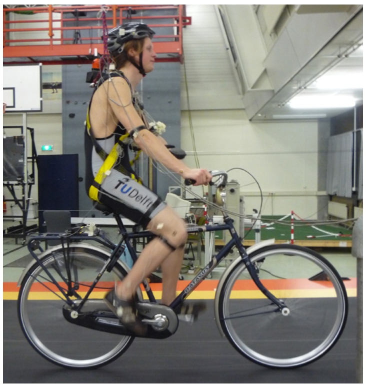
Abstract

The bicycle is an intriguing machine as it is laterally unstable at low speed and stable, or easy to stabilize, at moderate to high speed. During the last decade a revival in the research on dynamics and control of bicycles has taken place [1]. Most studies use the so-called Whipple model [2] of a bicycle. In this model a no-hands rigid rider is fixed to the rear frame. However, we know from experience that some sort of control is required to stabilize the bicycle and/or carry out tracking operations. This control is either done by steering or by performing a set of upper body motions. The precise control used by the rider is currently under study [3, 4].

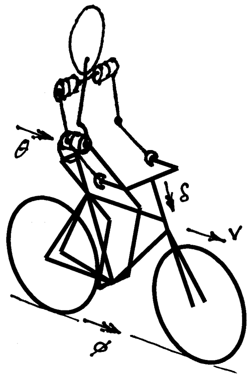
This paper addresses the influence of a passive rider on the lateral dynamics of a Whipple-like model. In this model the arms of the rider are connected to the handlebar and the rider can use both steering and lateral upper body motion for control. From observations of bicyclist riding on a large treadmill [3, 4] two major and distinct rider postures can be identified. The first is a rider on a city bicycle where the posture is upright with the arms hinged at the elbows and where the upper body does not move back and forth, Figure 1a. The second is a rider on a sportive touring bicycle where the posture is such that the upper body is leaning forward, the arms are stretched and where the upper body is able to twist, that is, rotate about its vertical axis, see Figure 1c.

Models can be made where neither posture add any degree of freedom to the system other than the extra degree of freedom for the lateral upper body motion. In both models, Figure 1b and 1d, the legs are rigidly connected to the rear frame and the upper torso is allowed to move laterally with respect to the rear frame by means of a single hinge located at the saddle. The model for the first, upright, posture is shown in Figure 1b. Here the upper arms are connected to the torso by a single hinge and the lower arms are connected by universal joints at the elbows and by ball joints at the handlebar. In the model for the second, leaned forward, posture, see Figure 1d, the arms are stretched and the upper and lower arms are modelled as one rigid body each, connected by universal joints to the torso and by ball joints to the handlebar. The upper body is allowed to pitch and twist. The geometry and mass properties of the two bicycles and the rider used in this study where measured by the procedure as described in [5].

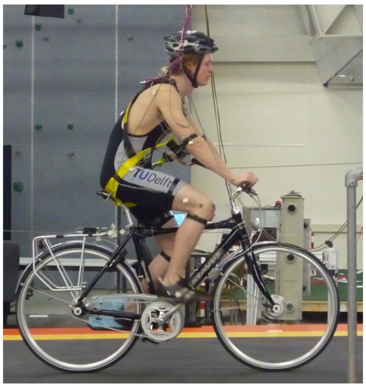
For both postures the open loop, or uncontrolled dynamics of the bicycle-rider system is investigated by examining the eigenvalues and eigenmotions in a forward speed range of 0 to 25 km/h. It is shown that such a passive rider can substantially change the eigenvalues compared with those where the no-hands rider is rigidly attached to the rear frame, the original Whipple model.



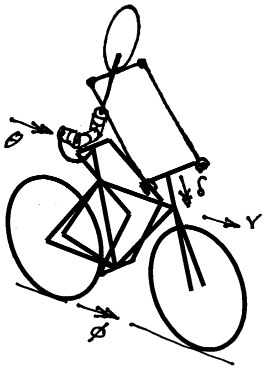
(a) Upright rider posture on a city bicycle.



(b) Model for a bicycle rider combination with an upright rider posture.



(c) Leaned forward rider posture on a sportive touring bicycle.



(d) Model for a bicycle rider combination with a leaned forward rider posture.

Figure 1: Two distinct rider postures and their Whipple like models. Degrees of freedom are the forward speed v, rear frame lean angle ϕ, upper body lean angle θ, and steer angle δ.

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IntelliBike: condition monitoring of our cycling infrastructure

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Abstract

This paper presents preliminary designs for the authors' IntelliBike: an innovative instrumented bicycle that can monitor the UK's cycling infrastructure in real-time. IntelliBike will link global positioning system data, real-time camera footage, vibration (ride-quality) data, environmental (noise, light and pollution) data and, depending upon feedback and ideas at conference, extend this into cycle journey analysis and improved lateral vibration monitoring (to enhance the ride-quality monitoring and safety aspects of IntelliBike) along the UK's off-road cycling network. A dedicated software platform will collate all data and provide an asset management tool aimed at better targeted cycling infrastructure maintenance. This applies the principles currently used on-road to cycleways thereby lending this sustainable mode a similar level of technology in its maintenance. The IntelliBike will be a software platform showing streamed video footage of each cycle-way with automated defect and hazard recognition, the correlation of measured on-bike vibrations with running surface quality, the correlation and mapping of camera data onto position data to support the vibration measurements, and the related monitoring/analysis of environmental parameters for the first time across Edinburgh's off-road cycle-way network. Self-funded UK-wide roll-out follows and the intention is to collaborate with EU partners, many of whom will be in the audience here, for extended application.

The proposed research contributes to the development of knowledge and practice in relation to sustainable infrastructure development. The UK's National Cycle Network comprises 16,000 km of running surface (The City of Edinburgh has a 150 km cycle network: 75 km on-road, 75 km off-road). This represents a significant civil engineering infrastructure asset that currently contributes to the provision of a sustainable transport mode option nationwide. Dr Beeching left many UK cities, and rural areas, with an extensive off-road cycle path network. Commuting and recreational cyclists have observed the often hazardous conditions on these paths. Furthermore, Danish and Dutch cyclists have witnessed simple, yet effective, actions taken by local municipalities to tackle cycling infrastructure maintenance issues. There are various simple, effective, measures that could be taken to improve the maintenance of such off-road paths. However, reliance on walk-over surveys and path users notifying the damage reporting agencies (*e.g.* Street Faults Contact Centre, RALF *et al.*) is not tackling maintenance in a resource-efficient manner.

The City of Edinburgh Council's stated 2020 target is for 15% of journeys to be undertaken by bicycle (*c.f.* national target of 10% across Scotland by the same date). Working with The City of Edinburgh Council's Cycling Officer (Mr Chris Brace), Spokes, Cycling Scotland, SUSTRANS, the authors are developing IntelliBike (Figure 1) for trial in Edinburgh and eventual UK-wide roll-out. Recommendations to The City of Edinburgh Council included improvements in the consistency and regularity of monitoring and data reporting.

Symbol	Instrument
⊗	Air quality sensors (2nd on rider helmet)
	Light meters (1 horiz. & 1 vert.)
	Sound level meter (or on helmet)
●	Accelerometers
	GPS system
	Cameras (1 horiz. & 1 vert.)
	Data-logger

Figure 1. IntelliBike schematic showing key instrumentation mounting points.

Experts in transportation policy, highway design and key organisations relating to the development of cycling as a mode of transport including: Sustrans, and Cycling Scotland have been consulted. The collecting of asset condition data and the development of a condition monitoring tool from a civil engineering perspective represents a research opportunity. In York, cycle mounted maintenance rangers, each towing a trailer of tools, helped improve cycling facility maintenance [1]. Cycle couriers in Cambridge were recently equipped with sensors that reported air pollution levels with their locations being monitored using mobile telephone signals [2]. Data went *via* Bluetooth to a laboratory where pollutants and exposure levels were recorded.

The Dutch Cyclists Union developed an instrumented bicycle to monitor air and noise pollution in Amsterdam [3]. This registers time, distance, speed, sound, vibrations, and waiting times. The camera mounted on the bicycle recorded the road surface type, its classification, and the construction of intersections and obstacles. It was used to inform decision makers in the re-routing of cycle paths to avoid areas of high air particulate and noise pollution from motor vehicles. In Odense, Denmark, four cyclists are equipped with mobile phone cameras for photographing defects to send to the roads and parks maintenance officer with a text description and location. The scheme included payment for each accepted message. This type of scheme is used for maintenance defects in Lothian through the ‘Clarence’ phone line, but represents a reactive approach rather than IntelliBike’s preventative approach to the management of cycling infrastructure.

An instrumented bicycle was used to measure particulate matter exposures along bicycle routes through a variety of land uses over summer in Vancouver. The data were used to help cyclists weigh-up the risks of bicycle commuting, and planning engineers as they sought optimal designs for designated cycle-ways [4]. No cycling infrastructure decision support tool to assist in the allocation of funds to maintenance was found: the IntelliBike’s linkage to engineering asset management is novel.

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Accurate Measurement of Bicycle Physical Parameters

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Introduction

Accurate measurements of a bicycle's physical parameters are required for realistic dynamic simulations and analysis. For the most basic models the geometry, mass, mass location and mass distributions must be measured. More complex models require measurements of tire characteristics, human characteristics, friction, stiffness, damping, etc. This paper concerns the measurement of the minimal bicycle parameters required for the benchmark bicycle presented in [4]. This model is composed of four rigid bodies, has ideal rolling and frictionless joints, and is assumed to be laterally symmetric. A set of 25 parameters is used to describe the geometry, mass, mass location and mass distribution of each of the rigid bodies. The experimental methods described herein are based primarily on the work done in [2] but have been refined for improved accuracy and methodology. Koojiman's work was preceded by [5] who measured a bicycle in a similar fashion and both [1] and [6] who have used similar techniques with scooters. We measured the characteristics of six different bicycles, two of which were set up in two different configurations. This is a total of eight different parameter sets that can be used with, but not limited to, the benchmark bicycle model. The accuracy of all the measurements are presented up through the eigenvalue prediction of the linear model and are based on error propagation theory with correlations taken into account.

The six bicycles, chosen for both variety and convenience, are as follows: *Batavus Browser*, a Dutch style city bicycle measured with and without instrumentation as described in [3]; *Batavus Stratos Deluxe*, a Dutch style sporty city bicycle; *Batavus Crescendo Deluxe* a Dutch style city bicycle with a suspended fork; *Gary Fisher Mountain Bike*, a hardtail mountain bicycle; *Bianchi Pista*, a modern steel frame track racing bicycle; and *Yellow Bicycle*, a stripped down aluminum frame road bicycle measured in two configurations, the second with the fork rotated in the headtube 180 degrees for larger trail.

Experimental Methodology

The benchmark bicycle requires five basic geometric measurements: wheelbase w, steer axis tilt λ, trail c, and wheel radii r_{F} and r_{R}. For each geometric measurement, care was taken to measure the parameter directly as possible with high accuracy measurement equipment. The mass of each of the rigid bodies was measured with a precision scale. The center of mass of each rigid body was measured by hanging each body from several points. The mass center is the intersection of the lines of support under the assumption of lateral symmetry. Wheel centers of mass were assumed to be at their geometric center.

The moments of inertia of the rigid bodies were measured in both torsional pendulum and compound pendulum configurations. Again the assumed symmetry of the bicycle was utilized to reduce the number of measurements needed. In-symmetric-plane moments of inertia were calculated from measurements of the periods of oscillation when the rigid bodies were hung as a torsional pendulum about the vertical at different orientation angles. Out-of-symmetric-plane moments of inertia were measured by swinging the rigid bodies as a compound pendulum. Periods were estimated from zero crossings of precision rate gyro measurements.

Results

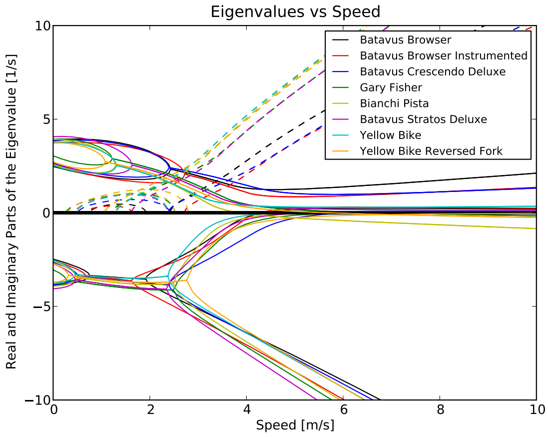

Figure 1. Eigenvalues versus speed for all eight bicycle configurations.

Parameter sets with measurement uncertainties for all six bicycles and all eight configurations will be presented, together with basic comparisons of the open loop dynamics of the different types of bicycles (eigenvalues, eigenvectors, Bode plots, etc.), Fig. 1. Notable results include high accuracy measurements and uncertainty propagation in the eigenvalues and eigenvectors.

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OBD: **O**pen **B**icycle **D**ynamics

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Abstract

Open Bicycle Dynamics (OBD) is an open source set of C library functions for analyzing the linear and nonlinear dynamics of the Whipple bicycle model, extended to include toroidal tires. Similar in some regards to JBike6 [2], OBD offers several extra features. Notably, it includes nonlinear and linearized equations of motion (linearized about any configuration), and three inputs torques – rear wheel torque, steer torque, and front wheel torque (for modelling rider actuation or joint friction). Steady turning equations of motion are also included, and allow for determination of the steer torque and ground reaction forces required to maintain a steady turn, as well as the stability of such a turn. Other useful quantities, such as the kinetic and potential energies, ground reaction forces at the wheels, linearized system A and B matrices (for any configuration), and kinematic constraint (for ensuring both wheels remain on the ground) are also included. The code has been validated against results presented by Meijaard et al. [3] and Basu-Mandal et al. [1]. OBD is written in C and is optimized for speed, platform independence, and minimal dependencies in order to compile and use. The source code and instructions for installation are available on the Internet at http://github.com/hazelnusse/OBD.

Implemented in C, simulations in OBD are extremely fast, making it well suited for parametric analysis or use in interactive 3D animations. Several examples of use are provided with OBD: eigenvalue generation, numerical integration of equations of motion, and an OpenGL 3D animation in which parameter values can be changed during simulation.

The current bicycle model included with OBD is the rigid rider model, but as other codes are developed for other models, they will be included into OBD as well. Work is currently underway to incorporate a leaning rider model (as in [4]), and welcome the contribution of other thoroughly tested models.

The end goal of OBD is to provide a place for high quality, peer-reviewed, and open source bicycle dynamics code to reside, in a form that is accessible and not tied to proprietary software. This will help the development and validation of new models by providing an easy and straightforward way to make concrete numerical comparison with models which are well established within the scientific literature. Additionally, it is our belief that this will foster more collaboration within the bicycle dynamics community by providing a common tool that can be used for analysis and validation.

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Hypotheses Formulation in Multibody Modeling: Application to Bicycle Transmission Dynamics

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Abstract

Being very active at the Université catholique de Louvain (UCL, Belgium) in the field of multibody dynamics, we are now focusing on an important aspect of this modeling activity: the statement and formulation of "reasonable and relevant modeling hypotheses". Of course, this problematic concerns all fields of engineering but in the particular case of modeling, we think that the focus must be reinforced at both research and educational levels.

In the field of mechanical design, the starting point is a client request to build a new machine (or process) or to enhance the performances of an existing one. The "translation" of this request into the well-known *project specification* represents the first step of the design process. In the field of modeling – multibody applications in the present case – we want to adopt a similar methodology for our own research and industrial collaborations: the "translation" of the client request (being in this case the modeling of a new system or the improvement, by simulation, of an existing one) into a so-called *modeling specification*. Among other things, the latter will contain an unambiguous formulation of the modeling hypotheses, which are needed to simplify the original system for obvious reasons: a reasonable number of parameters to be identified, reasonable time for model establishment and of course reasonable computational time.

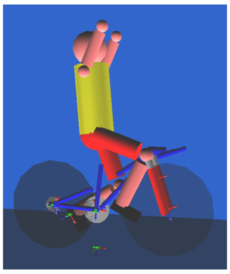
Our motivation comes from the following observation: facing a modeling R&D project, engineers working with commercial software have a tendency to build models (often including impressive 3D virtual representation) which are sometimes "a bit monstrous" with respect to the real original client problem. As an example, the understanding of the dynamic behavior of pneumatic suspension systems of railway vehicles does not necessarily require the introduction of complex wheel/rail contact models "just because it is a railway vehicle". On these observations, we have thus decided to focus on this important step — hypotheses formulation — for both students in mechanical engineering and researchers in multibody dynamics.

Let us briefly illustrate this problematic via the modeling of a mountain bike equipped with modern suspension systems (see Fig. 1). The precise question is: what is the influence of the pedaling forces on the bike bounce motion and suspension power dissipation? Let us restrict the bike

Figure 1. Mountain bike with closed-loop suspension mechanisms

(+driver) to a planar motion, by focusing on vertical and pitch motions of the various bodies. Fig. 2 gives an illustrative result which shows the power dissipated in dampers for different values of damping and stiffness coefficients, when the drivers is pedaling at constant velocity.

As regards *a posteriori* hypotheses, an interesting validation concerns the kinematics and dynamics of the roller chain transmission. With respect to bicycle bounce performances (which is the



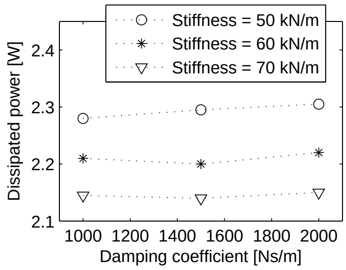


Figure 2. Bike bounce motion (left: Robotran MBS model, right: power dissipated in dampers)

project modeling objective), we can propose three different modeling approaches for the transmission, each of them being associated with a given "hypothesis level". From the most to the less restrictive one, these models are:

- Model 1: Constant transmission ratio via kinematic constraint
- Model 2: Torque-equilibrated chainring
- Model 3 - Motion transmission via a "rolling without slipping" kinematic constraint

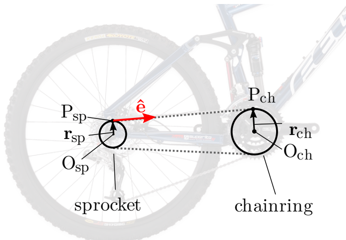


Figure 3. Motion transmission via a "rolling without slipping" kinematic constraint

Fig. 3 briefly illustrates this third model (in which the chain is still unnecessary). Instead of blindly imposing a constant transmission ratio between the front and rear angular velocities (Model 1), it is more subtle to notice that, assuming that the upper chain is permanently tight, it can be considered as a rigid body along $P_{ch} - P_{sp}$. A "rolling without slipping" constraint at the chain/chainring and chain/sprocket connection can then be formulated as follows:

The projection on the unit vector $\hat{\mathbf{e}}$ of the absolute velocities of the chain material points P_{ch} and P_{sp} must be equal at any time, that is:

$$h(q, \dot{q}) \triangleq (\mathbf{v}_{P_{ch}} - \mathbf{v}_{P_{sp}}) \, . \, \hat{\mathbf{e}} = 0, \;\; \forall t \tag{1}$$

which represents a velocity-level kinematic constraint.

The presentation will detail the three proposed models and will especially focus on their — non negligible — influence on the results, in particular the dissipated power in the rear and front suspensions.

Acknowledgements: this research work has been supported by the Belgian National Fund for Scientific Research (F.R.S.-FNRS) which is acknowledged.

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An intelligent Frontal Collision Warning system for Motorcycles

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Abstract

This article illustrate a novel Frontal Collision Warning system for motorcycles which has been developed in the SAFERIDER project [1] of the 7th EU FP, among other Advanced Rider Assistance Systems. The Frontal Collision Warning function (FCW) described here is based on a holistic approach, which combines road geometry, motorcycle dynamics, rider input and riding styles. The warning strategy is based on the correction of longitudinal dynamics derived from a previewed manoeuvre) continuously computed from the actual state of the vehicle. In normal driving conditions the reference manoeuvre fairly match with the rider one and no correction is necessary therefore no warning is produced. However, when large differences between actual and ideal accelerations are found the rider is warned to decelerate or brake. As soon as the correct value of deceleration is achieved the warning disappears improving the system acceptability. Warnings are given to the rider via a proper combination of haptic, visual and audio signals thanks to specific HMI device, which include an haptic handle among, a vibrating glove, a smart helmet, and a visual display.

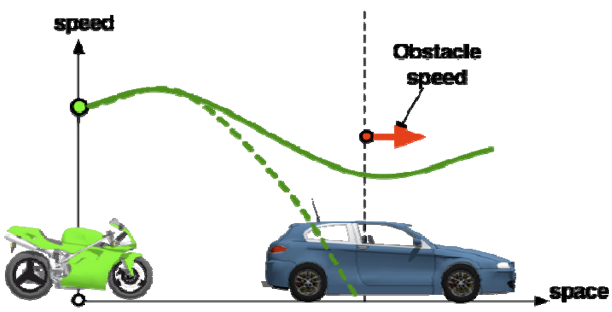

Figure 1. Typical scenario managed by the Frontal Collision Warning function

A typical scenario managed by the FCW function is shown in Figure 1: the motorcycle is running on a straight road when suddenly a vehicle ahead brakes, or a new vehicle cut in on the lane. In both cases the remarkable speed difference between the motorcycle and the obstacle ahead is a potential danger. In this situation, the FCW aims at suggesting the more appropriate action for the correct longitudinal control of the vehicle.

The FCW function calculates a preview "optimal-safe" manoeuvre based on a dynamic optimization approach which accounts for:

- an appropriate mathematical model of the motorcycle dynamics;
- an estimation of the actual dynamic state of the motorcycle;
- a model of the road geometry and attributes;
- the relative position and speed of the obstacle ahead
- riding safety, comfort and style
- the calculation of the riding risk

Figure 2 summarizes the three layers (perception, decision and action) architecture that the FCW function shares with other SAFERIDER functions.

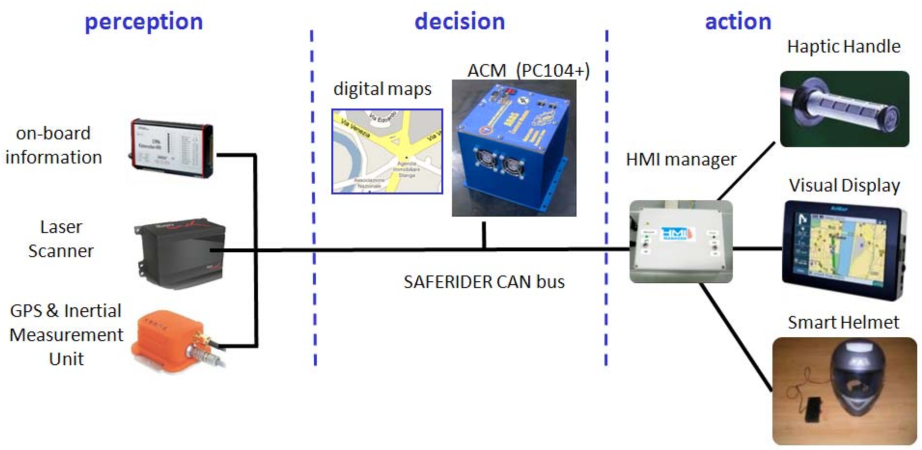

Figure 2. architecture of the Frontal Collision Warning System

The perception layer comprises sensors for the measurement of vehicle state and includes a GPS device, an Inertial Measurement Unit (IMU), a Laser Scanner and a Vehicle Interface module (VIF), which interface to the SAFERIDER CAN bus vehicle built-in sensors like speedometer, brake pressures and others. The action layer consist in the ARAS Control Module (ACM), which manages ARAS software and interacts with the other SAFERIDER systems, finally the action layer includes the HMI manager and a set of HMI elements: the visual display, the haptic handle and the smart. The HMI manager processes the warning provided by the ACM and properly activates the different HMI elements.

The article will explains in details the Frontal Collision Warning (FCW) concept, discusses the implementation aspects and presents preliminary tests.

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Modeling of a Motorcycle using Multi-Body Dynamics and Its Stabilization Control

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Abstract

In this paper, a new rider-motorcycle system including front and rear suspensions [1], [2], [3] is modeled using multi-body dynamics, and then the stabilization control system is designed based on the multi-body dynamics model. We have already modeled the rider-motorcycle system taking into account of the lean angle of the rider's upper torso [4], [5], however, including the front and rear suspensions will be necessary for dynamical analysis of a motorcycle during braking in a turn or straight running.

A dynamical model of the nonlinear twelve-degree of freedom rider-motorcycle system derived as shown in Figure 1. In addition to the lean motion of the rider's upper torso: θ_{wx} rotating around the x-axis of the rear frame of the motorcycle, the steering angle: δ and the rotation of the front and the rear wheel, this model includes the compression length of the front suspension: l_{UD} and the compression angle of the rear suspension: ψ, which are restrained with a spring and a dumper respectively. The rider's upper torso is connected to the handle with a spring and a dumper.

A front-steering assist control [5], [6], [7] stabilizes the motorcycle against applied impulse disturbance on the front wheel. For driving in a straight line at low speed, references [6] and [7] have experimentally verified stability with this control. Figure 2 shows a simulation result of the front-steering assist control during rear braking in a turn; (a) roll angle, (b) steering angle, (c) tu-

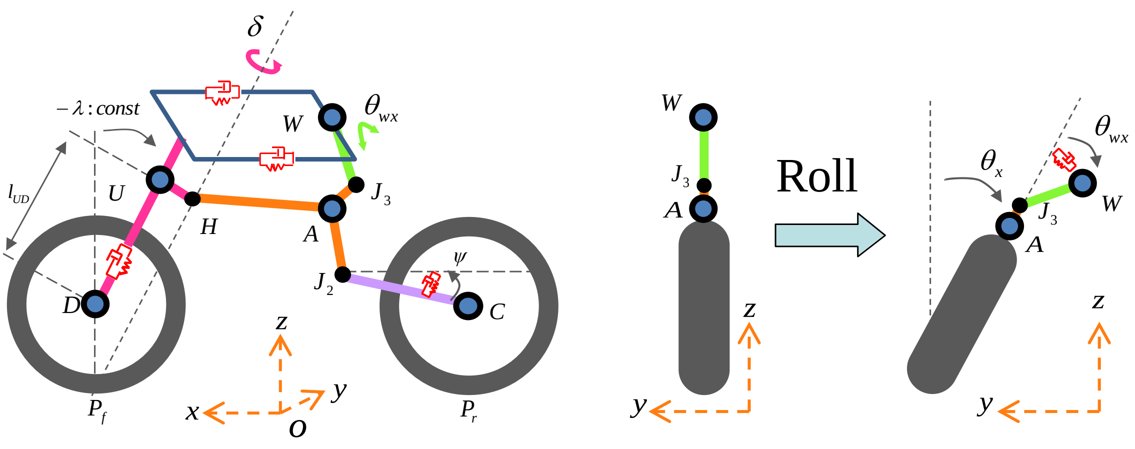

Figure 1 Dynamical Model of a rider-motorcycle system

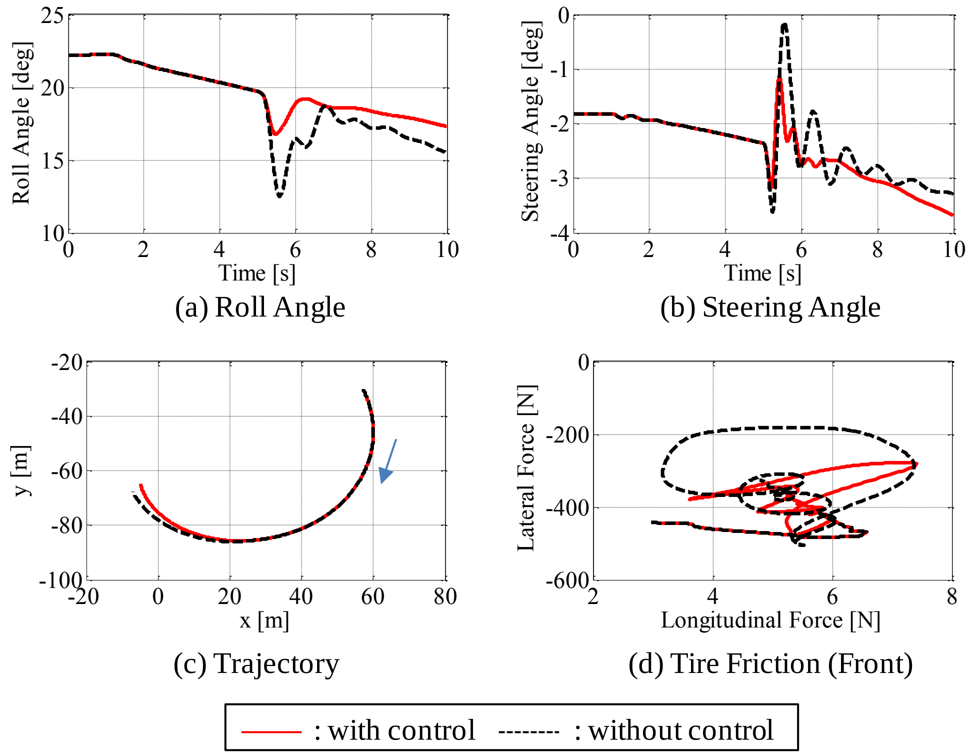

Figure 2 Simulation result with and without a steering assist control during rear braking in a turn

rning trajectory and (d) tire friction force. In the simulation, the speed of the motorcycle is decelerated from 50 km/h to 30 km/h, and 10 Nm of impulse disturbance is given to the steering torque. The controller [5] designed for steady circular turning aims to stabilize the motorcycle against applied impulse disturbance on the front wheel even during rear braking in a turn. With the derived dynamical model, we adopt the front-steering assist control to braking situations and show further investigation of the stabilization control using simulation.

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Experimental investigation on the shimmy phenomenon

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Abstract

Dynamic behavior of motorcycles, in terms of out of plane vibration modes, is strongly influenced by oscillations that can arise between the front and rear frames around the steering axis. Idealizing the behavior for a very large rear frame mass compared with the front frame, the shimmy phenomenon takes place [1].
This paper presents a basic experimental investigation on the shimmy phenomenon of a scooter front assembly. The tests were conducted at the DiME laboratory, adopting a test rig for different sets of parameters that influence the dynamic behavior of system.
The castor adopted for the investigation (Fig.1) is derived from a commercial scooter front assembly; it is joined to a rigid steel frame by means of a support that allows the castor to vertically translate and rotate around its steering axis the wheel rolls on a flat track belt [2].
The castor rake angle can be regulated by means of screw adjustment.
The vertical load, acting on the castor support, is set by means of a wire rope and a system of pulleys.
As the fork stiffness increased with the vertical load, due to the shortening telescopic suspension, the suspension springs were removed and replaced with rigid spacers. In this way, the influence of the vertical load and fork stiffness can be evaluated separately: indeed, they have opposite effects on the shimmy phenomenon. In particular, two castor configurations were considered, characterized by the maximum and the minimum extension of the fork respectively.
Finally, the castor was equipped with a 18 position (clicks) adjustable steering damper.
A basic survey was conducted to highlight the influence of some experimental parameters on the system dynamics; the tests were performed with different values of belt velocity for each castor configuration. The system was perturbed by hitting the handlebar extremity and noting if the shimmy oscillation arose; the steering damping was then tuned by setting the "click" in position where the system is stable.
Considering the force exerted by the steering damper, in the damper velocity variation range, approximately linear, a constant damping value was estimated for the first 9 damper positions. Taking into account the distance between the damper axis and the steering axis, the rotational damping coefficients σ, associated with each damper position click are:

click	0	1	2	3	4	5	6	7	8
σ [Nms/rad]	0.615	0.661	0.723	0.938	1.29	1.44	1.75	2.20	2.61

The investigation results are shown in the form of diagrams reporting the damping required to stabilize the system versus the belt speed, for the different castor configurations.
The castor configurations that were different from the nominal one were obtained by acting on the following parameters: the fork stiffness (both bending and torsional separately); the castor moment of inertia about the steering axis (I_c); the wheel moment of inertia (I_w), the rake (ε); the vertical load (N) and the tire characteristics.

As an example of results, Figure 3 shows the stability curve corresponding to the following parameter values: ε=27°; p=1.2 bar; I_c=0.37kgm2; I_w=0.324 kgm^2; maximum fork extension and two values of the additional vertical load: 20 daN and 40 daN.
It shows that the shimmy oscillation arises over a certain speed value and then, for greater speeds, thanks to the gyroscopic effect, it disappears [3] and no additional damping is required to stabilize the castor.
The upper curve (40 daN) in Figure 3 was adopted as a reference result and is reported in the other stability diagrams obtained by modifying one parameter at time to highlight and compare the influence of the modified parameter.
Figure 4 shows an FFT waterfall performed on the angular steering sensor signal, obtained for a given damping value and for different belt velocity values. The shimmy frequency has increased with an almost linear law from 5 Hz (at 10km/h) to 7 Hz (at 65 km/h) The straight line in the diagram represents the frequency which is synchronous with rotating angular speed.
The authors are very grateful to Professor V. Cossalter for his support throughout this research project.

Figure 1 - Castor on belt rig

Figure 2 – Handlebar with steering damper

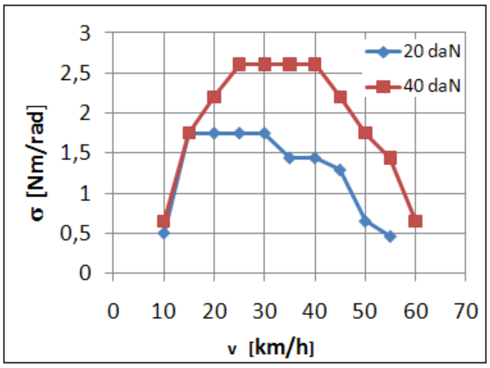

Figure 3 – Stability curve

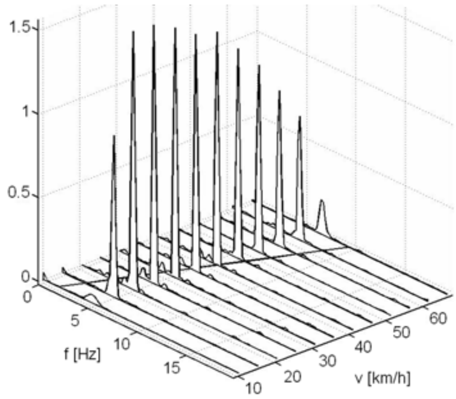

Figure 4 – FFT waterfall

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An eleven degrees of freedom dynamic model of a motorcycle

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Abstract

A motorcycle model with eleven degrees of freedom has been derived. The model is a set of nonlinear differential equations that describe the motion of the motorcycle as a function of the forces acting on the separate bodies. The model derived differs from most models in literature in that the rolling wheel is not modeled as a nonholonomic constraint, but includes a simple linear tire model, that is capable of predicting tire forces at large camber angles. The wheels are not restricted to the ground surface, so that wheelies and stoppies can be simulated with this model. Furthermore, the model is fully nonlinear and lateral and longitudinal dynamics are united.

A symbolic expression for the nonlinear equations of motion has been obtained. The motorcycle model that has been created has 11 degrees of freedom, 6 degrees of freedom for the motorcycle as an object in 3 dimensional space, rotation of the wheels, front and rear suspension, and the freedom to steer. The model originates from Lagrangian Mechanics. The way of modeling is originated from robot modeling [2], where the tire contact patches have similar meaning as the robot end effector. The model includes a dynamic mass matrix, a Christoffel matrix, and a set of nonlinear functions in the generalized coordinates and generalized velocities describing the tire forces, suspension forces, brake torques, engine torque, and steering torque. The Mass matrix contains about 18.000 characters. The Christoffel matrix 80.000, and the applied forces are expressed in 360.000 characters, from which 240.000 characters are in use by the front wheel contact patch force and moment. This figure can go down by 90 % by using subexpressions, and a different set of generalized coordinates.

In the equations of motion, steering torque, engine torque, and brake torque are used as control inputs. Furthermore, the model has been linearized and compared with the model of Koenen [3], and with the Jbike6 [4]. A bifurcation diagram has been made where the eigenvalues are calculated for different forward speeds. The eigenvalues with the parameters of the Jbike6 inserted in the model can be seen in figure (2). Comparison shows a positive correlation with both models. Furthermore, the model is compared against another nonlinear model, made in SimMechanics. This model shows similar dynamics to machine precision accuracy.

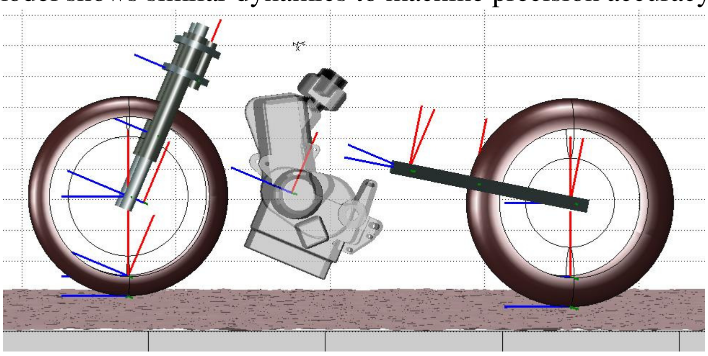

Figure 1. Motorcycle side view.

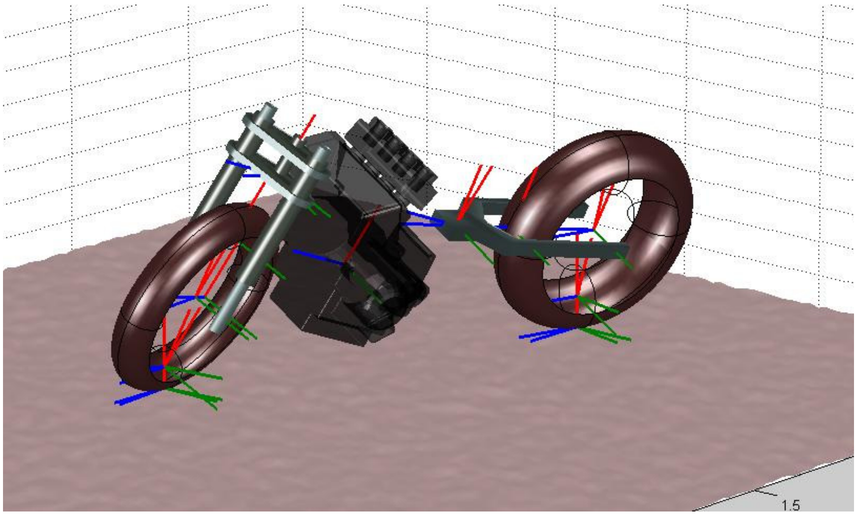

Figure 2. Three dimensional view of the motorcycle model

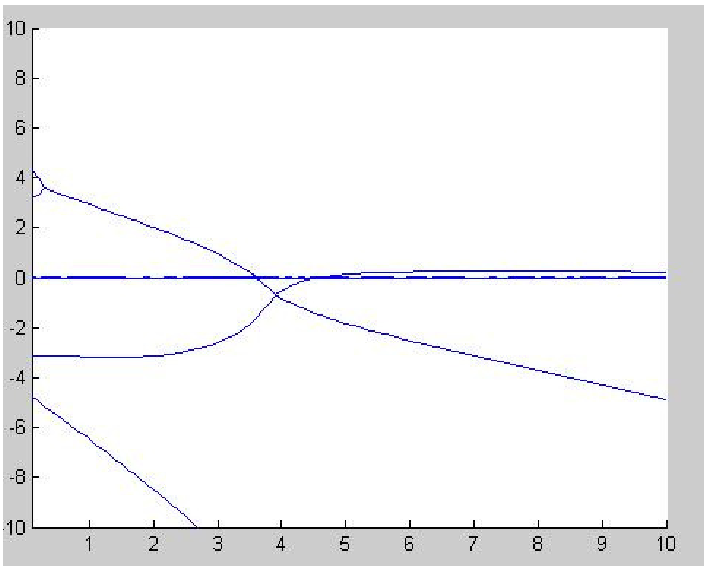

Figure 2. Bifurcation diagram of the linearized model with the Jbike6 parameters inserted

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Automatic Generation of Linearised Equations of Motion for Moving Vehicles

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Abstract

This paper demonstrates a method for automatic generation of the linearised equations of motion for mechanical systems; in particular, one that is well suited to vehicle stability analysis. Unlike conventional methods for generating linearised equations of motion in 'MCK' form, the proposed method allows for the analysis of systems with nonholonomic constraints, and allows linearisation around non-zero speeds. With this method, the algebraic constraint equations are eliminated after the linearisation and reduction to first order. The method has been successfully applied to an assortment of problems of varying complexity.

The linearised unconstrained equations of motion, combined with the linearised kinematic differential equations, are given in Equation (1).

$$\begin{bmatrix} \mathbf{I} & \mathbf{0} \\ \mathbf{0} & \mathbf{M} \end{bmatrix} \begin{Bmatrix} \dot{\boldsymbol{p}} \\ \dot{\boldsymbol{w}} \end{Bmatrix} + \begin{bmatrix} \mathbf{V} & -\mathbf{I} \\ \mathbf{K} & \mathbf{C} \end{bmatrix} \begin{Bmatrix} \boldsymbol{p} \\ \boldsymbol{w} \end{Bmatrix} = \begin{Bmatrix} 0 \\ \boldsymbol{f}_c + \boldsymbol{f}_\mathrm{a} \end{Bmatrix} \tag{1}$$

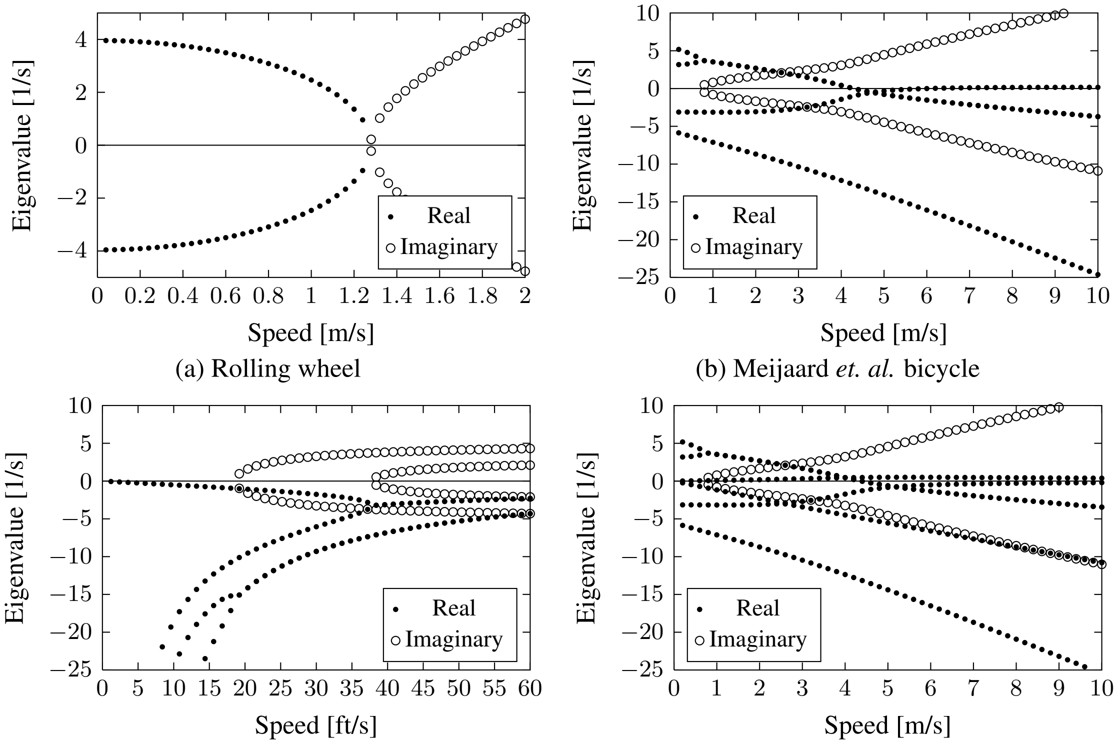
The $\mathbf{V}$ matrix results from the linearisation of the kinematic differential equations, and contains the skew symmetric matrix of the constant linear velocities of the bodies, arranged in the upper right 3x3 sub-matrix of the set of 6x6 matrices arranged along the diagonal. All other entries are zero. The $\mathbf{C}$ matrix contains the traditional damping matrix, plus terms due to the inertia forces, i.e., centripetal forces and gyroscopic moments. The stiffness matrix $\mathbf{K}$ is the sum of terms resulting from deflection of elastic elements and terms resulting from preloads in the system, i.e. the tangent stiffness matrix. The applied and constraint forces appear in the right hand side. The mass matrix $\mathbf{M}$ results from Newton's Laws, and is tri-diagonal as is typical.

The linearised constraint equations are written using a state vector combining both positions and velocities. The positions are expressed in a fixed global reference frame, where the velocities are given in a body fixed moving reference frame. When expressed in this form, identical coefficients describe the constraints as applied to global velocities and local accelerations. Because the positions and velocities are given as separate states, the holonomic constraint equations are applied twice; first to the positions, and again, in differentiated form, to the velocities. The nonholonomic constraints are applied only to the velocities. The combined constraint equations are given in Equation (2).

$$\begin{bmatrix} \mathbf{B}_\mathrm{h} & \mathbf{0} \\ -\mathbf{B}_\mathrm{h}\mathbf{V} & \mathbf{B}_\mathrm{h} \\ \mathbf{0} & \mathbf{B}_\mathrm{nh} \end{bmatrix} \begin{bmatrix} \dot{\boldsymbol{p}} & \boldsymbol{p} \\ \dot{\boldsymbol{w}} & \boldsymbol{w} \end{bmatrix} = \begin{bmatrix} \mathbf{0} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} \end{bmatrix} \tag{2}$$

The $\mathbf{B}_\mathrm{h}$ and $\mathbf{B}_\mathrm{nh}$ matrices represent the holonomic and nonholonomic constraint equations, respectively. An orthogonal complement matrix is used to eliminate the constraint equations and constraint forces, and similarly to define a new minimal system of coordinates. The method is explained in detail, and combined with a genetic search algorithm to find parameters that stabilise a narrow tilting vehicle in [1]. The method has been implemented in the MATLAB$^{\mathrm{TM}}$/Octave programming language, under the title 'EoM', and is freely available under the GPL licence on the author's website.

The results produced have been verified against a number of benchmark problems from the literature, such as the rolling wheel (r=0.5 m) illustrated in Greenwood[2], the Meijaard *et. al.* rigid-rider bicycle[3], and the Ellis truck and trailer[4], as shown in Figures 1a, 1b, and 1c, respectively. More recently, the method has been applied to a bicycle and trailer combination; results are shown in Figure 1d.



(a) Rolling wheel

(b) Meijaard *et. al.* bicycle

(c) Truck and trailer

(d) Bicycle with trailer

Figure 1: Eigenvalues vs. speed

The bicycle used with the trailer for the model was the previously mentioned benchmark, and the trailer was modelled as a rigid body, attached to the bike by spherical joint, rolling on two wheels, each identical to the rear wheel of the bicycle. The properties of trailer are given in Table 1, using the coordinate system from the benchmark.

Table 1: Trailer parameters

mass	15 [kg]	centre of mass	-0.75,0,-0.4 [m]	tow hitch	0,0,-0.3 [m]
I_{xx}, I_{yy}, I_{zz}	1,1,3 [kg·m^2]	left wheel	-0.9,-0.3,-0.3 [m]		
I_{xy}, I_{yz}, I_{zx}	0,0,0 [kg·m^2]	right wheel	-0.9,0.3,-0.3 [m]		

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Assessing slip of a rolling disc and the implementation of a tyre model in the benchmark bicycle

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Abstract

For the development of a real time bicycle/motorcycle model a compact formulation of the dynamic behaviour is required. The modelling environment MatLab SimMechanics was posed as a prerequisite.

For the development of the tyre model the wheel was considered as a disc with knife edge contact to the ground. A tyre model typically uses slip quantities as the input and calculates forces and moments as outputs. Longitudinal- and lateral slip are calculated as the components of a normalized slip velocity in the contact point. The location of this contact point can be denoted with a position vector **r** pointing from the wheel disc centre to the contact point.

The assessment of the slip quantities has been based entirely on the vector calculation presented by Pacejka in [1]; the contact point can be found in radial direction $\mathbf{e}_r$ at a scalar distance r from the wheel centre. Here $\mathbf{e}_r$ is recursively defined as being orthogonal to the axial $\mathbf{e}_s$ and longitudinal direction $\mathbf{e}_l$, where the longitudinal direction is the intersection of the road plane and wheel plane, thus the vector orthogonal to road normal **n** and axle direction $\mathbf{e}_s$. See Figure 1

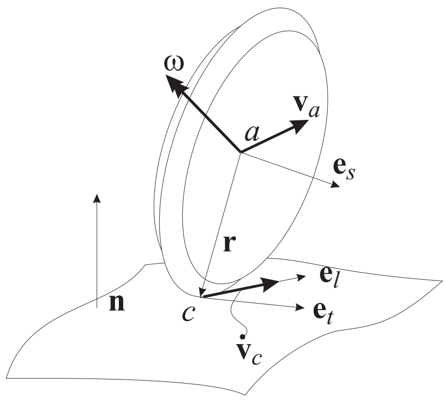

Figure 1. The wheel disc with contact point location **r**

In SimMechanics a single rolling disc has been modelled. With the motion of the centre of the wheel disc and the vector **r** available, the derivation of the velocity of the material point s on the wheel, momentarily in the contact point c, is straightforward: $\mathbf{v}_s = \mathbf{v}_a + \boldsymbol{\omega} \times \mathbf{r}$. This velocity of the point s needs to be normalized with the wheel speed v_x, to obtain the slip quantities α and κ that are the inputs for the tyre model.

$$\lambda = \frac{\mathbf{v}_s}{v_x}, \quad \tan\alpha = -\lambda_y, \quad \kappa = -\lambda_x$$

A simple linear tyre model defines the dissipative tyre contact forces, proportional to the established slip quantities. However normalizing slip with the wheel-centre longitudinal velocity $v_x \rightarrow v_{a,x}$ is practical, but incorrect. We show that the singularity caused by dividing by zero axle speed, poses difficulties for the numerical solvers when simulating the wheel as an Euler disc. The correct velocity to use in the normalization of slip is the propagation speed of the contact point $v_{c,x}$, which comprises $v_{a,x}$ and a term with $\dot{\mathbf{r}}$. Since $\mathbf{r}$ is the result of sequential cross products, its time derivation is awkward, yet possible.

Using the propagation speed in the denominator, the single wheel behaves as an Euler disc in simulation. Due to the finite slip stiffness assumed in the model the quantative behaviour deviates somewhat form the Euler disc with a non-holonomic constraint in the contact. The wheel model is further elaborated by implementing first order relaxation equations that enable more realistic transient tyre behaviour. The introduction of the relaxation equation is also known to facilitate simulating at zero velocity.

Two of these wheel-disc models were assembled with a rear-frame and a front-fork body in SimMechanics to build a so-called Whipple bicycle model [2]. With the bicycle model parameterised according to the benchmark, numerous time simulations have been carried out. The roll angular velocity response has been used to curve-fit a standard exponential response. The eigenvalue can be found as from the (complex) exponent of the optimally fitted time response. The eigenvalues obtained for many velocities can be visualized in the so-called root-loci plot. The figure shows excellent correspondence between the benchmark Whipple bicycle with kinematic rolling constraints, and our bicycle with linear transient tyre models.

The difference with kinematic rolling will be exaggerated by reducing the slip stiffness in our model. Also the effect of transient, compliant tyre behaviour will be studied by simulating the model using increased relaxation lengths. The conclusion is that the model can be parameterised to match the kinematic rolling, and can be used to as a starting point to implement more realistic (non-linear) tyre behaviour when required for specific (future) applications of this bicycle model.

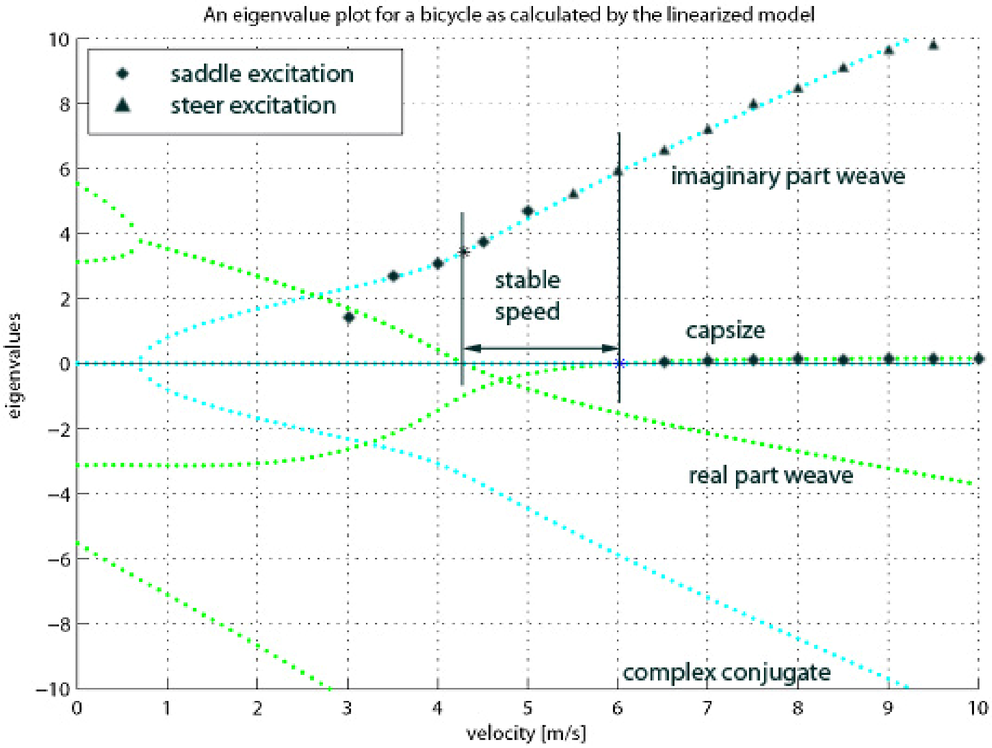

Figure 2. The root loci of the benchmark bicycle and a multi-body model of a Whipple bicycle with linear transient tyre models. The multi-body model has been excited on the saddle, eigenvalue marked with ♦, and on the handlebar, marked with ▲.

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Three Structural Component Linkage Front Suspension and Directly Connected Suspension for Motorcycles

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Abstract

Directly connected suspension was first tried and tested on a motocrosser to combat the menace of ‘square-edged’ bumps, which react to throw the rider over the handlebars. The mathematical calculations prior to testing this new design gave a 25-30% reduction in chassis pitch. Only, these calculations did not take into account the fact directly connected suspension upon reducing chassis pitch puts greater emphasis on the shock absorbers to perform more efficiently for any given settings and any given dynamic condition thereby further reducing chassis rotation. A superior linkage front suspension, other than the one first tried on the motocrosser, then had to be devised and developed if directly connected suspension was to become successful. In recent times, a three structural component one-sided design was tried and tested where around 40 advantages over conventional telescopic fork and (unconnected) single shock absorber rear suspension design were achieved. Only, the single-sided nature of such a design was not appropriate for strength and stiffness in relation to component weight nor was it appropriate to achieve equal degrees of lateral flex from either side of the bicycle/motorcycle. There was also restricted steering lock. Given all this, the lower ball joint was repositioned from inside the front hub to above the front tire where the same basic design was retained. This is the most successful design tested to date. It was imperative rider feel was retained at a level comparative to, or above, that of telescopic forks where both these three structural component linkage front suspension designs have the handlebars move slightly with suspension travel amplifying feel for the rider. Simplicity was the final consideration – three structural components, two ball joints and two sets of rotational bearings is all that these front suspension designs utilize, which could be considered as basic as telescopic forks are.

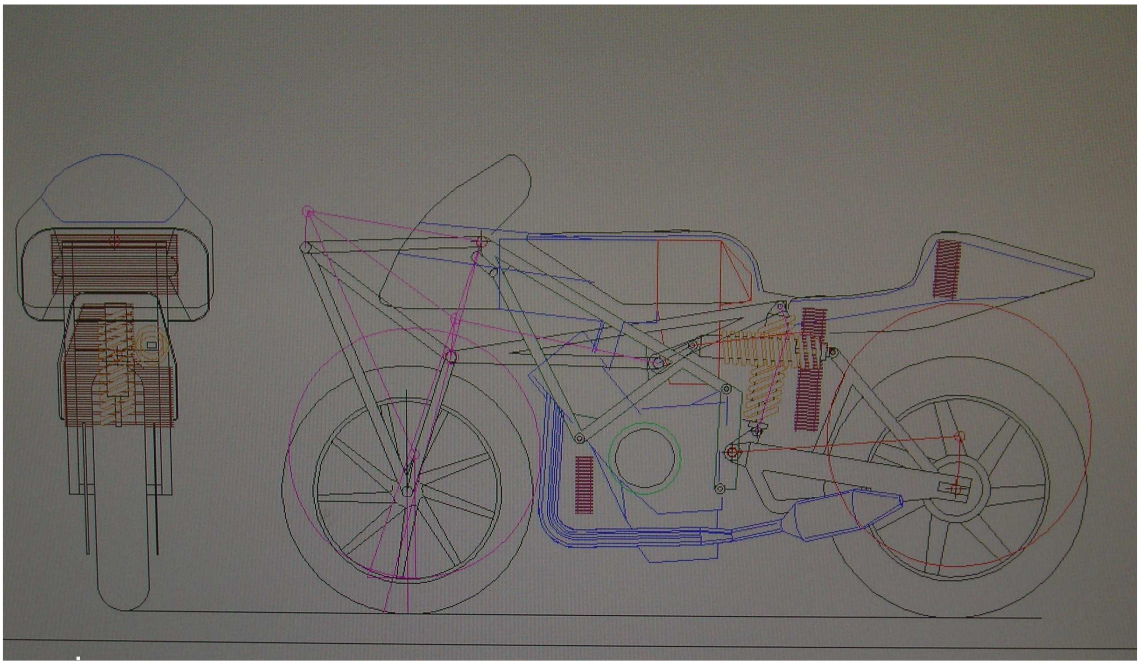
Directly connected suspension works by transferring a limited amount of motion load and weight onto the opposing suspension system, front to rear, and, rear to front, when suspension motion is encountered with either suspension system. This gives a self-leveling effect in all dynamic conditions and also provides superior suspension action over all other suspension designs. When one suspension system encounters a bump (and is only partially absorbed; where no conventional motorcycle suspension system fully absorbs bumps encountered) the opposing suspension system extends slightly to ensure the body of the motorcycle remains close to level. Directly connected suspension has been found to work too well as it has a natural anti-wheelie aspect to it, but this can be utilized by way of shorter wheel-bases to the benefit of increased cornering potential, handling and weight saving (both sprung and un-sprung).

Electronic active suspension is currently being developed for both telescopic forks and rear shock absorbers, but given the fact the telescopic fork is now at the end of its development cycle due to excessive cornering lean from improved tire grip, and the resulting cornering ‘chatter’, the fundamental design flaws cannot be overcome, therefore such a development is only a temporary solution. Race chassis technicians can only give a compromise but cannot address the problems relating to cornering chatter and retain high speed stability. A realistic alternative design has to be found soon to take motorcycle development to the next level. At best, electronic active suspension systems on motorcycles only have the potential to change the compression and rebound dampening where the most advanced versions will alter the spring preload. But it is the spring rate and the rising rate, and/or leverage ratio, of the suspension system which need to alter with the motorcycle from being upright to leant over during cornering to give a much

needed breakthrough in chassis technology. Directly connected suspension automatically does this. When cornering forces compress both front and rear suspension the outer ends of the respective shock absorbers are then displaced further away from perpendicular thereby increasing leverage ratios to give lighter suspension action, plus giving the side benefit of less compression of the motorcycle (compression of one suspension system acts to extend the other) to assist ground clearance, which has the secondary benefit of improved aerodynamics where reduced frontal area can be achieved when motorcycle designs are produced lower to the ground.

Directly connected suspension also has the benefit of altering suspension ratios during braking to more ideal. When braking forces compress the front suspension the rear extends to become lighter in action with the front, upon compression, becoming stiffer. Again, by means of displacement of the outer ends of the shock absorbers. With the final design of three structural component linkage front suspension shown below, the front control arm pivot point and the rear swinging arm pivot point can be positioned near vertical, one above the other, where compression of the front suspension will place load onto the rear suspension system (the outer end of the front control arm with ball joint attached remains above the front tire therefore it is the opposite inward end of the control arm which has to go downwards) to the benefit of superior braking and shorter stopping distances. Further, rear wheel braking places more load on the front suspension system than can be achieved with telescopic fork designs. When the swinging arm pivot point goes downward upon braking (with the higher swinging arm pivot point than the rear wheel axle where both want to become horizontal due to rear wheel braking dragging the rear axle backwards) taking the vertically above front control arm pivot point with it thereby loading up the front suspension, again to the benefit of braking performance.

This new linkage front suspension benefits from longitudinal chassis stiffness for superior braking performance and has greater control over lateral flex making it easier to achieve a higher natural frequency to avoid cornering chatter yet provide stability for straight-line travelling.



Endorsement

[1] A letter of praise from Massimo Tamburini (designer of the *Ducati 916* and *MV Agusta F4*) dated 10/06/08.

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Some Investigations on the Wobble Mode of a Bicycle

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Abstract

Learning how to ride a bicycle demands courage. After some unfortunate experiences with a bicycle toppling over at low speeds, taking a risk to speed-up will be rewarded, as the bicycle may stabilize its motion on its own. Some more practice, and we have learnt how to handle two potentially unstable modes of the bicycle – weave and capsize.

There remains a further mode, wobble, which is – once experienced in its unstable form – at least unpleasant if not hazardous at all. Wobble is related to the more general class of wheel-shimmy, which is a self-excited motion of the wheel about the steering axis, and thoroughly treated in [1, 2]. The experience and main findings from detailed analysis of this phenomenon at motorcycles, e.g. [3], can hence be transferred to the bicycle. Although key issues on the wobble mode at bicycles are given in [4], little else can be found in scientific literature on this topic.

This paper aims to contribute to the analysis of the wobble mode by examining a specific bicycle, that shows an unstable wobble mode at certain speeds, both on the basis of a mathematical model and by experiment. The considered bicycle is equipped with measurement devices (GPS-sensor, 6 degree of freedom inertial measurement unit for accelerations and angular velocities, steering angle potentiometer, wheel-speed sensor) and is depicted in Figure 1.

Figure 1. Test bicycle with measurement equipment.

The nonlinear equations of motion for bicycle and rider are derived by hand applying d'Alembert's principle. The model considers the roll and steering angle of the bicycle, yaw rate and lateral velocity, and moves at a constant longitudinal speed. The flexibility of the frame is modeled by introducing a rotational degree of freedom, allowing the steering assembly to twist about an axis normal to the steer axis in the plane of symmetry, restrained by a spring-damper combination. The rider has just a lean degree of freedom, which is represented by the relative roll angle of the upper torso with respect to the bicycle. The applied linear tyre model considers sideslip and camber angle of the respective tyre, and accounts for the time lag in a transient condition of the tyre. Details on the derivation of the system equations and the corresponding system model are presented in [5].

Figure 2 shows a time-sequence of the steering angle of a typical test run. The rider was sitting tight in an upright position, hands-off, and did not pedal after attaining the initial speed of about 7 m/s. The dominant frequency in the signal is about 6 Hz. At the end, intervention of the rider was inevitable.

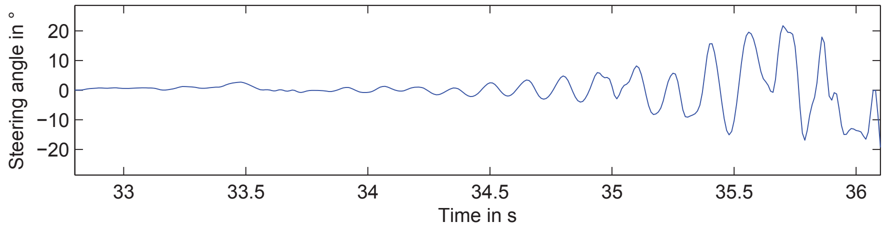

Figure 2. Measured steering angle of the bicycle with initial speed of about 7 m/s.

For stability analysis the system equations are linearized with respect to the straight running condition at constant speed, and eigenvalues are analyzed.

Mass and geometric properties of the bicycle have been measured, as well as the lateral stiffness of the main frame assembly. Other crucial parameters for the standard configuration of the bicycle, like the damping coefficient of the frame and some tyre parameters, have been estimated within the system model to meet the transition-velocity from stable to unstable wobble mode and the corresponding frequency found by test runs. Subsequently, the sensitivity and influence of various design parameters of the bicycle on the on-set of the unstable wobble mode are discussed and compared with measurements if possible.

Preliminary results give evidence that the key design parameters that influence the wobble mode are frame damping coefficient, mass distribution, geometric properties of the front frame assembly and tyre parameters in the mentioned order.

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Comparison of a bicycle steady-state turning model to experimental data

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Abstract

The design of the modern bicycle is the result of almost 200 years of trial and error. Recent work has helped us to understand the stability of a bicycle and has shown that the current bicycle configuration could be made more stable with relatively small adjustments to standard bicycle geometry [1]. However, stability is not the only characteristic that a human rider desires; a bicycle also needs to be maneuverable.

To examine maneuverability, we begin with the steady-state handing of a vehicle. In particular, one can examine the gain or sensitivity of a vehicle output (lateral acceleration) to a vehicle input (steering torque), which has important implications for human control [2, 3]. The goal of this study is to develop a parallel steady-state handling model of a bicycle and rider and an instrumented bicycle to test model fidelity.

We develop a steady-state handling model for a bicycle with rigid rider making a turn of constant radius with constant speed, similar to the linear steady-state model developed by Pacejka [4]. Relative to the dominant forces (lateral tire force and weight), we neglect air drag, longitudinal tire forces, and vertical tire moments. We assume the turn radius is much larger than the bicycle wheelbase and assume small steer and roll angles (≤ 5°). The equations governing steady-state handling are derived from Newton's law in the lateral direction and moment equilibrium about the vertical axis. The tires obey the linear elastic tire model

$$F_{yi} = C_{F\alpha i}\alpha_i + C_{F\gamma i}\gamma_i \quad (1)$$

where i is an index denoting the tire (front or rear), F_y is the lateral force on the tire, $C_{F\alpha}$ is the slip or cornering stiffness, α is the side slip angle, $C_{F\gamma}$ is the camber stiffness, and γ is the camber angle. We use three sets of tire stiffness values: those from Roland [5] and Sharp [6] for elastic tires, and those for an idealized tire model ($C_{F\alpha} = \infty$, $C_{F\gamma} = 0$). The gain or sensitivity is a function of velocity as given by

$$a_y/T_\delta = u^2/(Pu^2 + Q) \quad (2)$$

where a_y is the lateral acceleration of the bicycle and rider center of mass, T_δ is the steering torque, u is the bicycle/rider forward speed, and P and Q are constants that are determined by the tire stiffness values, head tube angle, trail, wheelbase, and weight distribution of the bicycle/rider system for our instrumented bicycle. The instrumented bicycle is a standard geometry mountain bike (head angle = 72°, trail = 58mm, wheelbase = 1.047m) fitted with 1.95" x 26" slick tires (Figure 1A). The instrumented bicycle allows us to measure steer torque (load cell), steer angle (optical encoder), bicycle lean angle and rider lean angle (3-axis accelerometer), and bicycle speed (magnetic reed switch).

The experiment considered two subjects following two level circular paths (radii 13.7 and 18.3m). The measured torque and velocity data during steady-state turning was broken into 5 second blocks and averaged to create discrete data points (Figure 1B). The steady-state turning model predicts a negative relationship between lateral acceleration and steer torque, which agrees with the experimental results (Figure 1B). All models explain the variance in the experimental data equally well (Table 1, r^2), while the idealized tire model minimizes the error (Table 1, SSE). An experimentally determined model follows by fitting Eq. (1) to the experimental data to find P and Q. Q is the only significant parameter of the model ($\alpha = 0.05$), and it is also significantly different from the Q of the other models ($\alpha = 0.05$).

Table 1. Comparison of different steady-state models and the fit to experimental data

Model	***P***	***Q***	**Sum of squares due to error (SSE)**	**r^2**
Idealized tire, subject 1	0	-5.806	87.73	0.726
Roland [5] tire, subject 1	0.0084	-5.779	101.23	0.726
Sharp [6] tire, subject 1	0	-5.677	95.80	0.726
Experimental data fit, subject 1	0.006707	-7.473	52.74	0.726

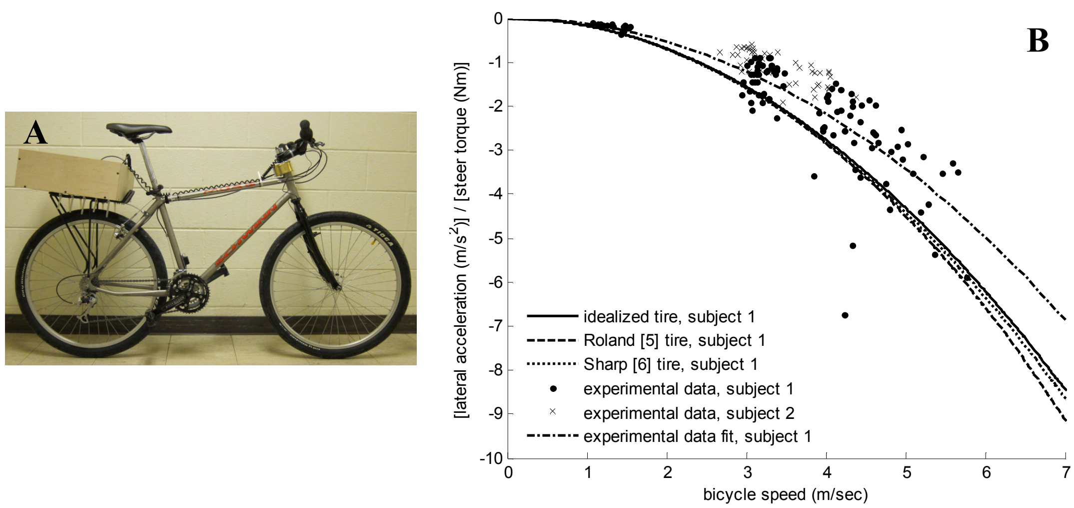

Figure 1. (A) The instrumented bicycle **(B)** The ratio of lateral acceleration over steer torque versus speed; the results of the model for three sets of tire parameters are plotted with the experimental data of 2 subjects and a best fit to the experimental data of subject 1.

Via the instrumented bicycle, we show that a steady-state turning model provides a good fit to experimental data. The model indicates that tire parameters do not have a pronounced effect on the control strategy that must be employed by a human rider.

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Steering Characteristics of Motorcycles

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Abstract

As the motorcycle market continues to mature and the need for greater safety and running performance increases, more and more makers are installing electronic control devices used in four-wheeled vehicles to control vehicle dynamics. While the installation of electronic control devices makes it possible to fine-tune the performance of motorcycles, the development cost for selecting the optimum control parameters has problematically risen. Therefore, in order to efficiently optimize the control parameters, a development method is needed that can predict the maneuverability for each control parameter and rationally select the optimum parameters from the viewpoint of maneuverability.

Enabling the development of such a method will require a simulation technology that can predict the ride of the vehicle based on its characteristic values and indexes that can be used for determining whether the predicted ride is good or bad. As a matter of fact, many simulation technologies have been studied. However, comparative validation between their results and actual ride data has rarely been published. Kageyama et al. [1] proposed a steering index for motorcycles and showed its theoretical background as well as the differences from the steering index for four-wheeled vehicles. So far, however, activities to validate the index against actual ride data have been insufficient.

One of the reasons why validation using actual ride data has not progressed is the fact that the slight shift in the motorcycle rider's operations and in the way the rider moves his/her body significantly changes the vehicle dynamics characteristics, making it difficult to obtain measurements with a high level of repeatability. However, if we focus our measurements only on the steady-state characteristics, we should be able to restrict the rider's influence to the shifts in his/her center of gravity position according to posture differences, and therefore should be able to obtain measurements with a high level of repeatability.

Therefore, our research restricted the ride pattern to steady-state cornering. At the same time, we expanded our measurement scope to include tire force, tire moment, and tire slip angle, all of which are presumed to greatly contribute to the maneuverability.[2] For the tire slip angle measurement data in particular, we used two methods to validate their accuracy and compared them with the simulation results. Next, we examined a case in which the rider's lean posture, which significantly impacts measurement results, is intentionally changed to a significant degree. Additionally, based on the actual measurement data, we calculated steering indexes for the steering characteristics (stability factors), tire slip characteristics, and hold-steering torque characteristics, and finally, validated them.

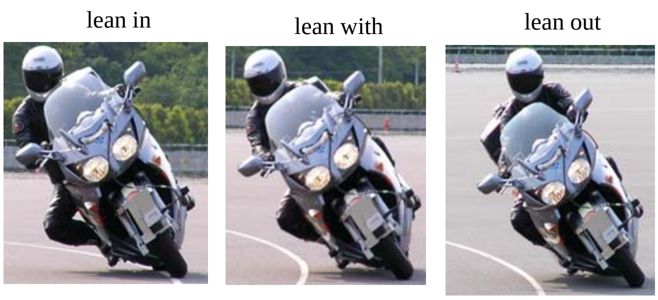

Figure 1. Rider’s posture difference

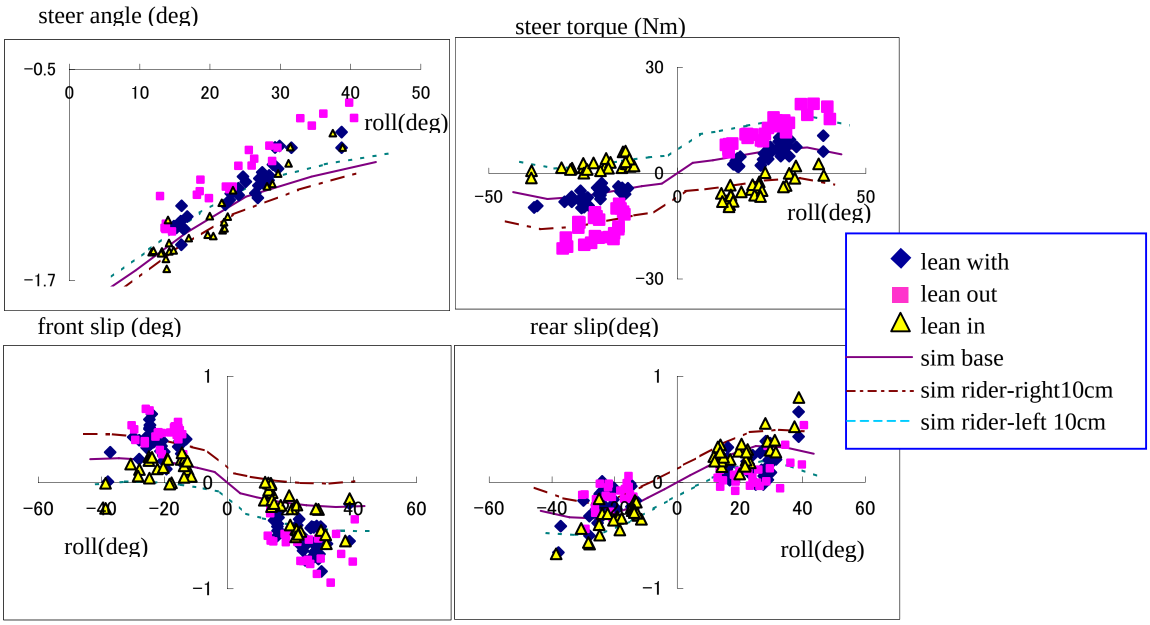

Figure 2. Averaged data for steady-state turning changing rider lean

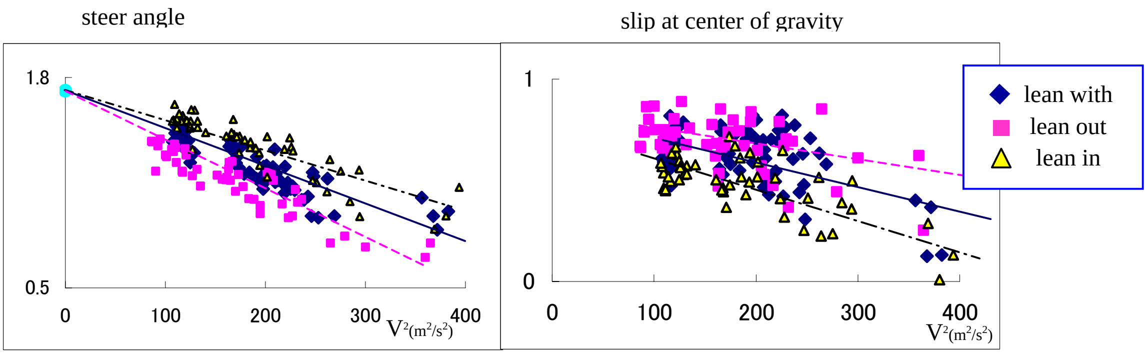

Figure 3. Averaged data for steady-state turning arranged to calculate steering idexes

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On the Validation of a Motorcycle Riding Simulator

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Abstract

This paper illustrates the characteristics of a top-of-the range motorcycle simulator and focuses on its objective and subjective evaluation. The simulator has been designed and built at the University of Padua over the last years; it consists in a motorcycle mock-up with functionally working throttle, brakes, clutch and gearlever mounted on a five degrees of freedom platform, a real-time multibody model of the motorcycle, and a dedicated audio and visual systems. Simulator purposes are to test in a controlled, safe environment devices such as ABS, Traction Control and other ARAS, to study riders behaviour and to train them. To extend results obtained on the simulator to the real world, it has been developed an innovative procedure for the objective and subjective validation of motorcycle simulators and it has been applied to the presented simulator.

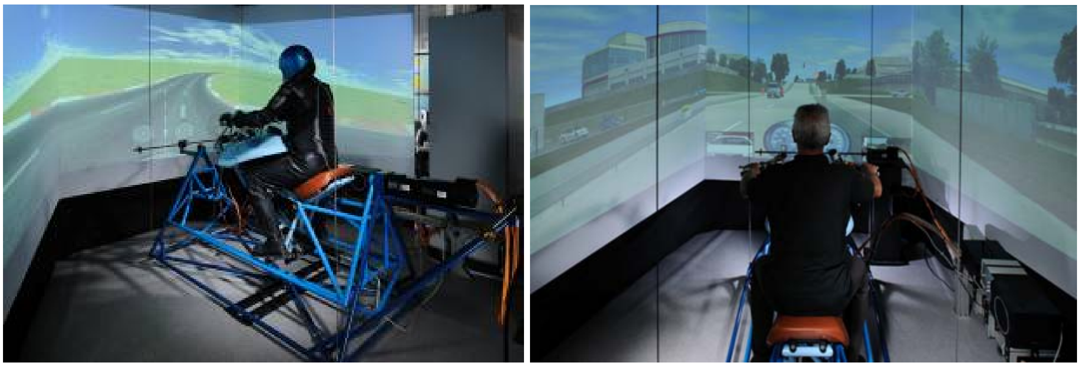

Figure 1. The UNIPD riding simulator

The **objective evaluation** consists in the comparison between the behaviour of the real and virtual motorcycle in presence of the same riding actions. Even if there are many riding conditions with several uncontrolled parameters, the literature about the objective evaluation of motorcycle handling characteristics [1-5] help us to focus on selected manoeuvres that are representative of the more general vehicle behaviour. In particular, the following three typical manoeuvres has been selected for the evaluation of riding feelings:

- Slalom (three different cone distance)
- Lane change (two different lane geometry)
- Steady turning (three radii)

The objective evaluation have been performed by comparing objective data collected in real and virtual maneuvers.

The aim of **subjective evaluation** is the enhancement of the riding sensations in term of visual, acoustic and motion cues. It is worth to highlight that each different kind of cues has different physical and technological limitation, in particular for visual cues there are limitation due to the fidelity of the scenario representation, as well as technological limitation in term of resolution and brightness of the visual devices; for acoustic cues there are technological limitation in the reproduction of the environment sound and noise; for motion cues there are both technological and (harder) physical limitations: indeed the reproduction of acceleration is limited in amplitude and duration by the travel of the motorcycle mock-up, further limit on the acceleration frequency bandwidth are correlated to the power of simulator motor. The riding sensations of the test riders have been collected by means of a rating questionnaire, which include both technical questions and question about perception and cognitive processes. The questionnaire has been developed by the aid of two skilled riders which also are expert in motorcycle dynamics.

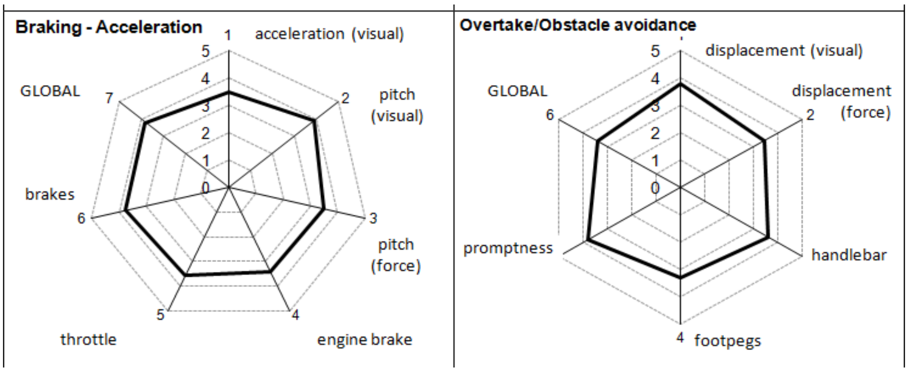

Figure 2. Subjective Rating of the Simulator

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A Virtual Rider for Reproducing Experimental Manoeuvres

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Abstract

The control of two-wheeled vehicles represents a challenging task. Indeed the stability of a bike is characterized by vibration modes which significantly change their behaviour with speed and acceleration and may become unstable under certain motion condition, see e.g. [1]-[3].

Different control strategies have been proposed in the past years, and the control of single-track vehicles remains an open research field. Nonlinear optimal control theory has proven successful, [4]-[5], but computational reasons make this approach not appealing for complex multi-boby model. PID approaches have demonstrated effective for constant speed or slowly varying manoeuvres, [6],[7]. Optimal linear time invariant controllers have been used for constant speed manoeuvres [8] and speed control [9]. The linear optimal control theory designed on a very simple motorcycle model has been used to control a complex multiboby code in [10]. A recent discussion on linear predictive control for motorcycle with constant speed simulations is devised in [11].

It is worth noting that most of the works available from literature deal with constant speed or slowly varying speed manoeuvres. On the contrary, this paper presents a virtual rider which controls the vehicle with both (strong) longitudinal and lateral accelerations. This is achieved by updating at each instant the control action based on the information of the approaching road section and the current vehicle state. This approach mimics the real rider behaviour, who looks ahead, learning a portion of the track, continuously using this information to decide when/how to steer and accelerate. In more detail, the approach is based on the predictive control theory framework. At each step the virtual rider computes the control action using an appropriate linear model which is derived from the full nonlinear multiboby model, see Figure 1.

As an example of application, the virtual rider is used to reproduce a lap of the Adria circuit (Italy): real bike speed and roll angle are used as target motion by the virtual rider, which controls the vehicle with longitudinal acceleration up to 1 g and roll angle up to 50°. In particular, in Figure 2 the roll and speed of the real vehicle (circles) are depicted together with the roll and speed of the multiboby model (solid line) ridden by the virtual rider. It is worth noting that the tracking is almost perfect, both in terms of speed and roll angle.

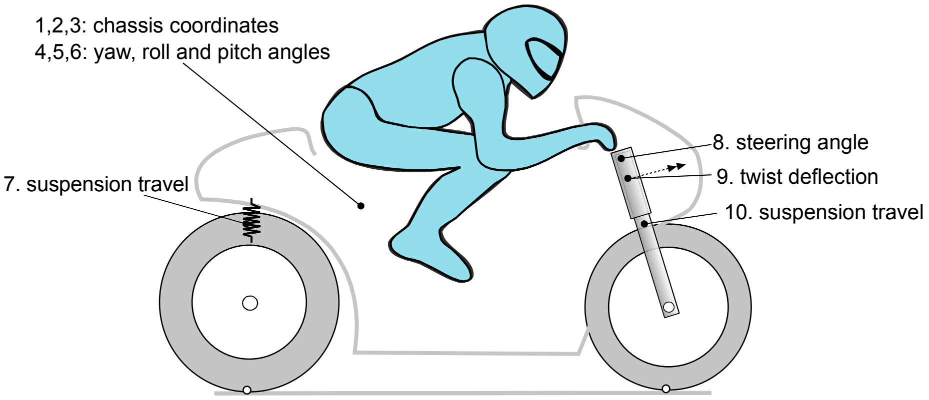

Figure 1. Degrees of freedom of the motorcycle model used for simulation.

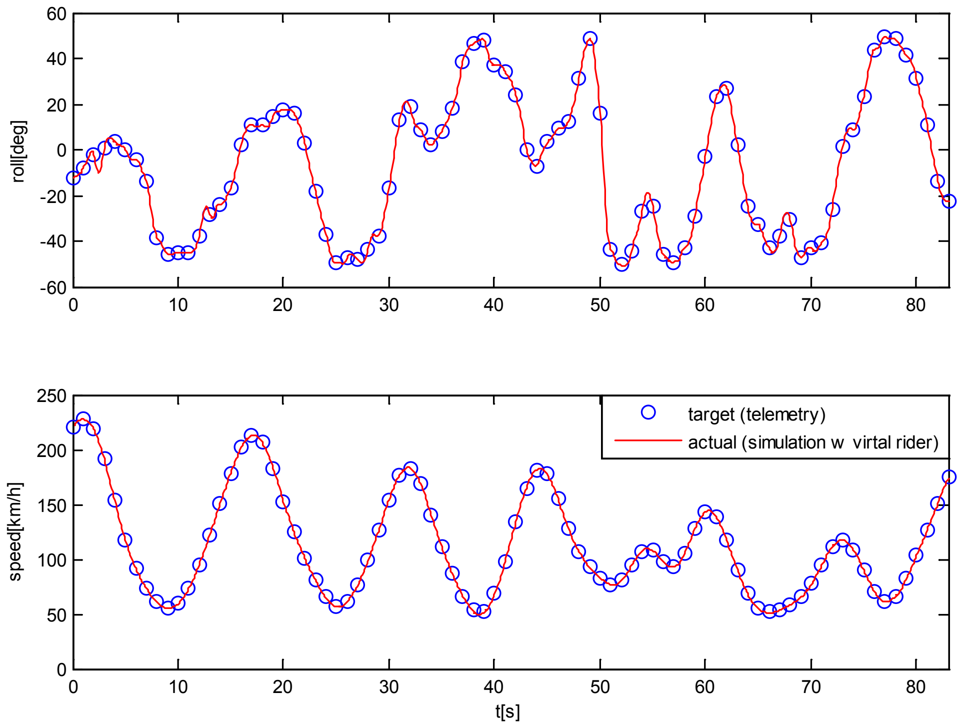

Figure 2. Comparison between simulation results (multiboby model with virtual rider, solid red line) and telemetry logged at the Adria circuit (blue circles).

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A bicycle model for education in machine dynamics and real-time interactive simulation

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Abstract

The bicycle model presented in this paper was originally developed for educational purposes at the University of Seville for teaching kinematics and dynamics of machines at the master level. The model is shown in Figure 1. The system is kinematically described by the following set of coordinates that can be identified in the figure:

$$\mathbf{p} = \begin{bmatrix} x_C & y_C & \varphi & \theta & \psi & \beta & \gamma & \varepsilon & \xi \end{bmatrix}^T \tag{1}$$

where the angle ξ that is used to locate the contact point on the front wheel is a non-generalized coordinate.

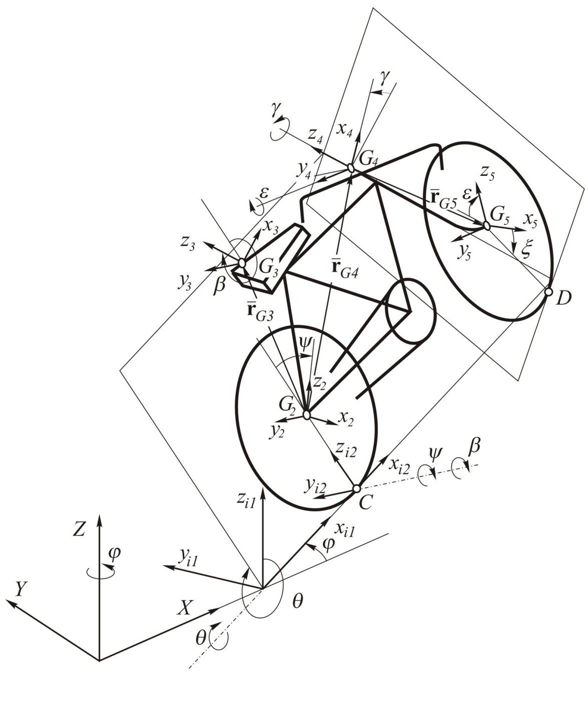

Figure 1. Biclycle model and coordinates description

The set of coordinates $\mathbf{p}$ is subjected to contact constraints $\mathbf{C}^{con}(\mathbf{p})$ (scleronomic), rolling without sliding constraints $\mathbf{C}^{rol}(\mathbf{p},\dot{\mathbf{p}})$ (non-holonomic) and mobility constraint $\mathbf{C}^{mov}(\mathbf{p},t)$

(rehonomic). This bicycle model can be used to explain the differential-algebraic (DAE) nature of the equations of motion that appear in multibody dynamics and different methods that can be used to deal with them. These equations that obtained in this investigation using symbolical computations take the form:

$$\begin{aligned} &\mathbf{M}(\mathbf{p})\ddot{\mathbf{p}} + \mathbf{D}^T \boldsymbol{\lambda} = \mathbf{Q}_v(\mathbf{p},\dot{\mathbf{p}}) + \mathbf{Q}_{grav}(\mathbf{p}) + \mathbf{Q}_{ext} \\ &\mathbf{C}(\mathbf{p},\dot{\mathbf{p}},t) = \mathbf{0} \end{aligned} \tag{2}$$

The equations of motion in DAE form given in Equation (2) are transformed to a minimal set of ordinary differential equations (ODE) using the generalized coordinate partition method [3]. The lean angle θ and the steer angle γ are selected as independent coordinates and the new equations take the form:

$$\mathbf{M}_i \ddot{\mathbf{p}}_i = \mathbf{Q}_i \tag{3}$$

Equations (3) are linearized about the vertical position and eigenvalue analysis is carried out for a range of forward velocities. The results are shown in Figure 2. This continuation diagram is compared to the one given in Reference [2] showing good agreement and it has been experimentally verified.

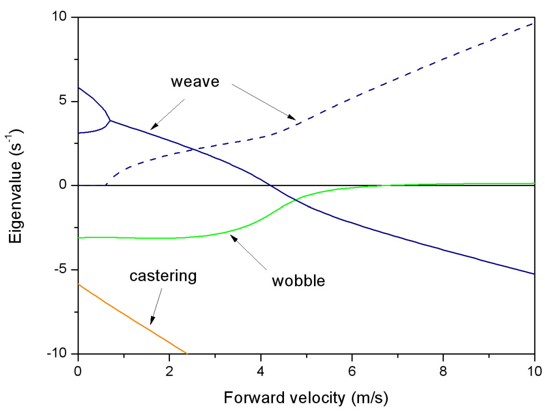

Figure 2. Continuation diagram of the bicycle stability

The symbolically obtained equations of motion of the bicycle are well suited for interactive real-time simulation of the bicycle riding. The simulator developed in this investigation uses steering wheels, joysticks and inertial sensors used in video games for virtual riding of the bicycle. The simulator also includes simple 3D visualization. Different strategies for virtual riding of the bicycle are tested. The inputs to the system may include the steering angle γ, a steering torque M_γ^x in the forward direction, a steering torque M_γ^z in the direction of the front fork axis of rotation and an upper body lean angle. The bicycle motion obtained with the simulator seems very realistic however path following control requires further development. The simulator is a nice application of control of underactuated systems.

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Build It Wrong, But Build It – A Bicycle Trek

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Abstract

The author initially became intrigued in the seeming mystery of bicycle stability in part as a result of the musings of Jones [1], and reported in *The Wall Street Journal* [2]. The author has spent forty years pondering numerous mysteries associated with the bicycle. A resulting activity came to being known as the University of Illinois Bicycle Research Project [3]. This paper provides an overview of the author's resulting four decades long bicycle trek. Using the concepts of empiricism along with open ended essay assignments to students, the author oversaw and directed the bicycle related research investigations of approximately 1,000 mechanical engineering undergraduate students. Moreover, the author performed some additional studies, often as sequels to the student investigations.

This paper endeavors to report on a selection of the most noteworthy contributions that resulted from the University of Illinois Bicycle Research Project. The paper's title makes reference to the concept of trial and error. While theory has incredible merits, complimentary information can be gained at a comparatively swift rate once the decision is made to tinker, build (and build crudely if necessary), and observe outcomes. The guideline to the students often became; "Build it wrong, but build it." A prior century of analytical thinking focused on the role of precession of the wheels of a bicycle as the crux of stability, and yet the bicycle industry simultaneously was going to considerable lengths to lighten bicycle wheels. Sales of bicycles with light weight wheels increased. Moreover, bicycles appeared to be vastly more forgiving and robust than suggested by extant theory.

Topics to be discussed include:

1. Precession cancelation and enhancement experiments
2. Front fork geometry and trail experiments
3. Rear-steered bicycle designs
4. Rocket bike push tests (with an asymmetrical rocket torque applied to the handlebars)
5. Steer torque and lean angle experiments on a bicycle ridden on marked circular paths
6. Musings on riding a bicycle on the moon, hence the predictions of bicycle dynamics in fractional gravity environments
7. Reduced scale model experiments of bicycles with gyroscope controlled steering
8. Designs to achieve a passive intuitive bicycle thereby eliminating the need for counter-steering
9. The passive SSTT (Stable Single Track Trailer) design challenge
10. The Anti-Wind Bicycle challenge

11. The non-existent bicycle block diagram dilemma

When all of the studies and experiments cited above are looked at as a whole, the results point to underlying truths as per the bicycle handling and dynamics. A bicycle remains upright by a combination or orchestration of mechanisms. The ease in riding a bike to maintain stability is made possible by a mixture of four dominant mechanisms; (1) rider skill in knowing to ease the bicycle's steering into the direction of fall, (2) the front steering configuration combined with a combined head angle and rake to create proper trail, (3) the role of precession that reinforces the turn of the front fork assembly into the direction of fall, and (iv) rider upper torso articulation. All four of these mechanisms, to varying degrees, work individually and in concert to make it easy to ride a bicycle. Moreover, extensive experiments demonstrate that none are necessary *per se* as rideable bikes demonstrate that each of the four mechanisms may be negated and yet the bicycle can still be ridden – and quite easily. As a consequence, two sayings have become associated with the ease of riding a bicycle – "Once you learn how to ride a bike, you never forget." "It's as easy as riding a bike."

In the author's extensive involvement in teaching thousands of children with disabilities to ride a bicycle, the instructions to the child boil down to three simple things – (i) pedal the bike, (ii) keep your head up and look forward, and (iii) smile. Of the three admonitions, the smile is perhaps the most important of all, for when a child doesn't smile this is associated with a high level of fear and anxiety. Fear and anxiety cause the body to be stiff as opposed to relaxed. Stiffness of the body brings with it a reduction in movement of the body joints, reflecting a reduction in the degrees of freedom. Conversely, a relaxed body tends to be more fluid, graceful and utilizes an increased number of body joints. As the body becomes more fluid, the energy required to perform a task is diminished.

The author concludes with an overview of the *Lose the Training Wheels* program now established in the United States [4, 5]. The goal of *Lose the Training Wheels* is to work with children with disabilities so as to allow the children to master the ability to ride a conventional two-wheeled bicycle.

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Modeling Mechanical Optimization in Competitive Cycling

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Abstract

A 3D model has been developed that combines bicycle mechanics, rider biomechanics and environmental factors into a single dynamic system. The aim of the model is to identify mechanical mechanisms that influence performance in a road cycling time trial with simulations representing actual cyclists competing in actual events.

The model is constructed using the Matlab toolbox SimMechanics to model physical entities and Simulink to model control structures. In SimMechanics, a 'machine' is built using blocks to represent rigid bodies linked by joints (including closed loops). The system is actuated by force or motion actuators applied to joints or bodies while sensors measure the resulting forces and motion. A range of constraint blocks allow limits to be placed on forces/motions and provide functions such as gears and rolling wheels. Rigid bodies and joints are linked with lines that essentially represent 2-way 'action-reaction' physical connections providing implicit inertial effects in a complete system. SimMechanics automatically derives the equations of motion leaving the developer free to concentrate on defining the mechanics of the system. Initial conditions are specified and a variable step ODE solver numerically integrates solutions that meet defined tolerances. The developed system operates in forward dynamics mode where forces applied to the model result in motion subject to constraints.

The main sub-systems comprising the model are shown in Table 1.

Table 1. Model sub-systems

Bicycle (Trek Madonne)	Rider (Typical 70 kg)	Environment (Course G10/42)
16 rigid bodies with dimension/mass/inertia	14 body segments (from literature)	Course track (from digital map)
Freedoms: x, y translation; roll, pitch, yaw rotation; steering, cranks/wheels rotation	Symmetrical two legged pedalling	Course gradient (from aerial laser mapping)
Holonomic + non-holonomic wheel constraints	Cyclic vertical/horizontal pedal force (phased 180°)	Bicycle/rider aerodynamics
Tyres (slip/camber forces, aligning/overturning moments)	Synchronised bicycle-rider roll	Environmental wind speed/direction
Geometry (COM, steer axis, trail, wheelbase)	Balance, counter-steering and path following	
Transmission		
Frame + wheel flex		

Initial model validation has been against the literature. Rider-less self-stability after a perturbation was simulated resulting in weave eigenvalues becoming negative at 4.2 m/s and capsize eigenvalues becoming slightly positive at 6.1 m/s. Both values compare well with the 2007 findings of Meijaard et al. [1] and the associated 2008 experimental validation by Kooijman et al [2]. Secondly, crank torque profile over 360° when the rider pedalled at 255 W was recorded from the model and found to correlate well (R^2=0.97) with experimental data reported in 1986 by Redfield and Hull [3]. Thirdly, the tyre model generated tyre cornering stiffness of 62N/degree (3 degrees slip, 4 degrees camber and 338 N vertical load) which closely matched the 60 N/degree (3 degrees slip, 10 degrees camber and 330 N vertical load) reported in 1972 by Roland and Lynch [4].

A field study analysed 20 experienced cyclists completing a time trial over an undulating 2.5 mile road course. The course was digitised and loaded into the model and the model parameterised with individual mass and aerodynamic characteristics. Wind strength and direction were recorded with an anemometer every 10 min during a trial and entered into the model. An error level of 1.4% (±1.5%) between actual and predicted individual time for the cyclist's 'best effort' over the course was obtained which is well below the 3.9% error for the road cycling model presented in 1995 by Olds et al. [6].

A subsequent investigation modelled and experimentally confirmed the theoretical advantage of adopting a variable rather than constant power strategy over an undulating road time trial course. The model generated an optimum power profile that minimised completion time subject to constraints on mean and peak power. Twenty time-trialists completed a total of 103 trials over the course utilising both strategies with the variable power requirement at ~80 m intervals dictated through an earpiece driven by a PDA/GPS attached to the arm. The model predicted a 4% time saving for the variable power strategy whereas a mean 2.9% actual saving was obtained in the field trials. The difference was partly attributable to unmeasured traffic drafting effects.

Further studies with the validated model are to investigate the mechanical performance advantages of bike/rider weight, saddle position, crank length, tyre characteristics and the contribution of muscular/non-muscular forces to pedalling.

The model is currently an incomplete representation of road cycling which is being addressed by the following enhancements:

- Asymmetrical pedalling
- Steering by rider arms
- Gears
- Vertical translation
- Magic Formula tyre model
- Pedalling by leg joint torques

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Design Sensitivity Analysis of Bicycle Maneuverability and Experimental Validation

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Abstract

The maneuverability of bicycle is dependent on various design parameters such as wheel base, head angle, trail, CG location, radius of wheel, etc., where some parameters are related to kinematic configuration of bicycle, while others are related to kinetic properties. This paper investigates the effect of design parameters of bicycle to the maneuverability of bicycle through sensitivity analysis and experimental validation.

Dynamic equations of motion for a bicycle are derived based on the bicycle model shown in Fig. 1, which has both kinematic and kinetic design parameters. By directly differentiating the equations of motion with respect to design parameters, sensitivity equations for bicycle dynamics are derived. The sensitivity equations can calculate sensitivity of any form of performance index expressed in terms of state variables.

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{C}\dot{\mathbf{q}} + \mathbf{K}\mathbf{q} = \mathbf{u} \tag{1}$$

where **M ,C, K** represent generalized inertia, damping, and stiffness matrix, respectively, and **q** and **u** represent state variable and external forces, respectively.

In order to evaluate driver's subjective feeling on maneuverability of the bicycle, two specific tests, steady state circular test and slalom test, have been defined. In the tests a test bicycle with changeable configuration, as shown in Fig. 2, has been used to research the effect of design change on the maneuverability. The ratings by drivers are measured using a questionnaire which is intended to measure maneuverability of bicycle with different configuration of bicycle.

During tests, state of the bicycle such as speed, roll angle, steer angle, lateral acceleration, lean angle of driver etc. are measured. Their correlation to subjective rating of maneuverability has been tried using data processing techniques such as averaging, RMS, and frequency analysis. By trial and error, the performance index that has good correlation with the subjective rating of maneuverability has been defined.

By performing sensitivity analysis on the performance measure, the effect of design parameters on the maneuverability of bicycle has been found. The results can be applied to optimal design of bicycle in terms of maximizing maneuverability of bicycle.

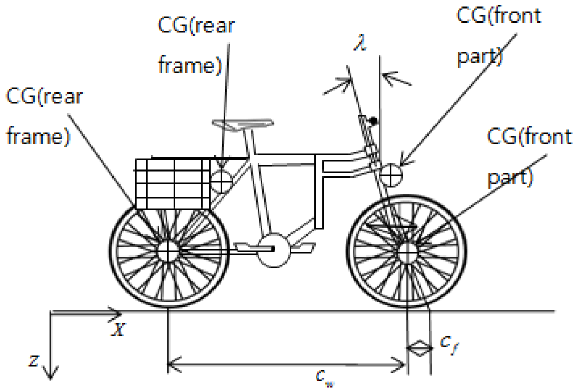

Figure 1. Dynamic model for bicycle with three major design parameters, wheelbase, trail, and head angle

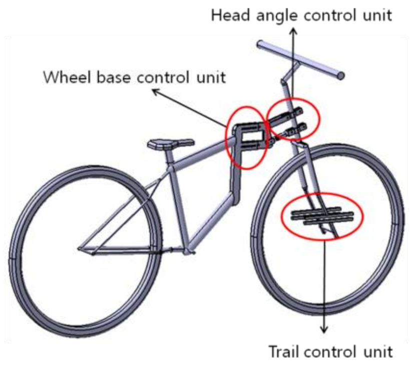

Figure 2. Experimental bicycle with changeable configuration

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Study on Characteristics of Motorcycle Behavior during Braking

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Abstract

For two-wheeled vehicles behavior, it is necessary to describe the roll motion because a main centripetal force at turning is camber-thrust. Therefore, the lowest degree of freedom for two wheeled vehicle compared with the four-wheeled vehicle rises. Furthermore, as the two wheeled vehicle has unstable or less-stable characteristics, a lot of researches for two wheeled vehicle have been poured into three typical modes, capsize, weave, and wobble. However, these modes except the capsize have usually little effect to ordinary behavior of recent motorcycle under the ordinary running speed. Therefore, when we evaluate the motorcycle, sometimes, characteristics of steady state turning and reaction force of the steer bar are used for motorcycle evaluation. As the first stage, characteristics of steady state turning of two wheeled vehicle on constant radius are analyzed, and the stability factor and side slip coefficient for two wheeled vehicle are proposed as follows;

$$\frac{\delta}{\delta_0} = 1 + K_\delta v^2$$

$$\frac{\beta}{\beta_0} = 1 + K_\beta v^2$$

Here δ : Steer angle δ_0 : Geometrical steer :angle
β : Side slip angle at C.G. β_0 : Geometrical side slip angle
K_δ : Stability factor K_β : Side slip coefficient

Next, using the two coefficients, the characteristics of two wheeled vehicle during braking are analyzed using quasi steady state method. In this analysis, cornering stiffness and camber stiffness are the function of each tire load as follows;

$$F_y = K_c\phi - K_s\alpha$$

Here K_s : cornering stiffness K_c :camber stiffness
α : slip angle ϕ : camber angle

These two coefficients, stability factor and side slip coefficient are calculated as a function of longitudinal acceleration shown in fig.1. The results don't include the effect of braking forces directly. However, motorcycle has two independent or combined braking systems between front and rear, and rider can control the front and rear brake individually. For such braking action, there is a possibility that the characteristics of motorcycle changes greatly. In this analysis, it is important to use a tire model which describes a relationship between longitudinal force and lateral force on each tire. Therefore, a simple tire model, which describes the relationship, is proposed. In this model, liner tire characteristics and non-liner friction coefficient are use, and an example result of this model is shown in fig.2.

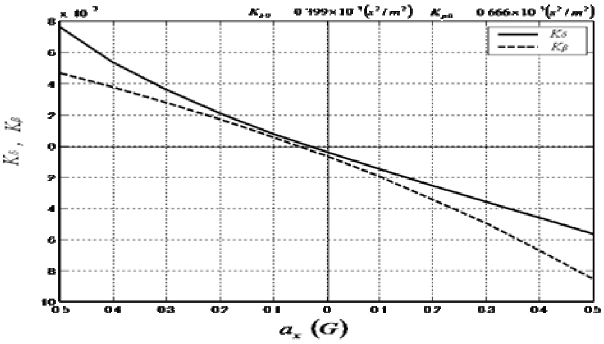

Fig.1 Coordinate System

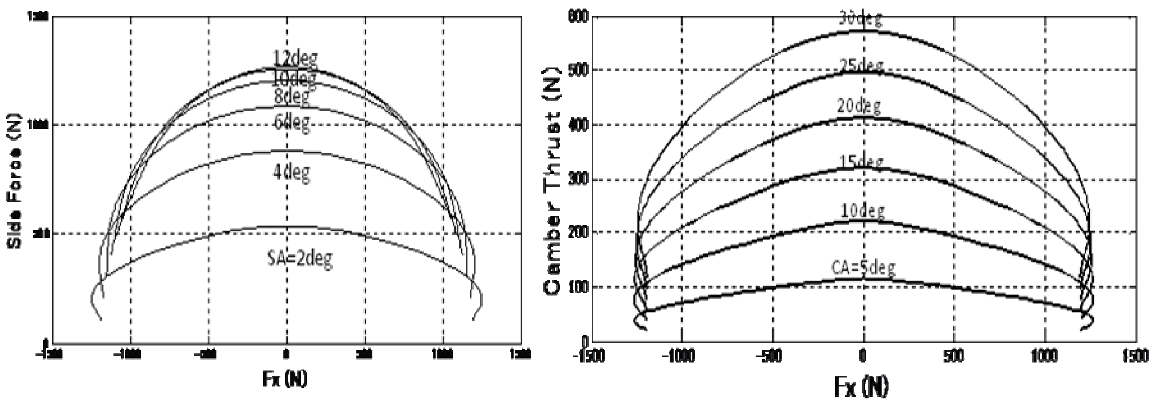

Fig.2 The calculation results of tire forces using the model

Using the tire model, the characteristics of motorcycle behavior during breaking are analyzed, especially described the effect of front and rear braking. The results are shown in fig.3. In this figure, dotted line shows non-braking and solid line shows 0.6G deceleration using front brake. In this condition, the effect of front braking is appeared just on side slip characteristics. As the characteristics of the two-wheeled vehicle at braking change depending on distribution of braking between front and rear., a detailed examination is needed from the standpoint of safety.

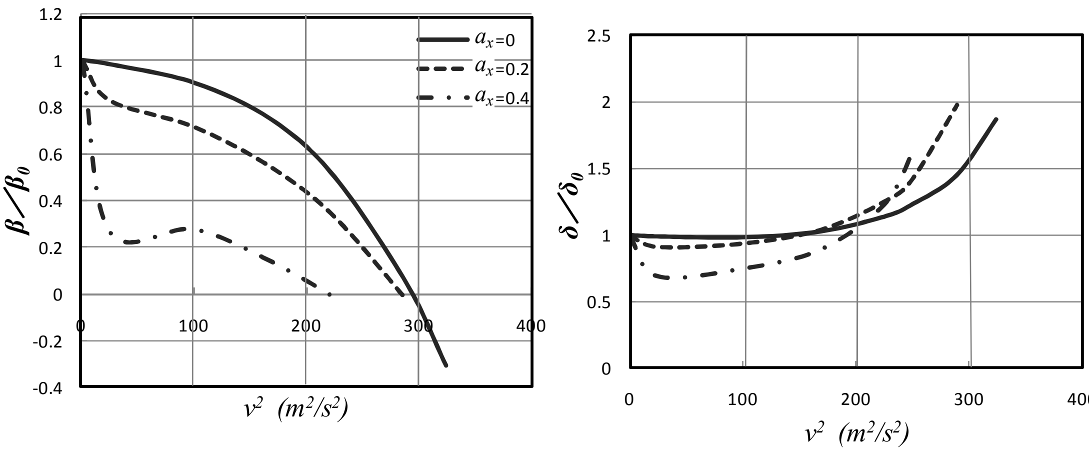

Fig. 3 The calculation results of quasi steady state turning during braking

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Influence of the Front Suspension on the Transient Dynamics of Motorcycles on Braking

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Abstract

The front suspension layout has crucial influence on the vertical and lateral dynamics of the motorcycle. The most widely used front suspension is the telescopic fork.

Alternatives offer some advantages, which can be verified through steady-state analysis. But those front suspensions have never found broad application in racing-oriented motorcycles. Instead, many riders claim to miss control on braking. To understand this criticism better from a scientific point of view, the focus of this work is set on the transient dynamics of motorcycles in the braking maneuver, which was inspired by the publication of Tony Foale [1].

For an analytical analysis, a simple and linear multibody-model is developed (Figure 1). It consists of three bodies. The model represents a simplified motorcycle with a rigid chassis body and two massless and ideally stiff wheels of the same diameter, an abstract front suspension and a direct connection between chassis and rear wheel. To keep the model as general as possible, the front suspension is represented by its instantanious center and a wheel related, linear spring (k) and damper (c) unit.

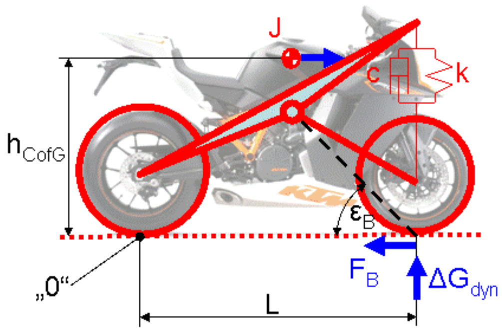

Figure 1. Front wheel load during sudden brake force variations.

For a first approach, the adhesion limits of a tyre are not considered.

Under this conditions, the transfer function between the braking force, which is the output created by the rider, and the dynamic front wheel load, can be formulated as

$$\frac{\Delta G_{dyn}}{F_B}(j\omega) = L(k + j\omega c)\frac{h_{CofG} + L\tan(\varepsilon_B)}{kL^2 + cL^2 j\omega - J_0\omega^2} - \tan(\varepsilon_B)$$

with J_0: Motorcycle inertia around point “0”.

Out of this equation, the system answer to a sudden brake force variation can be calculated by the inverse laplace transformation through correspondence tables. The results for typical motorcycle data are illustrated in Figure 2.

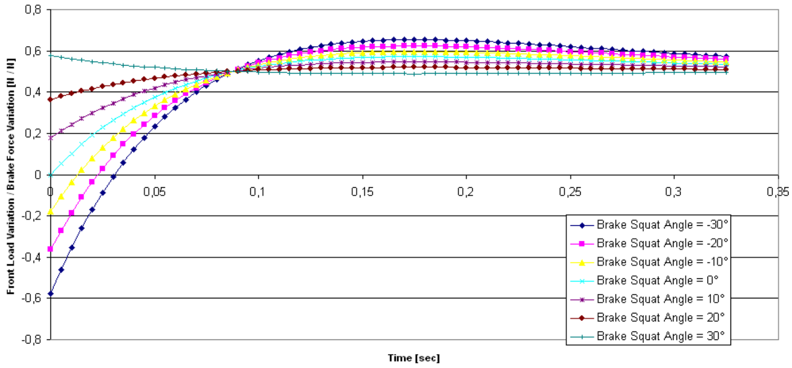

Figure 2. Front wheel load during sudden brake force variations.

These results suggest that a front suspension with a highly negative squat angle, like the telescopic fork, initially unloads the front, before loading it up to the steady-state dynamic load. Vice versa, a highly positive squat angle initially creates a very high wheel load on a quick brake force increase, which is later fading down to the stationary value. Opposite effects can be expected for a brake force decrease.

It can further be considered that there is a relation between the current wheel load, the thereby transmittable braking force and the tyre slip, which is usually described through the tyre model. This means generally, that at a constant brake force, the adhesion coefficient and thereby the slip is reduced by higher wheel loads.

The conclusion ,is, that a highly negative squat angle offers reduced grip on initial braking, as the dynamic wheel load is applied with a delay. Vice versa, on instant brake release or grip loss the dynamic wheel load increases due to the squat kinematics. This can help to safe a possible front wheel slide.

For a more realistic analysis, multibody simulations with VI motorcycle have been used. They are performed for three different front suspensions, which have different brake squat angles.

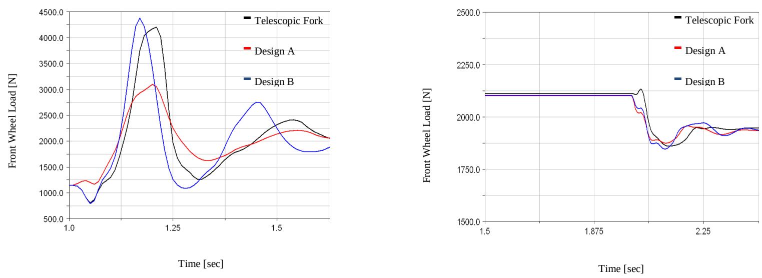

Figure 3. Wheel load due to sudden brake force increase (left) and decrease (right) for different front suspensions.

Indeed, the results show the same tendencies expected from the simple model. Additionally, they show the relation to other effects, like the rear suspension movement and the influence of a complex tyre model, which were neglected in the simple model.

As a conclusion of this study, on the one hand additional objective weight was given to the subjective criticism of many alternative front suspensions, often mentioning the missing feeling for the front on braking. The main goal, to deepen the understanding for this criticism from a scientific point of view, was reached. On the other hand, out of this aspect it will be possible to develop evaluation criteria for new generations of front suspensions, which might find more sympathies from the riders.

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Motorcycles Dynamic Stability Monitoring During Standard Riding Conditions

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Abstract

The target of this work is to improve driver safety related to dynamic instability, often the cause of accidents involving powered two-wheeled vehicles as explained in the UNECE Transport Review of Road Safety [1].The paper explains the feasibility study of an advanced driver assistance system (ADAS) with regards to the real-time instability conditions identification of a generic motorcycle during typical use. As well known, these types of systems have been typically applied and relied on in the automotive sector, however have not yet broken through for two-wheeled vehicles. Recently there has been an increase of interest and awareness for the application of ADAS systems to motorcycles as can be deducted by the SAFERIDER project [2].The first step of the research consists in an investigation on the methodologies that can lead to a positive outcome in identifying the approaching instability of a vibration mode of the motorcycle, and choosing the most efficient algorithm considering the final application on a diagnostic real-time system. An initial study of the vibration modes of a motorcycle led to the isolation of the weave mode and the wobble mode as the most likely to occur during motorcycle use [3]. Normally these two vibration modes have isolated frequencies [Figure 1]; while the weave mode is typically a low frequency oscillating mode (1-4Hz) which consists of the front assembly and rear assembly oscillating with an opposite phase one to the other, the wobble mode is a oscillation of the front assembly around the steering axis with little movement of the rear assembly and occurs at a higher frequency (8-11Hz).

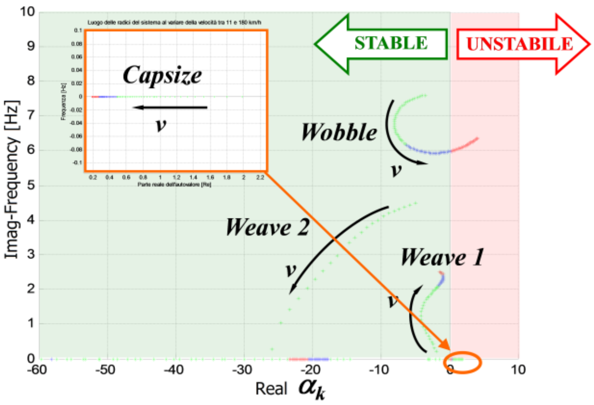

Figure 1. Root locus plot in straight running; speed varies from 3 to 50 m/s.

To determine when the weave mode and the wobble mode are moving towards a possibly critical situation, it is possible to estimate the respective damping coefficient of the considered modes. If the damping coefficient tends to a null value a dynamic instability is occurring. Thus the damping coefficient has been considered as the instability index.In the paper various methods will be proposed and discussed as possible candidates to estimate the frequency and damping coefficients of the two modes. The methods considered are based both on a one DOF system (fitting of a one degree of freedom frequency response function, peak picking method, phase differentiation method), acceptable because the frequencies of the vibration modes are distant one from the other, and a multi DOF system (Prony's method). The methods have been evaluated from both a numerical result and time complexity point of view, due to the fact that the chosen methodology must be sufficiently accurate and be able to run in real-time conditions.
In Figure 2 a flow chart containing the logical procedure with which a possible dynamic instability is identified and evaluated is shown. As can be seen, an averaged autocorrelation function of the extracted time signal is calculated in order to obtain a decay function and eliminate noise.

This function is passed into the frequency domain and the peak picking method is applied. Two damping coefficients are identified, relative to the weave and wobble modes. Should one of these damping coefficients be below a critical value, Prony's method is launched. Prony's method identifies critical frequencies and damping factors if present, and returns a visible warning message based on the results.

Figure 2. Flow chart explaining the logical procedure with which the time signal is analysed.

The proposed method must be successfully applicable to several motorbike typologies. Thus, to prove the efficiency of the methodology, in the first step of the research, several experimental tests have been carried out on different motorcycles. The developed intelligence has been adapted to offline post-process signals acquired by sensors while on road running tests. Figure 3 box A shows the acquired time history from sensor mounted on the steering assembly and the vehicle speed. In part B there is the spectrogram of the time history. Box C illustrates frequencies identified by Prony's Method, while in box D damping factors are shown. At time 53 seconds there is wobble oscillation. As can be seen the spectrogram gives a pick of amplitude (in red), while Prony's method identifies frequency and damping coefficient. By the analysis of achieved results, the same experiences allowed to define both the most sensitive sensors for the identification of vibration modes and the outer prefixed threshold.
Next actions are the integration of the intelligence on a real time on board system able to communicate with driver. Then, it will be possible to preview an active control device [4] able to reduce the occurred danger situation.

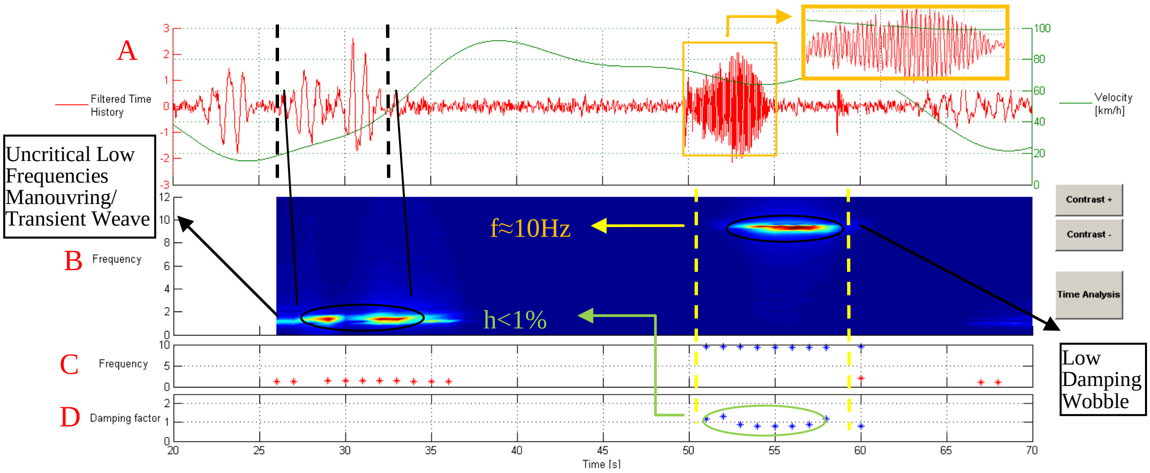

Figure 3. From top to bottom: (A) time history acquired by sensor and vehicle speed, (B) spectrogram, (C) identified frequencies, (D) identified damping coefficient.

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Application of the rigid ring model for simulating the dynamics of motorcycle tyres on uneven roads

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Abstract

Active control systems find their way into motorcycle technology for support of the rider, often aimed to enhance the safety without reducing driving pleasure. In order to develop these new systems in the virtual environment an extended description of the tyre behaviour is required. More specific, the higher order tyre dynamics needs to be modelled for the full range of driving conditions. This includes large lean angles, combinations of lateral and longitudinal slip, and various road unevenness. This publication describes the adaptations of a passenger car rigid ring model for uneven road simulations (i.e. TNO's MF-Swift) for motorcycle tyres in order to simulate cornering on uneven roads. The main extensions are related to the quasi-static tyre behaviour for rolling over obstacles. Further, the effects on the high frequency response are discussed.

Background

Due to the different construction and the large inclination angles that may occur, motorcycle tyres behave different from passenger car tyres. For this reason, the modelling techniques developed for passenger car tyres may not be directly applied. For instance de Vries and Pacejka [1] had adapted the well-known Magic Formula model for motorcycle tyres, which formed the basis for the widely used MF-MCTyre model of TNO [2]. Later Besselink et. al [3] adapted the original Magic Formula model for passenger car tyres in a different way to handle large inclination angles for both passenger car and motorcycle tyres resulting in a single Magic Formula for large camber angles, without making sacrifices with respect to accuracy for either normal passenger car or motorcycle tyres.

To be able to simulate the tyre dynamic behaviour at higher frequencies than is possible with a 'standard' Magic Formula tyre model, TNO and Delft University of Technology developed the MF-Swift [4] tyre model for passenger cars. The MF-Swift model is a rigid ring tyre model that consists of the following four components:

- Magic Formula
- Contact patch slip model
- Rigid ring
- Obstacle enveloping model

New developments

To investigate the applicability of the MF-Swift concept for motorcycle tyres a joint research project between TNO, Honda and Eindhoven University of Technology has been carried out. This research has lead to adjustments of the basic MF-Swift model so that it can be applied for motorcycle tyres. The main extensions concern the tyre-road contact under larger camber angles.

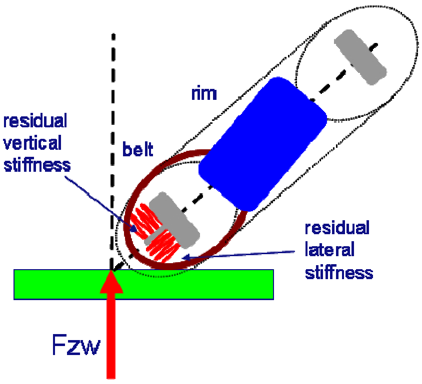

Figure 1. Schematic representation of the adapted MF-Swift model for motorcycle tyres.

Results

In this paper the adapted MF-Swift model for motorcycle tyres (cf. Fig. 1) is described and comparisons with existing models (MF-Swift and MF-MCTyre) are made to clarify the added value of the new model. Further, parameter identification and experimental validation of the model is discussed. Finally, it is concluded that the new model shows a significantly improved response for uneven road simulations for a motorcycle tyre due to modelling of the belt dynamics and the extended contact description. By comparison to MF-MCTyre and MF-Swift the improvements are quantified. As an example some results of a high speed cleat simulation are shown for longitudinal and vertical forces in Fig. 2.

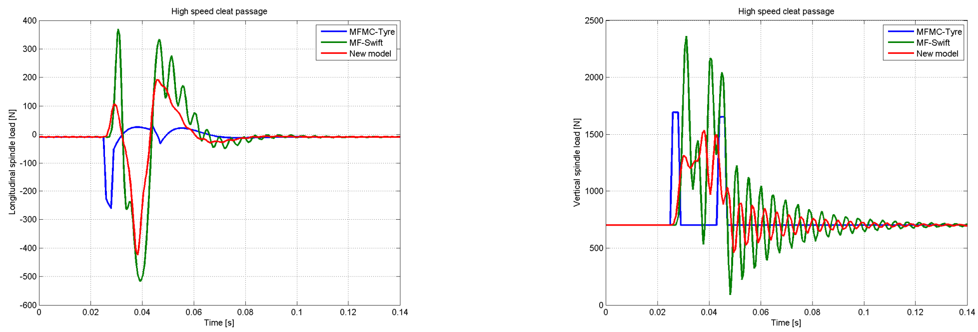

Figure 2. High speed cleat test simulation results for different models.

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Measuring Dynamic Properties of Bicycle Tires

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Abstract

Dynamic tire properties, specifically the forces and moments generated under different circumstances, have been found to be important to motorcycle dynamics.[1][2] A similar situation may be expected to exist for bicycles, but limited bicycle tire data and a lack of the tools necessary to measure it may contribute to its absence in bicycle dynamics analyses.[3][4] This paper describes tools developed to measure these bicycle tire properties and presents some of the findings.

Cornering stiffness, also known as sideslip and lateral slip stiffness, of either the front or rear tires, has been found to influence both the weave and wobble modes of motorcycles. Measuring this property requires holding the tire at a fixed orientation, camber and steer angles, with respect to the pavement and its direction of travel, and then measuring the lateral force generated as the tire rolls forward. Large, sophisticated, and expensive devices exist for measuring this characteristic of automobile tires. One device is known to exist for motorcycle tires, and it has been used at least once on bicycle tires, but the minimum load it can apply is approximately 200 pounds, nearly double the actual load carried by most bicycle tires.[5]

This paper presents a device assembled for less than US$1000 that measures bicycle tire cornering stiffness. It takes advantage of any sufficiently long, level, rigid, and smooth stretch of floor adjacent to a plumb, straight, rigid, and smooth wall to provide the test track. Several purpose-built tracks are also described. A flat and straight track avoids issues created by either vertical or horizontal curvature.

Figure 1. The test device and its track.

Although steer and camber angles must be measured separately, they can be precisely and finely set with rigid turnbuckles in any combination. The orientation of the tire is also enforced and wheel flex minimized by two sets of guide wheels that run on the braking surface of the rim. One is at the bottom of the wheel, near the contact patch, to prevent flexing due to the lateral force generated at the contract patch. The second is at front of the wheel to prevent rotation about the steering axis due to torques generated at the contact patch.

Nearly any sufficiently accurate force measuring system may be used to detect the generated lateral force, and this implementation uses a system from Pasco intended for classroom experiments.[6] The Pasco sensors themselves are rated for only ±50 Newtons, far less than the maximum expected lateral force from the tire, and so a simple lever mechanism, similar to the one Pasco uses on their stress-strain apparatus, is employed to scale down by five-to-one the lateral force generated by the tire, well below the 50-Newton maximum. Three force sensors are used: two to measure lateral force, and one to measure torque.

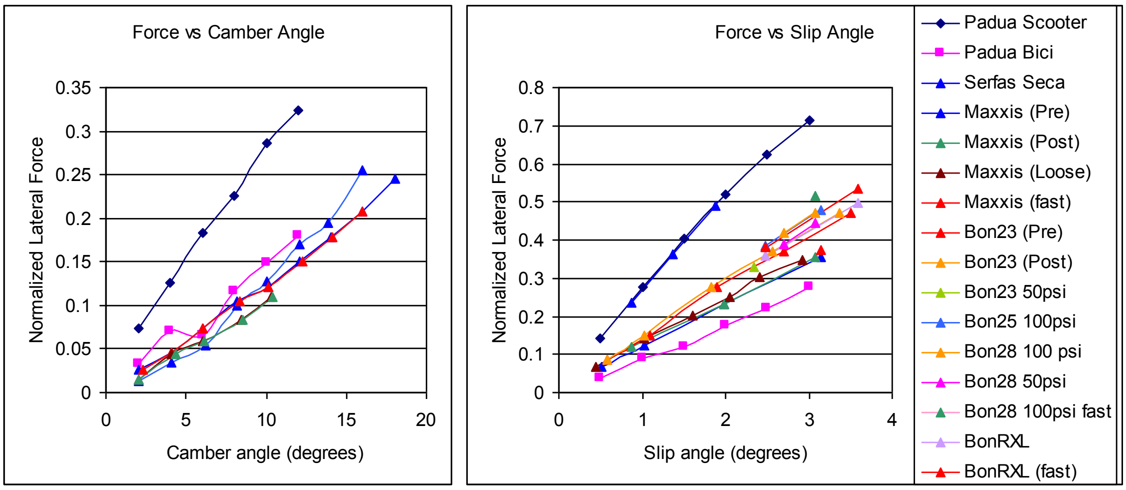

Figure 2. Test data shown along with data reported by Cossalter for a scooter and a bicycle tire.

The bicycle tires tested generate a larger lateral force for a given slip angle and a smaller lateral force for a given camber angle than the bicycle tire tested by Cossalter.

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Author index

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