MODEL AND DESIGN OF A RACE CAR PROTOTYPE

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Abstract. *Race cars and competition prototypes, are high performance vehicles which demands a thorough understanding of their dynamics in order to be adequately designed. Among the critical components the front and rear suspensions play a critical role in driveability and overall performance. This work presents a dynamical model of the vehicle, body and suspension mechanisms, based on a Formula SAE prototype, in order to investigate its dynamical behavior in curves. The tyre modeling takes into consideration their rigidity and friction coefficients, allowing the study of the load distribution. The drivers characteristics are implemented in the form of a simplified control system. Based on the proposed model parameter variations are done, in order to identify suitable values for the suspension geometry but also to provide a better insight in the design process as well. With the built prototype measurements are taken and compared to the simulations in order to verify the model.*

Keywords: *vehicle dynamics, measurements, tyre model, control system*

1. INTRODUCTION

The Formula SAE competition, sponsored by the Society of Automotive Engineers, is presented as a contest between race cars prototypes, high performance vehicles which demands a thorough understanding of their dynamics in order to be adequately designed. Nevertheless it is actually a competition between different designs, made by teams from mechanical engineering students of different universities, where the engineering contents of the project counts the most. Among the power-train, brakes and body design the understanding of the front and rear suspensions, which play a critical role in driveability and overall performance, is a key point in the project.

This work presents a dynamical model of the behavior of a prototype vehicle, including body and suspension mechanisms, based on a Formula SAE car, in order to investigate its dynamics, specially in acceleration and braking as well as in curves. The tyres are modeled in a simplified way but taking into consideration their rigidity and friction coefficients. Since the work does not aim at the power-train behavior its torque or power characteristics and the shifting of gears will not be included in the model. The power-train is actually substituted by a fairy simple speed control based on a proportional controller.

Based on the proposed model parameter variations are done, in order to design the suspension geometry but also to provide a better insight in the whole design process as well. With the built prototype measurements are taken in order to verify the model and the work discusses the necessary instrumentation.

2. MODELING

In order to obtain valid information about the performance of the vehicle from simulated results the mathematical model of the vehicle used should reflect the main geometric and inertial characteristics of the real model as well as a correct description of the interactions with the ground. A dynamical model based on the Newton-Euler formulation is appropriate to deal with the formulation of the equations of motion and easily determine reaction forces needed to design components. Nevertheless the vehicle, considering its suspension geometry and components, is actually a multi-body system consisting of four double wishbone suspensions and respective wheels, a steering mechanism comprising of two links with a steering gear and the vehicle body itself, each modeled as a rigid body. Table 1 summarizes the bodies in the model:

Table 1. Relation of bodies in the vehicle multi-body system

Qty.	Component	Subsystem	Mass [kg]
4	Wheel	Suspension (2 front + 2 rear)	28
4	Wheel support	Suspension (2 front + 2 rear)	20
4	Suspension A (superior)	Suspension (2 front + 2 rear)	2.5
4	Suspension A (inferior)	Suspension (2 front + 2 rear)	3,4
4	Spring-damper link	Suspension (2 front + 2 rear)	2,8
4	Pushrod	Suspension (2 front + 2 rear)	2,4
1	Steering gear box	Steering	3,1
2	Steering link	Steering (front right and left)	1,2
1	Vehicle body	Structure	230

The whole model comprises thus 28 bodies with a total mass of approximately 295 kg, without driver.

The connections between the listed bodies are mostly modeled as rotational joints, allowing the angular motion around a properly oriented axis between the corresponding elements, and thus consisting of a possible degree of freedom of the system. Figures 1 and 2 shows the CAD models of the front and rear left double wishbone type suspensions.

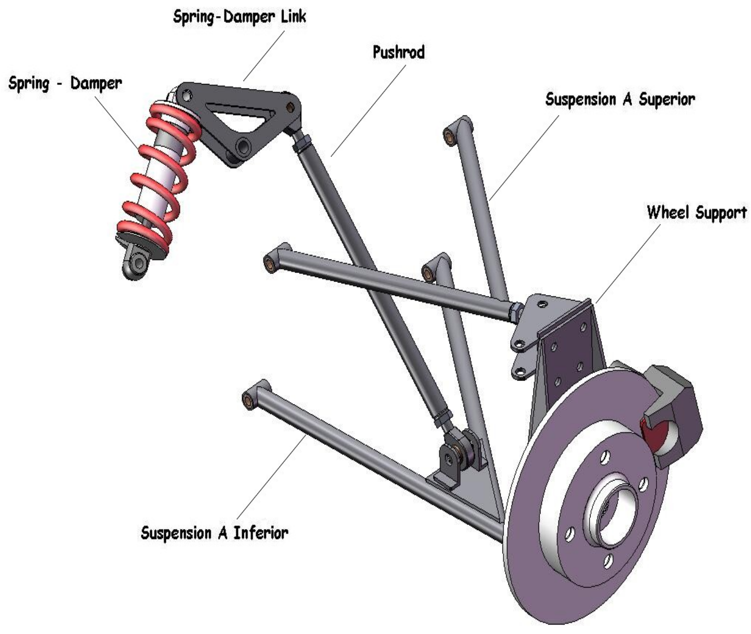


Figure 1 – CAD model of the front left suspension

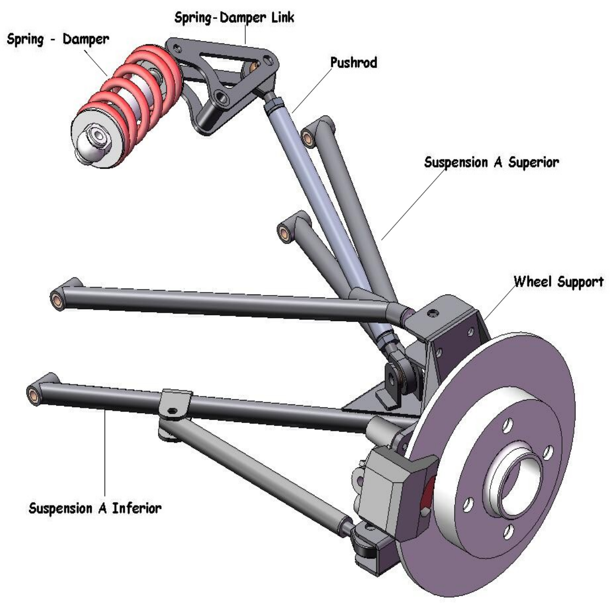


Figure 2 – CAD model of the rear left suspension

It can be seen that the suspension mechanisms build closed loops with the vehicle body, thus reducing the actual degrees of freedom of the whole system. Other closed loops are built in the steering mechanism where one translational joint accounts for the result of the position of the steering wheel, not modeled itself. The vehicle body, in principle, has the possibility of all rigid body motions in an inertial reference frame. These motions are restricted by the ground contact of the wheels. Figure 3 shows the evolution from model systems, from CAD representation to the dynamical model which represents the real vehicle.

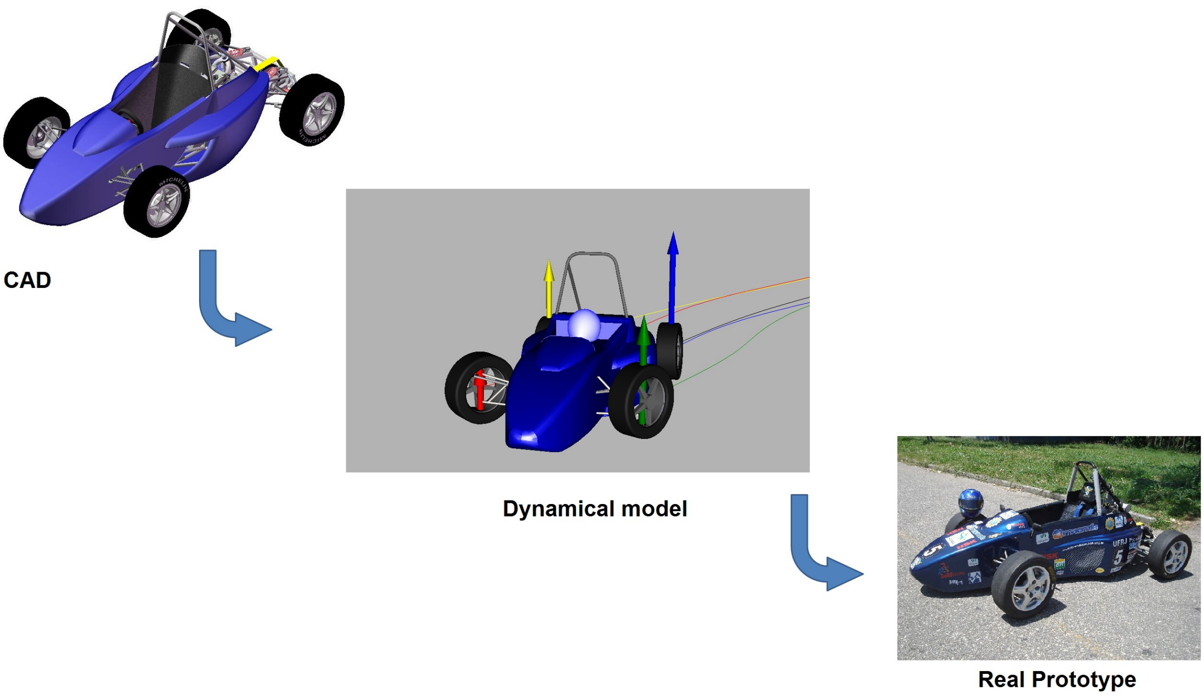


Figure 3 – Vehicle models – CAD / Dynamical model / Real Prototype

It can be easily seen that the task of deriving motion equations, with their interactions through the joints is not an easy task to be accomplished by hand. The geometric coordinate transformations needed and also the correct accounting of the inertia effects in a 3-dimensional motion will lead to long expressions prone to mistakes when derived by hand.

Once determined these equations of motion must be integrated in order to obtain the time histories of the joint coordinates, i.e. the dynamics of the system.

The use of a dedicated software to model and analyze multi-body systems is thus mandatory if one expects to obtain reliable results. In this work the task of modeling and the simulation itself is accomplished through the software Universal Mechanism (Pogorelov 1997), which allows the dynamic analysis of rigid body systems to be investigated. Each of the already mentioned components has an associated image, for displaying purposes, an associated inertia tensor and the assigned position of its mass center. These quantities define, for the sake of the simulation, each of the bodies. The joints between bodies and actions (forces and moments) upon the system are also modeled within the software interface. The resulting mathematical model is established as a set of equations of motion which are symbolically obtained by a standard Newton-Euler procedure and are also optimized to generate a DELPHI code. This code, together with a set of pre-configured libraries, is compiled obtaining a full simulation program with its own graphical interface and analysis capabilities.

The force system acting upon the vehicle is discussed in the next section, including the contact forces of the ground contact.

2.1. Forces, torques, tyre and ground contact models

Along with the modeling of the suspension mechanisms that allow the determination of the dynamic part of the equations of motion, of crucial importance is the correct determination of the force models involved in the interactions of the vehicle, specially with the ground. These forces can be divided in the following way:

- ✔ internal forces affecting the dynamical behavior, i.e. the forces on suspension springs and dampers;
- ✔ external forces of interactions, i.e. gravity and ground contact;
- ✔ external control forces, i.e. acceleration and breaking torques;

the driver characteristics is accounted for in form of the position of the steering mechanism. The corresponding driving forces arise from the tyre model.

As a first approximation, which is in turn quite accurate, the suspension elements are regarded as linear springs and dampers. The parameters were established from the actual elements in use. The geometry of the system, allowing large displacements of the suspension bars introduces non-linear effects on the suspension, even with the use of linear elements.

The weight of the vehicle is accounted for by simply indicating the direction of the gravity acceleration vector, allowing the calculation of the related forces.

The ground contact model of the tyre used in the present work is a simple one, considering the numerical contact condition between an imaginary circle and a specified surface, representing the ground. This surface may be described by a mathematical expression, allowing for the simulation of driving and riding conditions even on uneven terrain. The parameters of the contact are expressed by a joint stiffness, accounting for the stiffness not only of the tyre but for that of the ground in series. The present work considers the ground as being rigid, so that the contact stiffness is provided solely by the tyre. The model includes also a dissipation effect in form of a linear damping in parallel with the contact stiffness. The friction effects on the directions perpendicular to the contact normal are expressed by corresponding static and dynamic friction coefficients. The present model uses coefficients near 0.8 for the friction.

Since the tyres of a competition vehicle are rather wide the tyre model used considers actually the contact between three different circles with the ground. One circle is positioned on the center of the tyre and the other two on both extremities. This allows an approximated identification of the pressure distribution along the tyre width, and its effect on the vehicle dynamics.

With the possibility of changing all these parameters the dynamical model allows easily the simulation of several different vehicle setups, not only regarding its overall dynamical behavior but also transient ones during acceleration or cornering.

2.2. Speed control and driver characteristics

An important feature of the simulations is the possibility of a further comparison with actual measurements on a real vehicle. In order to establish a well controlled and well defined test procedure is essential that parameters are kept constant within certain limits during the test. A natural parameter to be controlled by the driver is the vehicle´s linear velocity. The simulation incorporates a very simple speed control in the form of a proportional control on the driving torque at the rear wheels. For the case of a break test not only a breaking torque must be applied at the four wheels but the balance of the breaking action between front and rear axis is an interesting parameter to be investigated. Following these ideas the torque T acting at the rear wheels is calculated through equation (1) below:

$$T = c_1 \times (\omega - \omega_d) + c_2 \times (1 - c_3) \times \mathrm{sgn}(\omega) \tag{1}$$

where the coefficients c1 relates to the proportional speed control, c2 is a breaking torque and c3 represents the proportion of the breaking action which is directed toward the front wheels. The sgn function accounts for the direction of the torque, reducing it to zero at still stand. The actual and desired angular velocities of the wheel are ω and ω_d respectively. The front wheels in turn only contribute to the breaking action and their torques are expressed by the following equation (2):

$$T = c_2 \times c_3 \times \mathrm{sgn}(\omega) \tag{2}$$

Other easily accomplished test is the slalom, where the steering wheel is turned to the right and left sequentially continuously. The simulations presented in this work, for the sake of easiness, do not include a driver model, accounting for the reaction times and the perception of the trajectory. The slalom is simulated by simply imposing an harmonic motion of the steering gear where the amplitude and frequency can be altered.

2.3. Simulation

The mathematical model is translated in a simulation software, automatically generated by the universal mechanism software package. This simulation module allows the determination of different initial conditions for the motion being simulated and also the variation of the parameters introduced during the data input process. The principal simulation results here presented concerns the cornering behavior and the influence of the breaking balance-bar.

The graphical interface of the simulation model is quite flexible and easily permits the determination of X-Y-curves relating simulated time but also different dynamical variables. Vector graphics allows the visualization of forces, accelerations, velocities and other vector quantities, dynamical or cinematical. The trajectories of specific points in different bodies can be traced, together with a motion animation, in an animation window with several user-defined properties. The dynamical data resulting from the simulations can be saved for a further processing or graphical comparisons.

The behavior of the vehicle is simulated for a curve with constant steering position. The vehicle begins from rest and increases its velocity in a smooth plain ground to 2,7 m/s, 6,3m/s and finally reaching 8,7m/s. Figure 4 exhibits the velocity profiles in two different simulations. A first case is simulated over smooth ground and then the same situation is explored over an uneven terrain with harmonical pattern of 5mm height. It can be seen that the instant of time where the vehicle changes its speed is slightly different between the calculations.

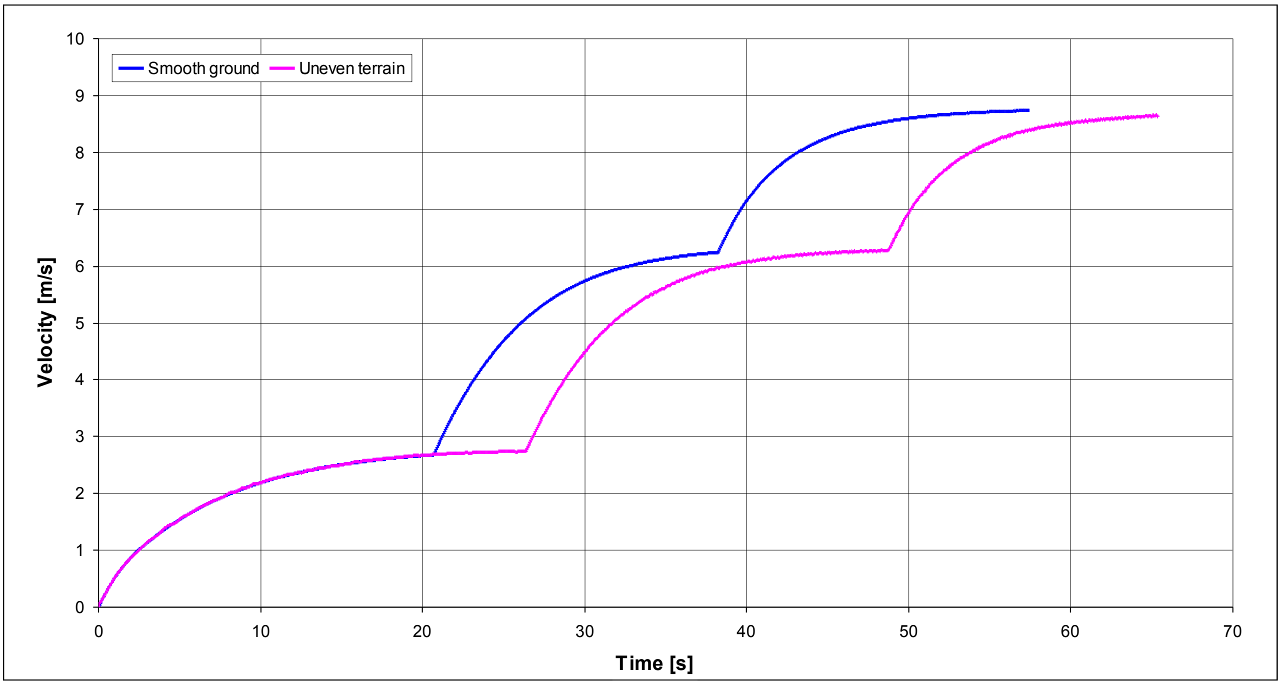


Figure 4 – Velocity profile in a curve motion

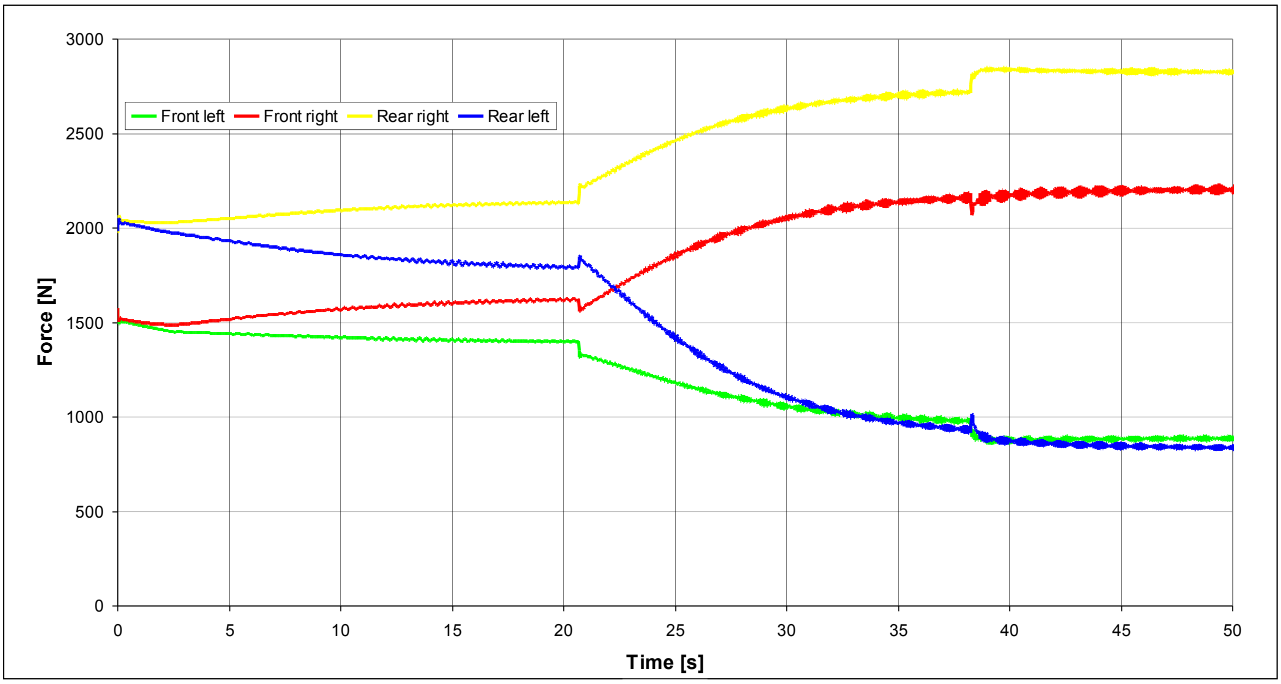


Figure 5 – Forces acting on the suspension springs for smooth ground

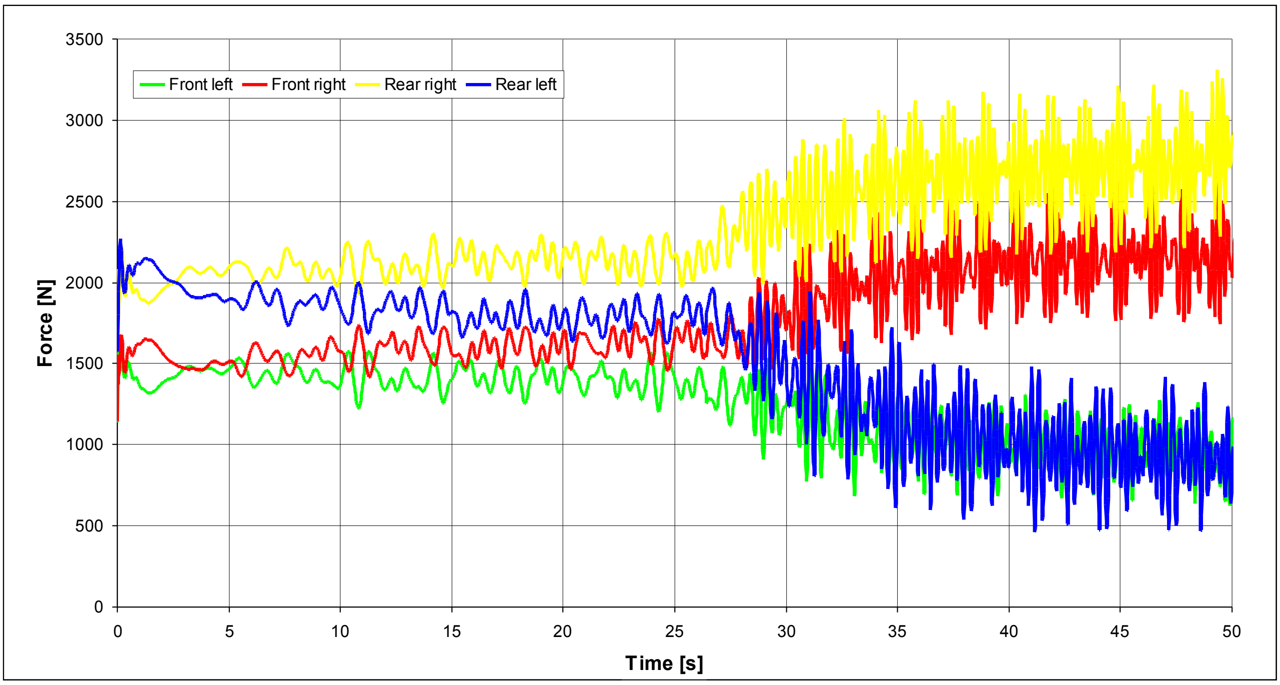


Figure 6 – Forces acting on the suspension springs for uneven terrain

Figures 5 and 6 show the forces acting on the suspensions during the curve. It can be seen the load transfer among the suspensions, specially during the acceleration phases. The effect of the uneven ground can be seen on the force fluctuations in Figure 6. The change in speed changing the excitation frequency of the force on the suspension influences the overall sensation on the vehicle due to their frequency responses.

Another simulation set explores the acceleration test for a 75m straight path followed by a constant breaking. Extreme choices of the break balance, only front breaking, only rear breaking and a 50% balance are shown. Figure 7 illustrates the velocity profiles of each simulation during the acceleration and breaking phases. It can be noted that the moment of the brake appliance is almost the same for the cases with 50% balance and only rear breaking. The beginning of the break phase in the last simulation was about 0.2s earlier.

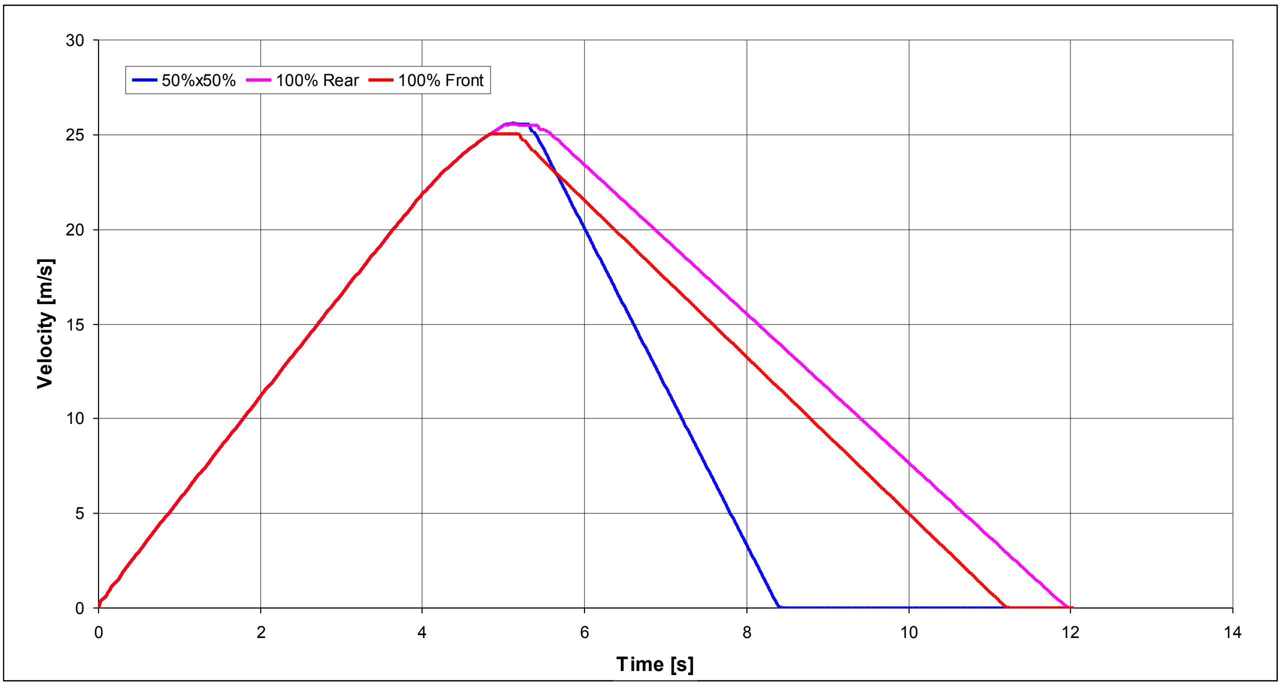


Figure 7 – Velocity profiles in acceleration

Nevertheless it is easily verified by the time needed to reach the still stand that the breaking efficiency greatly varies with the balance. With these simulations one is able to establish an optimal compromise for the actual vehicle.

In Figures 8 and 9 the load transfer between front and rear suspensions can be seen for the acceleration as well as for the breaking phases. The extreme condition of torque transfer can be recognized on the transient fluctuations in every case shown.

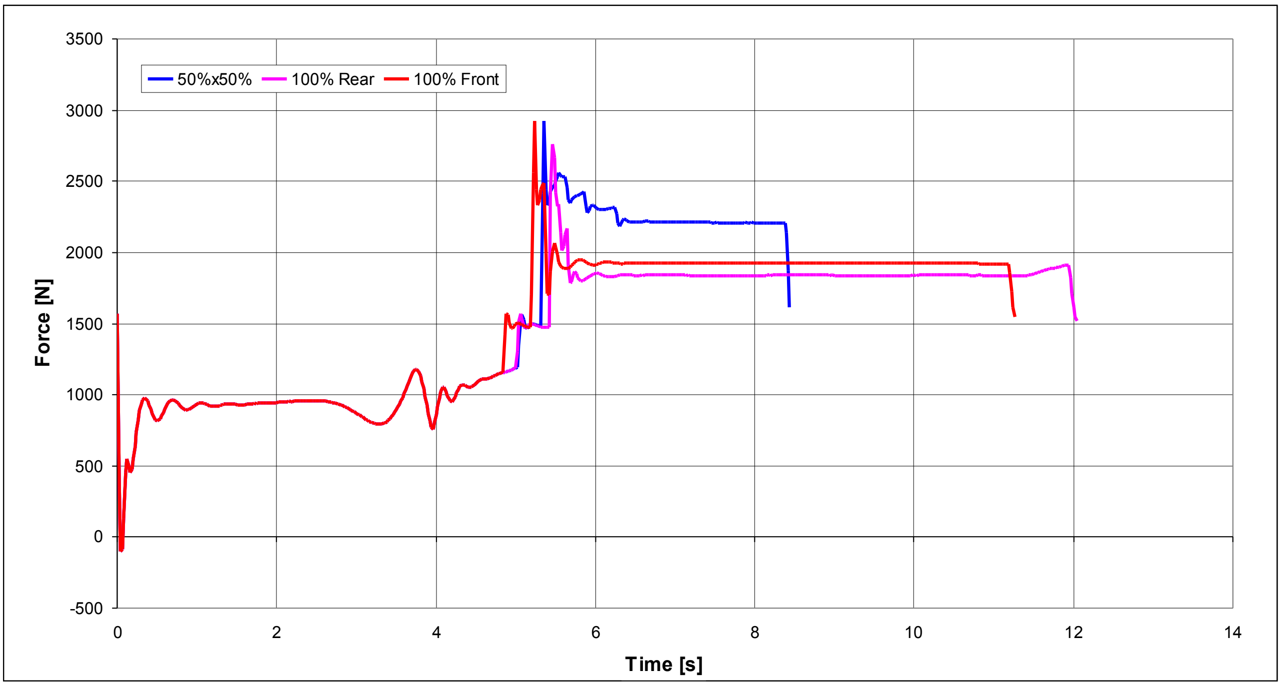


Figure 8 – Forces on the front left suspension spring

In both figures the opposite effect on the front and rear portinons of the vehicle can easily be recognized. Not only the best break balance can be investigated but also the influence of the suspension stiffness, even for the acceleration condition.

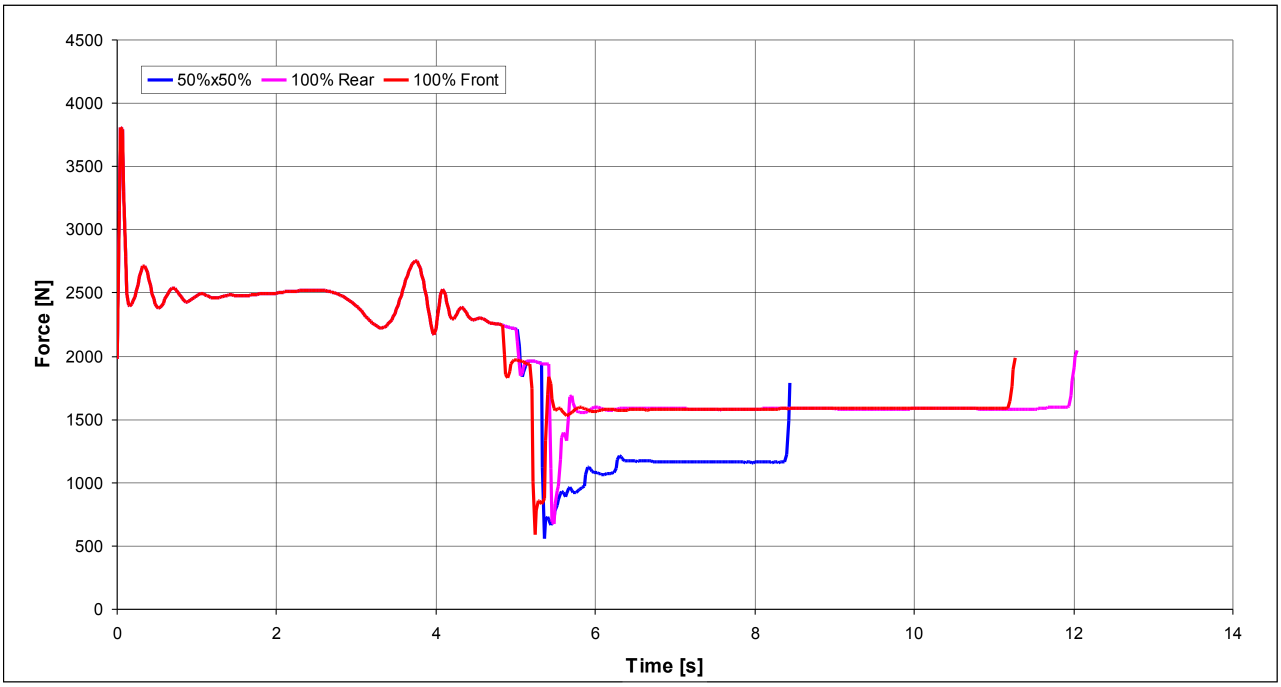


Figure 9 – Forces on the rear left suspension spring

3. INSTRUMENTATION

The mathematical and simulation models presented need to be verified against the actual behavior of the real prototype shown below in Figure 10.

Figure 10 – FSAE prototype modeled used in experimental investigations

With the recent introduction of low-cost wireless data transmission hardware a wide variety of options are opened for the experimental investigation of a running vehicle. The designed FASE prototype includes the use of a low-cost inertial platform (Silva et alli 1998), comprising of 3 accelerometers and 3 angular velocity sensors, in order to obtain dynamical data of the overall behavior of the vehicle. The platform is shown in Figure 11.

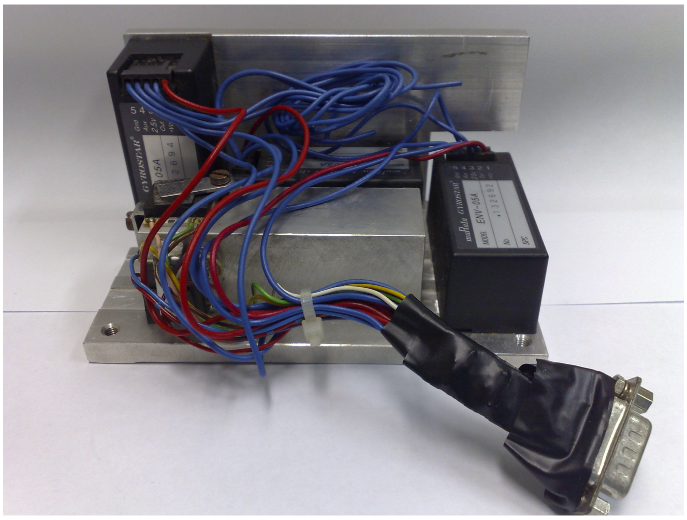


Figure 11 – Inertial platform showing accelerometers and angular velocity sensors

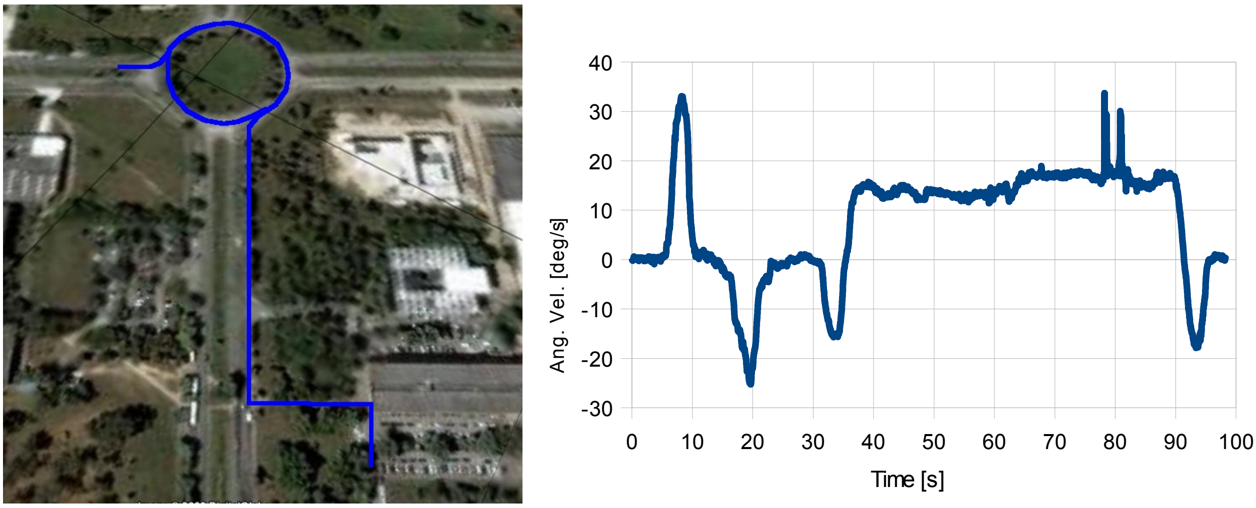
Not only dynamical variables can be acquired but also power train variables such as oil, water and intake temperatures, ignition times, engine speed, throttle position etc... The breaking system is studied on the basis of the accelerations measured by the inertial platform and the dynamical pressure on the breaking lines, along with the instrumentation of the steering wheel angle.

The data acquisition is made by the use of the National Instruments LabVIEW® software and a 4-channel wireless transmission hardware NI-WLS9234, shown in Figure 12, capable of dynamic data acquisition up to 52 kHz sample frequency and 24 bit A/D-conversion.

Figure 12 – Data acquisition hardware with standard wireless network transmission protocol

Data transmission is achieved through standard IEEE 802.11b/g wireless and Ethernet communications interfaces thus allowing the use of standard personal computers and peripherals and standard operating systems and acquisition software.

Figure 13a shows a test trajectory in the Campus of the University used to verify the data acquisition in the prototype. The measured angular velocity around the vertical axis during the test circuit can be seen on Figure 13b. It can be noted that the sensor adequately represent the changes in direction of the vehicle in the path.



a) Trajectory (source GoogleEarth) b) Measured angular velocity around vertical axis

Figure 13 – Dynamical measurements

4. CONCLUSIONS

The paper presents the model of a competition vehicle including non-linear geometrical effects on the suspensions, a simplified but efficient tyre model and speed control but still lacking a driver´s model.

The possibility of changing parameters allows the verification of different race setups even for complex transient behavior. From the basis of these evaluations through simulations the prototype can be adequately designed, allowing the engineer to better understand its whole dynamical characteristics.

Simulation results are shown, in order to exhibit the model capabilities, with the behavior on a fixed radius turn, and on the breaking balance. These examples were actually used during the vehicle design process of the prototype.

The instrumentation of the vehicle, with the aid of wireless data transmission, allows the experimental verification of the simulated results presenting a very interesting design process. The model is developed towards the simulated execution of driving tests to be compared with real ones.

- ✔ Further developments include, among other questions:
- ✔ driver model to study drivers characteristics influences;
- ✔ tyre model with deeper phenomenological and operational parameters like rubber type, air pressure, etc...;
- ✔ thorough experimental investigation for validation and parameters database construction;
- ✔ real time computer based driver model to allow a robotic driving of the prototype.

5. REFERENCES

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