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DYNAMIC ANALYSIS OF MOTORCYCLE BEHAVIOUR ON THE ROAD WITH STEERING PLATE STRUCTURAL OPTIMISATION

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Abstract

In this paper Visual Nastran software for the multibody simulation is used with the purpose of determining the solicitations in the vehicle body executing some characteristic manoeuvres on the road. The interaction between the tyre and the ground is determined using a linear analytical model that, with regard to Pacejka formulas, is more simple and able to give suitable results for a large field of variability. At the purpose a control program, written in Visual Basic, is elaborated to communicate with Visual Nastran by the Technology OLE Automation, that is a fundamental characteristic of the programming at objects for Visual Basic and for the programs operating in Microsoft Windows field.
At the end the use of ANSYS Design Optimisation modulus permitted the structural optimisation of the steering plate, reducing considerably its weight with regard to original one.

Key words: Motorcycle Dynamics; structural optimisation; multibody technique; OLE technology

1. Introduction

The analysis of the behaviour on the road of the vehicles by dynamic simulation software had a remarkable development in the last years, attracting the interest of many industries and becoming argument of study and research at world level. The dynamic behaviour of the motorcycle is very complex and is very influenced by the variation of the characteristic geometrical greatness and by the pilot guide style; the characteristics commonly used to evaluate the motorcycle behaviour are the handling and the directional stability.
Today the classic road test is the more spread method to analyse the motorcycle dynamics, obtaining the trim and evaluate the influence of the cinematic characteristics on the stability and manoeuvring of the vehicle. This test results are always influenced by the particular guide style and by personal feeling of every pilot; besides this test require a great time quantity and the collecting of an enormous data quantity, to elaborate successively. The purpose of this work is the development of a method and of an analysis instrument, that is fast, objective, flexible, able to facilitate the technicians work in the design initial phases, developing and accelerating the vehicle set. The analyses were effected by the software MSC Visual Nastran for the dynamic multibody simulation; it is provided with an immediate graphical interface, with a great possibility of interaction with other software in Windows room. A program controlling the simulation was written in Visual Basic, to elaborate and transferring to tyre simulation software the forces in the contact tyre – ground, besides simulates the pilot behaviour by making execute to the model the wished manoeuvres. This paper makes referring to a commercial vehicle for the design of multibody model (fig. 1) and the determination of geometric and inertial greatness.
In the first part the paper examines the dynamic behaviour of the motorcycle, engaged in a series of characteristic manoeuvres: slalom, steady turn, lane change, braking and acceleration executed with different speed and inclination angle. Visual Nastran permits the obtaining of the constraint reactions between the body constituting the model and the effecting of structural analysis by a FEM module, using the data obtained by the simulations.

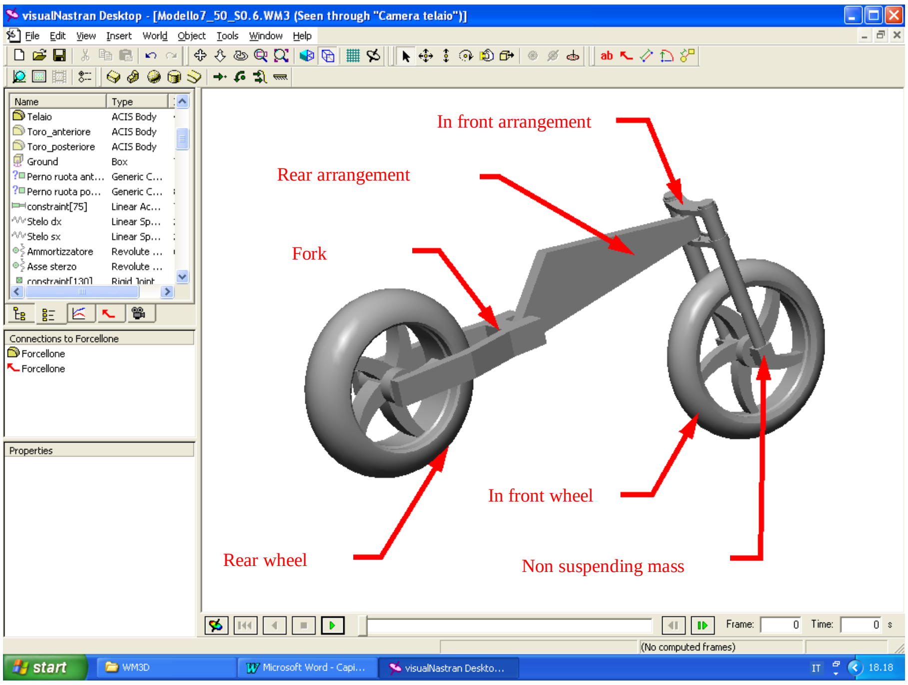

Figure 1. – Motorcycle multibody model

To exploit this possibility and to emphasise the importance and the potentiality of the developed instrument analysis, a study of the acting stress on the in front arrangement was conduced during the typical manoeuvres previously listing and during the crossing of a step. In this way the more onerous load condition were established and the mechanical more stressed component was located as the steering plate; a structural optimisation is conduced on this element by ANSYS software, to obtain a mass reduction and a more uniform distribution on the stress state of the element.

2. Motorcycle model

The motorcycle multibody model was building taking the characteristic measures, the mass distribution and the cinematic of the suspensions from a commercial motorcycle. It is a road vehicle of sorting type, equipped by a two-cylindrical motor 1000 cc, with in front suspension with telescopic type of fork and rear suspension with progressive cinematic oscillating fork. Visual Nastran permits the assembly of multibody models constituted by several rigid bodies assembled one another by a large series of constraints in an apposite library. The motorcycle was modelled as a system of six rigid bodies: rear wheel, in front wheel, rear arrangement, in front arrangement, non-suspending mass and fork.

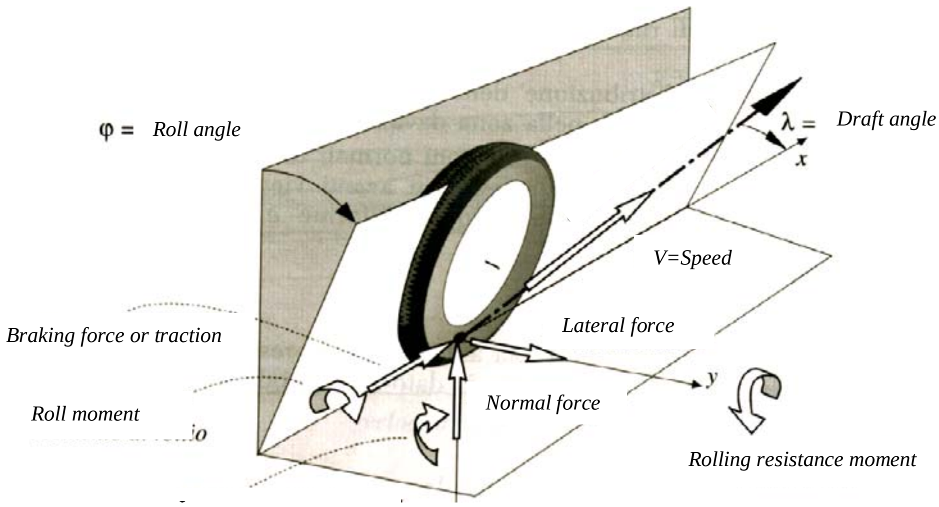

Figure 2 – Forces and moments acting on the tyre

The rear arrangement includes the chassis, the motor, the tank and all the mechanical components to they connected. The in front arrangement is constituted by the steering axis, by the handle, by the plates and the fore suspension; the non suspending mass consists of the fork rod, the caliper and all the mass non weighing on the elastic elements of the fore fork. The degrees of freedom of the six rigid body are reduced to eleven with the assembly and can be associated to the position of the barycentre of the rear assembly, to the rolling angle, to the pitch angle, to the yaw angle, to the steering angle, to the suspension range and to the rotation of the wheel around the own axis.

3. The tyre

The tyres constitute the interface between the vehicle and the road plane; thanks to their strain they maintain the contact between wheel and ground also in presence of little hollows, assure mobility and manoeuvrability to vehicle transferring to the wheels the forces due to the actions of the pilot by steering, brakes and accelerator. The interaction between tyre and ground is performed by the change of reactions around the contact point, constituted by three forces and three moments (fig. 2).

The action of the longitudinal forces generates strain in the contact zone between tyre and road. The speed on the wheel periphery is a few different than vehicle feed speed, due to the strain and stress on the tyre; this phenomenon is named slip and formula (1) defines the longitudinal slip coefficient:

$$\kappa = -\frac{V - \omega R}{V} \qquad (1)$$

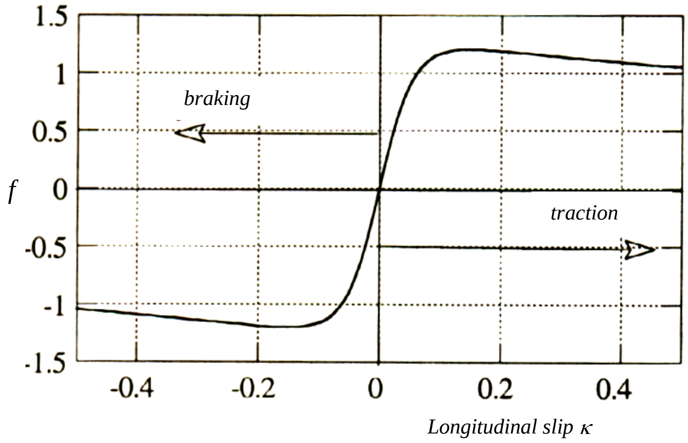

Figure 3 – Longitudinal friction coefficient f versus longitudinal slip κ

where V is the speed component in the direction of longitudinal axis of the vehicle, ω is the wheel angular speed and R the rolling radius. The slip is positive in the case of traction and negative in braking. Longitudinal force developed to the contact tyre-ground is depending on the vertical load and on the longitudinal slip; the ratio between longitudinal and vertical force at the sliding limits is equal to adherence coefficient. The fig. 3 shows a typical adherence coefficient versus the longitudinal slip; the maximum value is influenced by the ground conditions and by the material of the tyre tread. Instead the lateral force exerting by the tyre on the ground is dependent on both roll and drift angle; this last is defined as the angle, measured in the horizontal plane, between the motion direction and the intersection of the wheel mean plane with the road surface. It can be expressed by:

$$\lambda = -arctg(\frac{V}{V_x}) \qquad (2)$$

where V is the vehicle speed and V_x its component along the intersection between the wheel plane and the road one. The lateral force due to roll is due to the tyre strain in the contact points with the ground, while the lateral drift force is due to the tyre rolling and sliding with regard to the longitudinal axis (drift motion). The tyre ability to react to lateral forces is defined by rolling stiffness k_φ and drift stiffness k_λ , given by the following relationships:

$$k_\varphi = \frac{1}{F_z} \cdot \frac{\partial F_y}{\partial \varphi}\bigg|_{\substack{\varphi=0 \\ \lambda=0}} \quad (3)$$

$$k_\lambda = \frac{1}{F_z} \cdot \frac{\partial F_y}{\partial \lambda}\bigg|_{\substack{\varphi=0 \\ \lambda=0}} \quad (4)$$

they are the first partial derivative in the surrounding of the point origin, of the function representing the lateral friction coefficient of the tyre. In static condition the lateral force, developed in the contact tyre – ground, is given by the following formula:

$$F_y = (k_\lambda \cdot \lambda + k_\varphi \cdot \varphi) \cdot F_z \quad (5)$$

The force needs a certain time τ depending on the characteristic of the tyre and by the dynamic condition of the motion. In this paper a linear tyre model is used, so that the following relationship furnishes the lateral force:

$$\frac{L_r}{V} \cdot \frac{dF_y}{dt} + F_y = (k_\lambda \cdot \lambda + k_\varphi \cdot \varphi) \cdot F_z \quad (6)$$

where V is the speed and L_r is the relaxation length of the tyre, representing the space covered by the wheel when the force assumes the 63% of the final value. Formula (6) permits the separate determination of both drift force and roll force contribution; it is useful in the dynamical simulation to determine the directional behaviour of the vehicle. The figure 4 suggests the comparison between the tyre linear model adopted in this work and the *"Magic Formula"* developed by H. B. Pacejka. The draft reportes the values of the lateral forces versus the derive angle and versus roll angle, normalized with regard to normal loads.

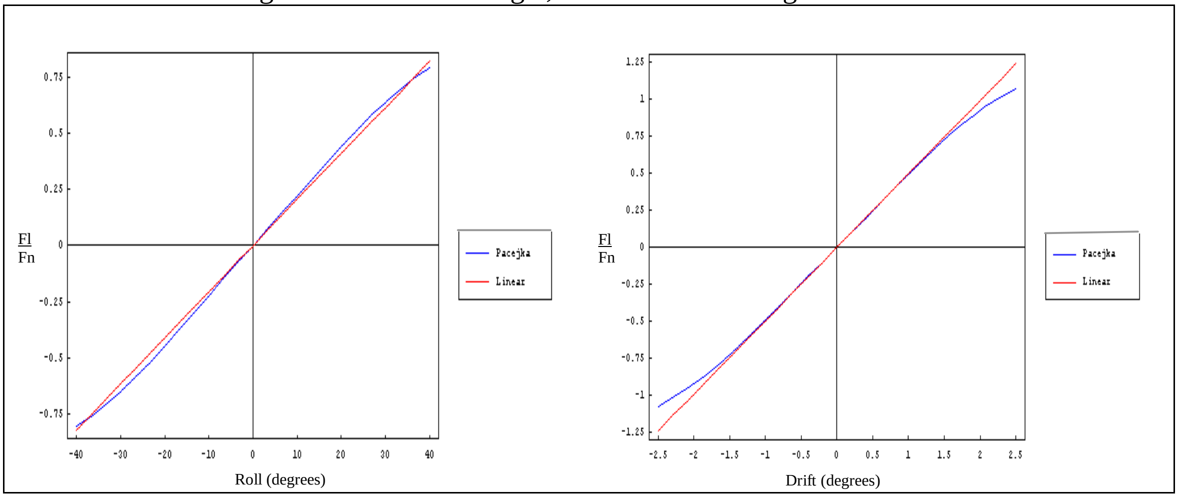

Figure 4 – Ratio lateral force / normal force versus drift and roll angles

4. The control program

The analysis of the dynamic behaviour of the motorcycle is very complex; the vehicle is unstable and at variable set up, and needs a control guarantying the directional stability and executing the required manoeuvres. In the reality these jobs are entrust to the pilot intervention by the steering, the gas command and the brake plant. The forces acting on the motorcycle and ones on interaction between tyre and road depend on many factors and follow very complex mathematical law. In the greater part of cases *Visual Nastran* is not in degree to implement such functions and when it is possible the data elaboration is very slow. To obviate this inconvenient elaborating a control program, written in *Visual Basic* interacting with *Visual Nastran* by the technology *OLE Automation* (*Object Linking and Embedding*). This is a fundamental characteristic of the programming at objects of *Visual Basic* and of the programs operating in *Microsoft Windows* room; it consists in software that integrates the operating system, permitting the interaction, based on *Client – Server* logic, against different application provided with *OLE* interface. Thanks to this technology the *Visual Basic* program *(Client)* is able to control *Visual Nastran* simulation *(server)* acquiring data by the object *Meters* and modifying the parameters of the analysis by the objects *Inputs*. The control program is provided with a graphical interface permitting to the user the setting up the characteristic parameters of the simulation and of the model (fig. 5). The program body can be divided in four principal modules:

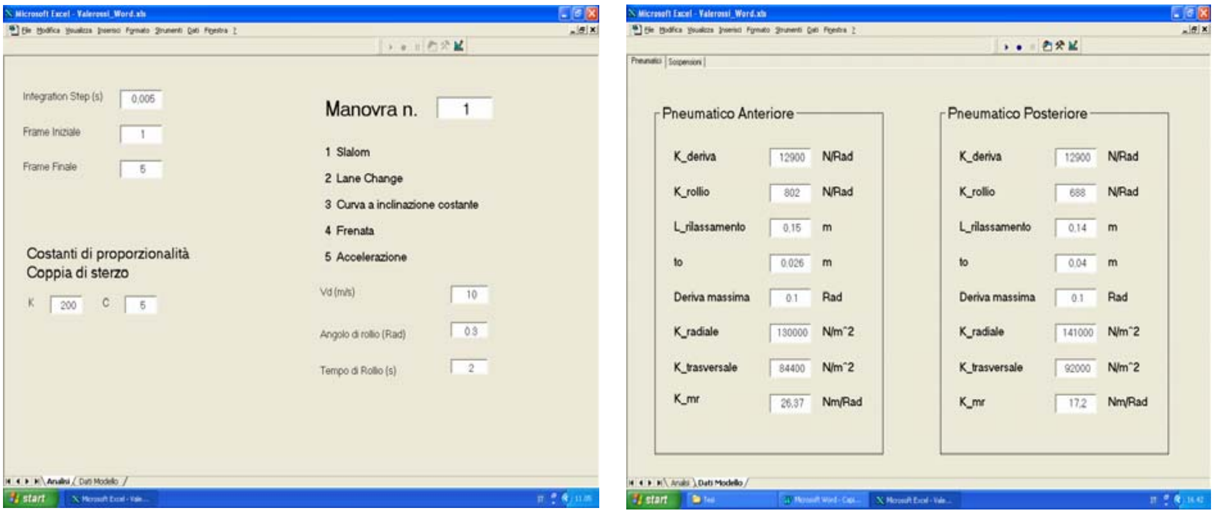

Figure 5 – Window of the control program setting up (Italian language)

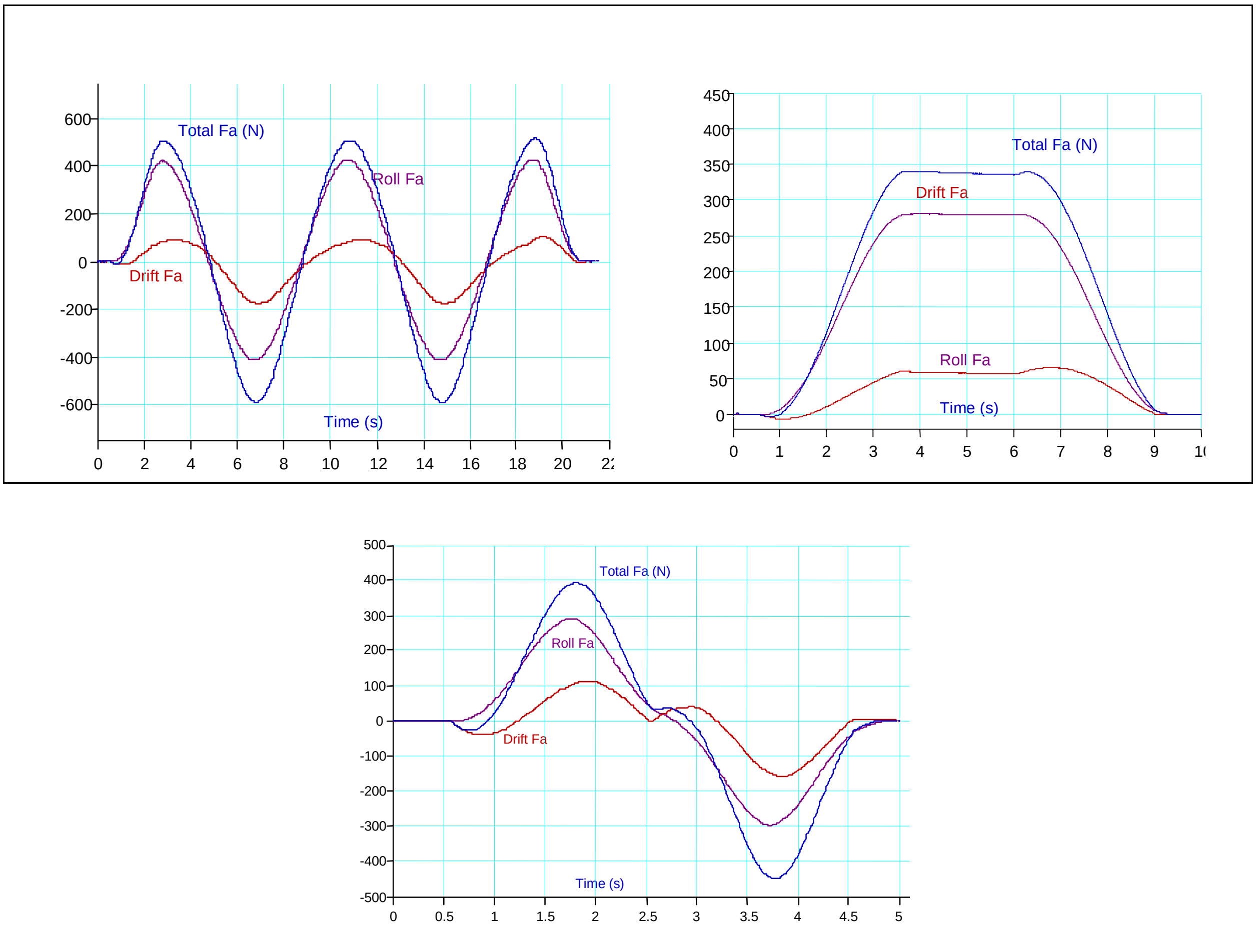

Figure 6 – Trend of reaction lateral force, of the roll and drift ones acting on the fore tyre during the simulations of Slalom, Steady Turn and Lane Change manoeuvres respectively.

- An *acquiring module*, importing the data from *Visual Nastran* simulation.
- An *elaborating module*, elaborating the acquired values to calculate the cinematic and dynamic characteristic greatness of the motion.
- A *force calculation module* calculating the acting forces on the model

- A *control module*, guarantying stability and directionality to the model simulating the pilot action trough the application of the steering couple.

5. Simulation of typical manoeuvres

According to the manoeuvre, the control program elaborates the trend versus the time of the roll angle; at every integration interval the codes compares the actual roll value with wished one and effects a correction of the trajectory in the case of discrepancy, applying to the steering a couple τ_s calculated according to the following relationship:

$$\tau_s = k \cdot (\varphi_d - \varphi) + c \cdot \frac{\partial(\varphi)}{\partial t} \tag{7}$$

where φ is the value of the actual roll angle, φ_d the wished roll value, *k* and *c* represent the proportionality constants of the first and second order of the control system. The system controls the vehicle speed comparing it with one planning out by the user. If the speed is greater than the wished one the control applies a braking force in the fore wheel, otherwise increases the value of the traction on the rear wheel to accelerate the vehicle.

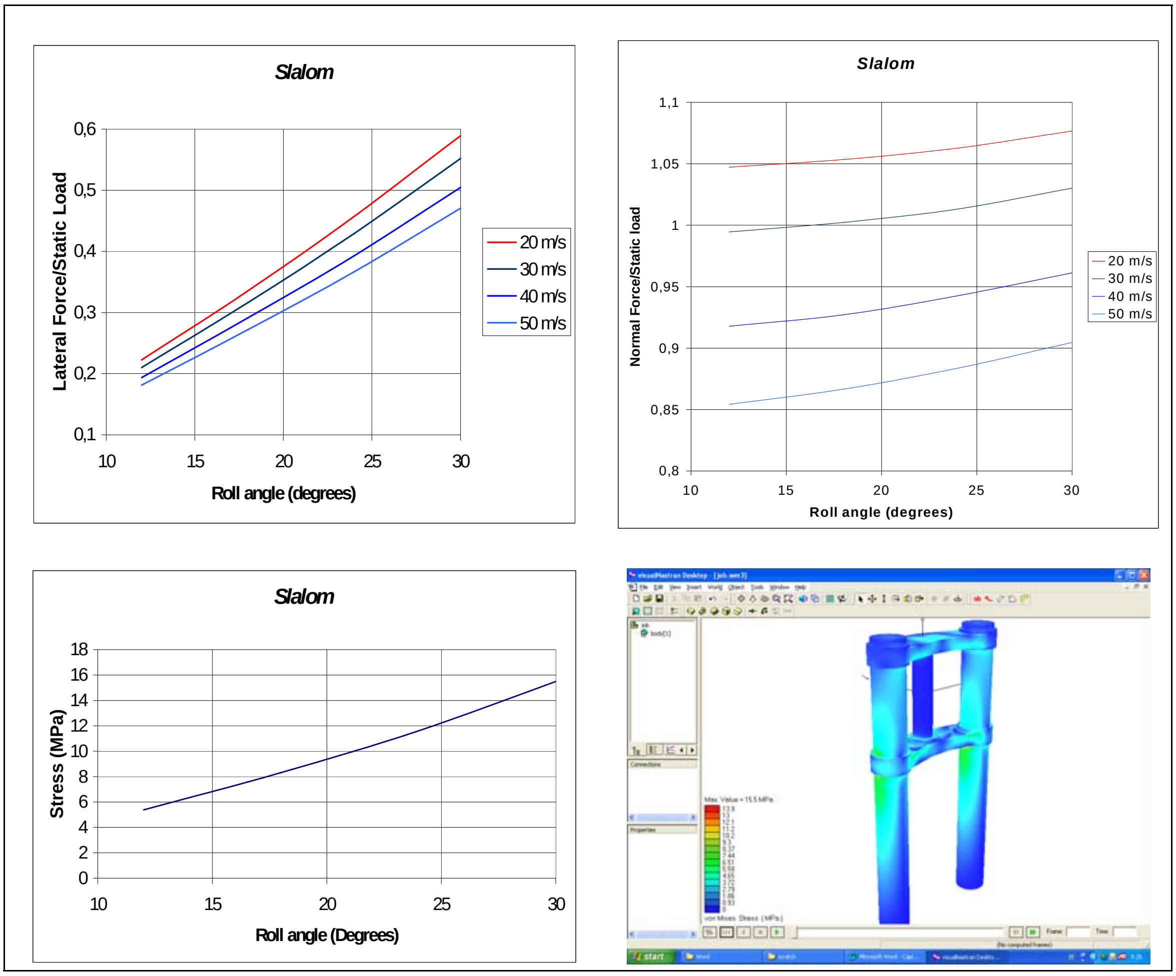

Figure 7 – Roll: trends of the maximum lateral and vertical forces versus the roll angle and to vary the feed speed; trend of the maximum stress versus the roll angle, distribution of the maximum stress in the fore arrangement.

To determine the acting loads on the fore arrangement a series of typical manoeuvres for the motorcycles in the road use: the manoeuvres of *Slalom*, *Lane Change* and *Steady Turn* were executed for increasing speed 20, 30, 40 e 50 *m/s* and for roll angles 12°, 18°, 24° e 30°. The braking test is executed simulating the maximum achievable deceleration to the limit of the sliding of the fore wheel; the manoeuvre of overcoming of a step of 5 cm was simulated for increasing speed from 8 until 20 m/s. The drafts reported in fig. 6 typical trend of lateral and reacting forces acting on the fore tyre obtained during the simulation of the manoeuvres of *Slalom*, *Steady Turn* and *Lane Change*.

6. Stress Analysis on the fore arrangement

The more stressed elements of fore arrangement during the manoeuvres of the motorcycle are the steering pin, the plates and the sheaths of the telescopic fork. The stress analysis on these elements is effected by the module *FEA* (*Finite Element Analysis*) of *Visual Nastran*, permitting the finite element analysis using the solicitations obtained by the dynamical simulation. The analysed mechanical models have geometrical characteristic and inertial responding to ones of the motorcycle referring.

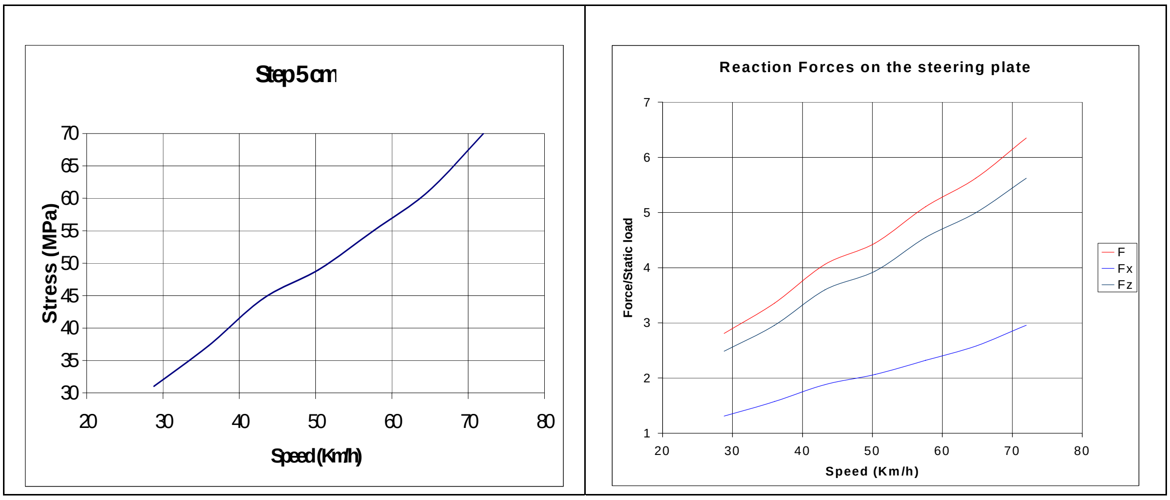

Figure 8 – Step 5 cm: trends of the forces and of maximum stress acting on the fore arrangement versus the vehicle feed speed

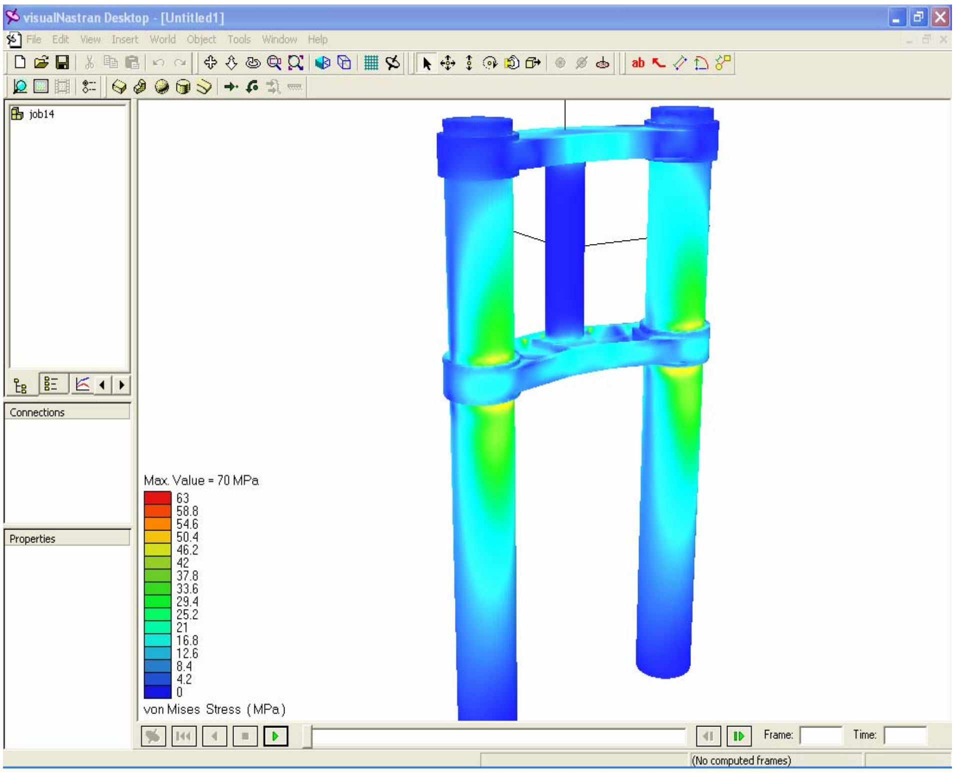

Figure 9 – Step 5 cm: distribution of the maximum stress acting on the fore arrangement

Figure 7, 8 and 9 report the trends of the forces and of maximum stress acting on such elements for the step overcoming and for Slalom manoeuvre, varying of the angles of roll and of the feed speed. The stress analysis highlight that the greater stress are in correspondence of the connection between the inferior steering plate and the telescopic fork sheaths and the more onerous manoeuvres are the overcoming of the step of 5 cm at the speed of 72 km/h, the breaking at the adherence limit of the fore tyre and the slalom manoeuvre.

7. Structural optimization of the inferior steering plate

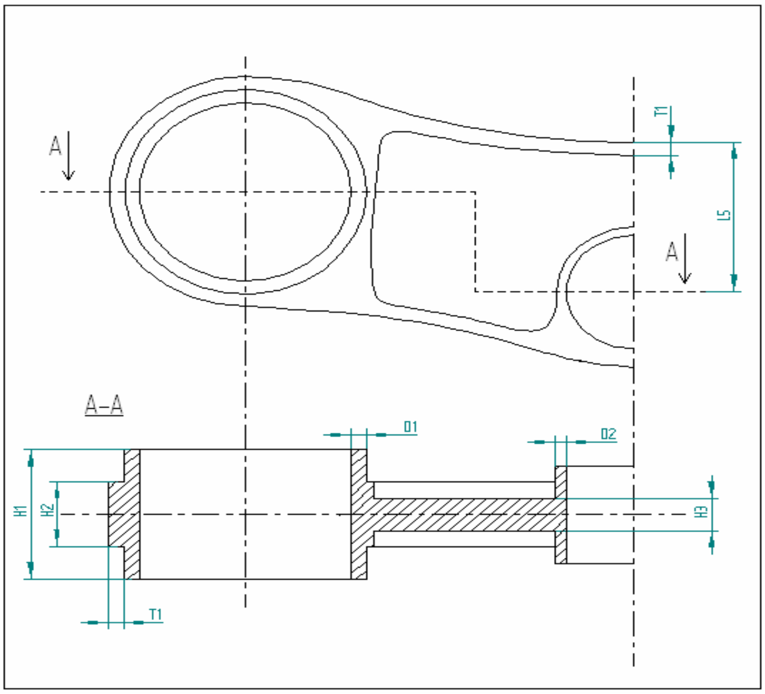

Figure 10 – Independent variables of design for the steering plate optimisation

The modest entity of the calculated maximum stress induced the Authors to effect the structural optimisation of the steering plate, to obtain a best distribution of the stress state, associated to the mass reduction of the examined component. The analysis was performed by ANSYS program, with the apposite module named *Design Optimisation.* The optimisation procedure requires the realisation of a parametric model of the structure to analyze by making use of parameters constituting the independent variables of the design; varying the values within the time limits defined by the user, the calculation program minimises the objective function representing the design characteristic to optimise.

DESIGN VARIABLES				
	LIMITS		INITIAL VALUE	OPTIMUM VALUE
	Min	*Max*		
H1 (*mm*)	30	45	40	30.02
H2 (*mm*)	13	26	20	14.2
H3 (*mm*)	3	12	10	8.8
T1 (*mm*)	2	6	5	3.3
D2 (*mm*)	2	6	5	3.4
D1 (*mm*)	2	6	5	2.01
L5 (*mm*)	38	45	44	40.01
STATE VARIABLES				
σ_{max} (*N/mm*^{2})	Breaking	141	53	94
σ_{max} (*N/mm*^{2})	Step	250	141.3	249.97
OBJECTIVE FUNCTION				
Total Volume (*mm*^{3})			$25.97 \cdot 10^4$	$14.05 \cdot 10^4$

Table 1 – Variables and objective function of the optimisation process

Other constraints on the problem are introduced by the state variable definition; they represent the design requirements subjected to limitations. The structural optimisation of the steering plate was executed choosing the volume reduction of the component as objective, considering uniform the material density that is equivalent to minimise the structure weight. The constraint on the state variable is the maximum value of Von Mises stress, calculated by FEM step by step.

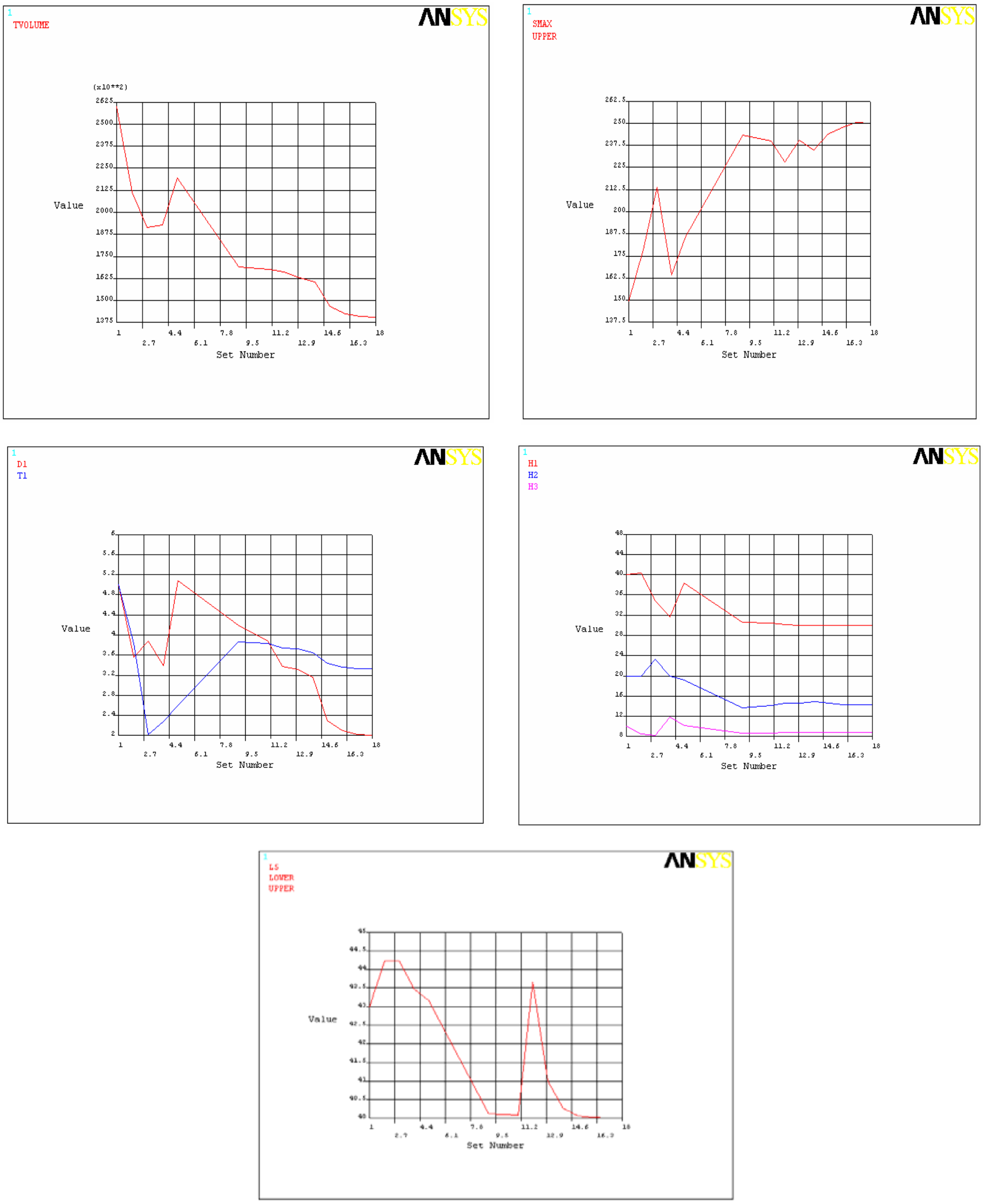

Figure 11 – Trend of objective function (top, on the left) and state variable (top, on the right)
In the centre and on the bottom trend of design variables during the simulation

The design variables are represented in fig. 10 and reassumed in tab. 1. The table shows the imposed constraints, the initial and the final values obtained during the optimisation process. To determine the maximum stress to introduce as superior limit of the state variable, a comparison is made between the more onerous load condition for the inferior steering plate during the breaking to the adherence limit and during the overcoming of the step of 5 cm with a velocity of 72 km/h. For a motorcycle of road type the manoeuvre of overcoming of a step a high velocity is an event of occasional type, that is verified rarely during all the

vehicle life, hence a verify at static resistance was considered sufficient, applying a high security coefficient (n=3).
The verification of the fatigue resistance in the braking manoeuvre is executed using Sodeberg method with a security coefficient equal to 2. The values of limit stress calculated for braking and step solicitations are reported in tab. 1 with the respective stress acting on the structure original and optimised one; at the optimisation end the state variable reaches the limit value in correspondence of the solicitation deriving by the step-overcoming manoeuvre.
The figure 11 shows the values assumed by the objective function and by the state variable at every optimisation step, besides shows the trends of the design variables. The figure 12 propose the comparison between Von Mises stress acting on the original steering plate and on the optimised one. One can note that the stress state is more uniform on all the structure and that the percentage reduction of the total mass reaches 45,9% of the original value.

8. Conclusions

The technique set in this paper is able to facilitate the choices that may be executed during the design of a motorcycle. It use the software Visual Nastran 4D with the multibody technique; the description of the phenomenon of the longitudinal and lateral adherence is executed by a Visual Basic program interacting with Visual Nastran, set by Giannitrapani Author. At the purpose a linear model of the tyre behaviour is used with satisfactory results in a large field of variability of the drifts and sliding, with regard to Pacejka magic formula.
The interaction between Visual Basic and Visual Nastran permitted the accurate analysis, in a suitable elaboration time, of simulations regarding the typical manoeuvres, characterising the road behaviour of a motorcycle, as Slalom, Steady Turn, Lane Change and the step overcoming. The simulation permitted the stress analysis on all the motorcycle parts and also the application of sophisticated optimisation procedures with the purpose to improve the efficiency of the parts equilibrating the solicitation and reducing the masses.
The optimised used technique is ANSYS one with the Design Optimisation module: the procedure was applied to the steering plate as example, obtaining a reduction of the original volume equal to 46%.

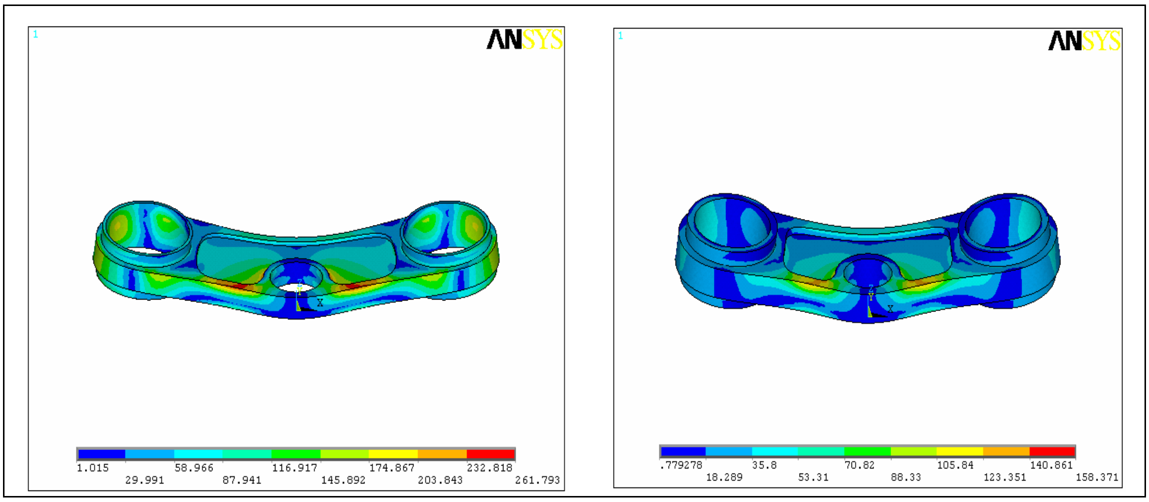

Figure 12 – Stress distribution on the original structure(on the right)
and optimised one (on the left).

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