Three-Wheeler Performance Optimization: Dynamics-Based Design Of A First Prototype

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ABSTRACT: The mobility is an increasing problem in cities. Everywhere, teams develop new technologies and vehicles to face this problem. To succeed in designing such vehicles, engineers have to deal with mechanical design, manufacturing but also with dynamics. The present work describes an optimization method especially dedicated for high dynamic vehicle purposes and its application to an innovative vehicle: an electric titling three-wheeler. In particular, this paper describes special mechanical arrangements implemented to satisfy severe geometrical specifications, and to obtain the highest dynamic capabilities.

1 INTRODUCTION

Due to urban zone densification and energy rarefaction, some facets of life habits have to be revised. The mobility doesn't derogate from this trend and is one of the major future challenges. Automotive industry is developing new solutions to cope with the increasing problem of mobility, the need for energy efficiency and customer requirements. Facing this multiplication of objectives, often conflicting, it is quite unlikely that one particular solution would satisfy all customers in all daily needs as it was with the car until now. Several new kinds of vehicles appear, each of them being able to answer a particular use. In the special case of urban and personal mobility, tilting three-wheelers seem to be a promising solution. Small and agile, they improve the traffic flow while the associated reduction of weight allows better energy efficiency.

Because of the increase – in number and quality – of the criteria imposed to tomorrow's vehicles, the industry must propose new types of morphologies, incorporate new technologies and detect a maximum of synergies between the latter. Thus we observe a constant increasing design tasks complexity while the development times are shorter than ever. There is a real need for global design methodologies that include, from the earliest stage of the process, a multitude of components among which the dynamics takes place.

This work aims at developing a design methodology especially dedicated to road vehicles. The method has the particularity to enable to manage the trade-off between dynamic performances and mechanical feasibility. The method is being applied to a new three-wheeler under development in our laboratory. The next section describes the main characteristics of this vehicle [1]. Afterwards, we summarize the optimization method developed and applied to the three-wheeler: more details can be found in [1] and [2].

In the present paper, we will explain how the design of a first prototype can be achieved on the basis of the optimization processes. In particular, we will describe some very specific mechanical arrangements especially designed to maximize the dynamic performances of the tilting vehicle suspensions.

2 VEHICLE DESCRIPTION

As mentioned above, the vehicle developed in this work is a tilting three-wheeler. The selected morphology is the reverse trike type: two wheels are mounted on the front suspension. The rear wheel is motorized and uses, in the manner of a motorbike, a swing-arm suspension. The front suspension is the most challenging part of the vehicle. It must satisfy severe dimensional constraints, especially a low width of 0.8m. Due to this low width, the vehicle stability requirement imposes to tilt the vehicle when it turns. Directly linked to the maximal lateral acceleration, the maximal tilting angle must be chosen as large as possible. Added to the suspension own motion, it leads to extremely large wheel travel and to bothersome geometrical interactions between different parts.

This kind of suspension is already used in existing vehicles as for example the Peugeot Metropolis, Piaggo MP3 and Quadro 3. But in our application, we stabilize the vehicle, for driver convenience reasons, by controlling the tilting motion via an actuator. By this way, the driver only manages his trajectory but not his tilt like he must do with bikes, motorbikes and commercial three-wheelers. Direct tilting control is used to stabilize the vehicle. From a practical point of view, it is implemented as follows: a mechanical component called “T-pendulum” (in green in Fig. 3) is articulated on the same axis as the bottom wishbone. The top connections of the shocks are not fixed to the chassis, as usual, but to this T-pendulum, conferring the required tilting degree of freedom. Finally, the tilt motion is actively controlled via a torque applied between the chassis and the T-pendulum.

The complexity, the number and the interdependence of issues involved make this vehicle design a hard optimization problem to handle.

3 OPTIMIZATION METHOD

To succeed in optimizing the dynamic performances of a vehicle while taking a large variety of design constraints into account, a numerical optimization process has been developed [1]. The method consists in partitioning both the criteria and the parameters of the problem. Some parameters are set to satisfy some criteria formulated as *constraints* while others are determined to optimize some criteria formulated as *objectives*.

In the process, depicted in Figure 1, an optimizer acts on a series of independent parameters linked to the objectives. A trial consists of 4 main parts. The data conditioner set parameters to ensure satisfying all types of constraints: geometrical, frequency-based and analytical. Then, the resulting vehicle is tested in both a dynamic test and a quasi-static equilibrium analysis. Finally, based on criteria recorded during these tests, a global objective function, F_0, is computed to allow the optimization process to iterate.

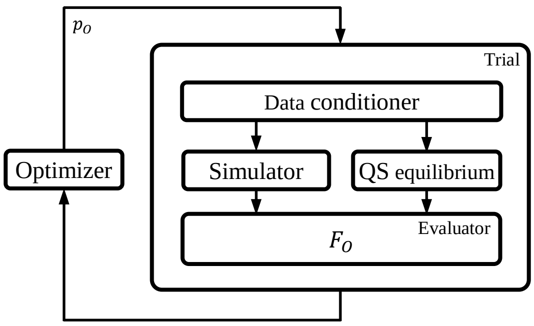

Figure 1. The optimization scheme

The process has multiple advantages. First, thanks to the data conditioner, the optimizer doesn’t deal with any constraints whatever their nature: geometrical, kinematic or dynamic. Secondly, each vehicle simulated in the hybrid test, i.e. quasi-static and dynamics, which is the most costly part of the process, satisfies all the constraints. Thirdly, despite of the complexity of the problem, the total computing cost remains acceptable: around 3 hours on a 50 cores cluster.

4 PROTOTYPE IMPLEMENTATION

Based on the solution of the above optimization process, we have designed a first prototype of the three-wheeler. Due to high compactness and severe dynamic performance requirements, the mechanical design is rather unusual as shown in Figure 2. The underlying techniques have been sometimes borrowed from car or motorbike designs or they are completely specific.

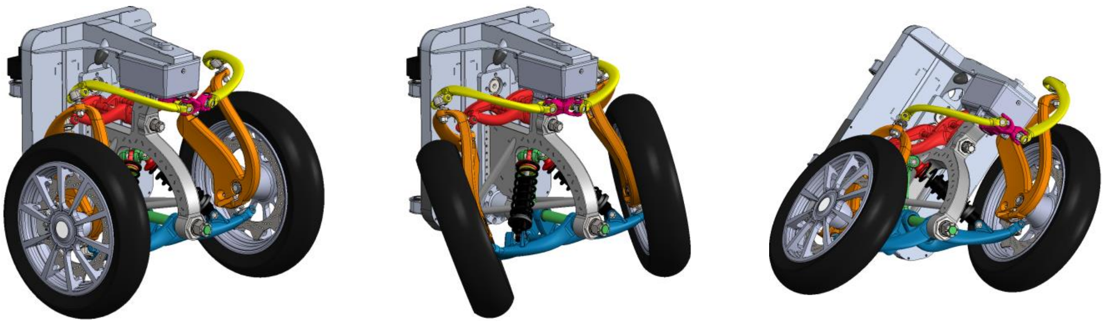

Figure 2. The front suspension: neutral, max. steering, max. tilting configurations.

In the following of this section, some specific mechanical arrangements are presented. The goal is to highlight the conflicts between design and dynamics requirements, which are numerous and complex in the case of the three-wheeler. In each case the result is a trade-off between both points of view.

4.1 *Reduction of deflections and of vibrations*

The optimization method described above is based on multibody dynamic simulations. In this case, a rigid body formalism is used to keep the time cost of optimization acceptable. The deflections of bodies under efforts and the associated high frequency vibrations are not taking into account. However, the selected morphology and the mechanical implementation of optimization results have to target to highest stiffness for suspension components in order to maintain the deflections and vibrations to an admissible level. There are multiple raisons to consider it.

First, the deflections induced by efforts throughout the suspension can modify the orientations of the wheels with respect to direction of travel leading, in general, to a loss of dynamic capabilities.

Secondly, the low frequency vibrations are transmitted to vehicle interior and to the driver whereas the high frequencies are partially transformed into noise. Both these elements affect dramatically the comfort of the driver. Even worse, the vibrations that are transmitted everywhere bring, on their paths, multiple mechanical damages whose the most known are screws and nuts loosening, fatigue cracking and breakings.

Moreover the own stiffness of suspension parts, a key decision in the axle design is its general morphology. A serial structure provides decoupling between 3 main functions for the front axle: suspension, steering and tilting whereas a parallel structure is lighter and is less subjected to deflections and vibrations. With references to market, the three-wheeler Piaggo MP3 uses a serial structure. For this implementation, one can observe parasite vibrations, visible to the naked eyes. The Peugeot Metropolis and Quadro 3 use a parallel structure. We opted for the same choice.

4.2 *The combination of titling and steering motion*

The most specificity of this vehicle is its roll motion called tilting. As for a bike or a motorbike, the tilt angle is directly linked to the lateral acceleration and then to the curve radius and the speed. In fact, the maximum tilt angle must be as large as possible in order to increase the dynamic capabilities of the vehicle. Nevertheless, the tilting angle involves large displacement of the front axle wheels. The max tilt angle must be determined as a trade-off between the limit of dynamic capabilities and the mechanical complexity of the front suspensions. In this respect, we target to reach a maximum lateral acceleration 0.7 m/s². By this way, the maximum tilt angle is fixed to 35°.

Due to urban targeted use, the vehicle must provide an agile handling especially for manoeuvre, for which the turn radius must be as low as possible. To reach a turn radius of turn of 2.8 m, which is in the common range for this kind of vehicle, the maximum steering angle of the front wheels must be pushed to 40°.

The combined tilting and steering angles for such amplitudes are outright not compatible with standard ball joints used as connection between wishbones and carriers, and at the end of the steering rods. Standard ball joint are built so that the main rotation can be unlimited while the maximum amplitude in the two auxiliary axes extends on a range from 2x10° to 2x30°depending of the mechanical implementation. The upper limit of this range corresponding to weakest models, those are non-suitable for vehicle applications. As done is some off-road prototypes in order to increase the wheel travels or in drifting race cars in order to extent the steering angle, we need specific ball joints allowing large amplitude of motion in, as a minimum, two directions. The practical solution consists in a combination of a traditional ball joint with an additional pivot perpendicular to the main axis of the ball joint. Obviously, the arrangement proposes only 3 true degrees of freedom, the additional one is redundant but replaces the ball joint one by extension of its amplitude.

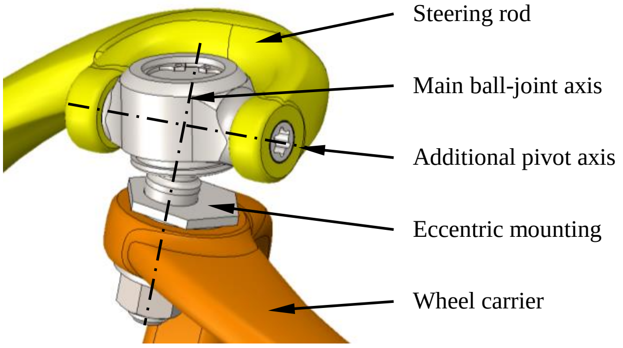

Figure 3. The customized ball joint.

This mechanical implementation is more complex than a standard ball joint but allow targeted steering and tilting amplitude. However, its cumbersome design involves other drawbacks explained bellow.

4.3 *The narrow track*

The key specification of this vehicle is the overall width, targeted to 0.8 m as it must be able to go through a standard door. This imposes a lot of severe design constraints to the front axle. Moreover, because of the tire width, the track is reduced to 0.7m.

In general, the width between the wheels is split in four mains types of zones. The structure consists of the frame on which other elements are fixed. The useful space is dedicated to the driver, passengers and interfaces with them. The auxiliary space is used to place engine and fuel tank, in case of a fuel car, or electric motor and battery, in case of an electric car, as well as all other auxiliary components. The suspension zone is reserved to all moving parts of the axle in their motions due to degrees of freedom: suspension motion, steering and tilting in this case.

In the case of the three-wheeler, the track is so reduced that it is impossible to envisage having some space between wheels to be used as a useful or an auxiliary space. The cockpit comes behind the front suspension unit leading a significant increasing of the wheel base, compared to a motorbike for example. The motorization is realized at the rear wheel.

Furthermore, the optimization, for which the suspension part geometry and their anchor point locations have been considered as parameters, shows that the best setup maximizes the length of wishbones and tends to merge their axles along the median plane of the vehicle. Then, the structure is reduced to a simple narrow panel centered on the median plane. Such structure does not provide a sufficient mechanical resistance and must be reinforced by two triangulation bars. The location of these bars and the close moving part designs must be carefully analysed to avoid any interactions. It explains partially the twisted shape of the bottom wishbones.

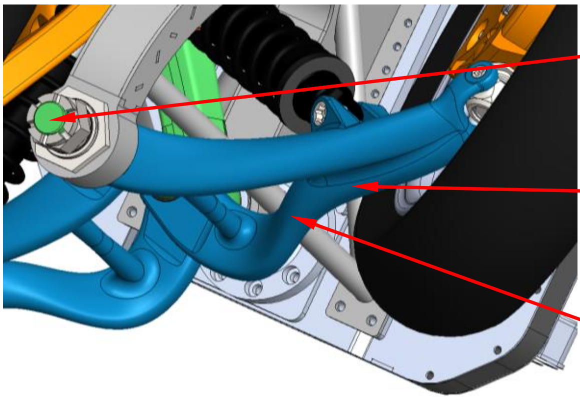

Figure 4. The front axle structure design and related interactions.

As the space between wheels is completely full, the steering mechanism must be pushed up. Two different morphologies have been envisaged: a rack system based on a translational motion and a steering box based on a rotational motion. Ones more, the optimization method tries to maximize the steering rod lengths promoting the steering box option. Independently of the rod lengths, the rotation motion gives better results than the translation. Then, a steering box is finally preferred to a rack.

The motions of the steering rods are so large than their shapes must be curved to avoid interactions with the box itself. Two consequences of this particular shape must be noticed. First, the curve must be oriented in space in order to avoid the box. Then, the rotation around the main axis of the steering rod cannot be free as it is the case, in a given range, for traditional car steering rods. To face this requirement, the ball joint is replaced by universal joint at the internal end of rods. The deleted degree of freedom corresponds to the inconvenient rotation of rod. Secondly, the curved shape of rod makes difficult or impossible the elongation of its own body in order to set the toe of front train. The easiest way to implement the toe setting is to mount the ball joint pin with respect to wheel carrier via an eccentric mounting. Similar arrangement is used for ball joints between carrier and wishbones in order to set the camber, the caster and the caster angle (combined settings).

4.4 *The location of the bottom ball joint of the wheel carrier*

Another troublesome results from the optimization process is the location of the ball joint between the down wishbone and the wheel carrier which is very close to the axle, as close as possible in fact. Also, it must be deeply embedded as possible inside the rim. The point location is all the more critical because, in the three-wheeler, no steering assistance is present to correct an inadequate steering torque that could be caused by a wrong caster and/or a wrong scrub radius. This problem is again reinforced by the cumbersome design of a specific ball described above.

A first problem is the interaction with the front suspension brake system. Figure 5, (right) illustrates the traditional arrangement for wheel, spindle and brake system. The optimized position for the bottom wishbone ball joint is located and his cumbersome zone is also represented by a dotted circle. The unwelcome interaction between the ball joint and the brake disc is well visible. This is counteracted by uncommon mechanical arrangement for the front suspension brake system: a peripheral brake is used instead of a traditional one (Fig. 5 left) to free up space around the optimal ball joint position.

There are several drawbacks due to the peripheral brake. The major one appears in case of disassembling the wheel. It imposes to disassemble the brake caliper from the carrier, to allow disengaging the wheel out of the spindle. On the contrary, because the wheel hub does not support the brake disc anymore, it can be “melted” with the rim. It reduces the unsprung masses with the well-known good repercussions on the dynamic behavior.

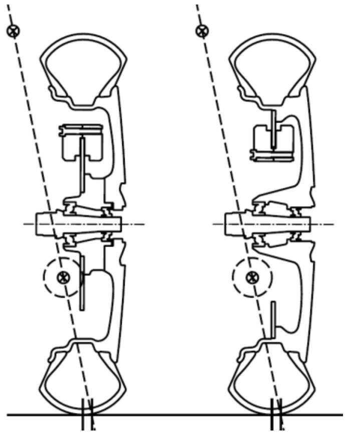

Figure 5. Front brake alternative.

Another consequence of the ball joint location is related to the motorization. The wheel hub size is limited, leading to difficulties or the impossibility to insert an electric actuator to form an in-wheel-motor. Anyway, this technical option was previously rejected for its negative influences on unsprung masses. This consideration closes definitively the question about the front wheel motorization; a transmission via a universal joint, or other similar systems, having been excluded because totally incompatible with the front suspension kinematics.

5 CONCLUSION

Throughout this paper, we developed the way of design of an innovative urban electric tilting three-wheeler. Based on white paper sheet, we started by the selection of a morphology adapted to the severe specifications imposed to the vehicle. Then, we developed an optimization method able to increase the dynamics capabilities of the vehicle by changing the model parameters. The method deeply described in other references, was applied with success to the three-wheeler. The unusual specifications lead unusual geometry whose the mechanical implementation was not straight forwards. A significant work was done to converge to the best kinematic, possible for implementation and suitable regarding the dynamics. As shown in this paper, numerous mechanical arrangements had to be adapted, combined, or redesigned in order to satisfy the optimized geometry. From the start of the project, the watchword was “Thing out the box” to find innovative solutions. The obtained three-wheeler must be efficient with respect to its specifications and in particular to dynamics. But it leads a complex front axle for which specific designed parts are required in the most of case. Despite the charge of work, we hope than a prototype will be assembled into the next months. It would allow measuring the optimization method and implementation efficiency. The future will indicate if the resulting complexity can be justified by the need of better mobility solutions.

REFERENCES

[1] Verlé, Aubain & Fisette, Paul 2015. Design of an active tilting three-wheeler: optimization of the dynamic performances via multibody simulations. *Proceeding of The 24th International Symposium on Dynamics of Vehicles on Roads and Tracks*. Graz, Austria, 2015.

[2] Verlé, Aubain & Fisette, Paul 2016. Generalized quasi-static equilibrium computation for vehicle dynamics optimization, *Proceeding of The 4th Joint International Conference on Multibody System Dynamics*. Montreal, Canada, 2016.