Synthesis of spatial parallel mechanisms for a vertical and longitudinal all-terrain suspension

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Abstract. Fast wheeled motion on unstructured grounds requires highly efficient suspensions that damp shocks vertically but also horizontally, which is an original contribution of the author. This work describes nine 2D and 3D kinematics, most of them with parallel structure, that are suitable for guiding a wheel and providing simultaneous damping in two directions. Steering and power transmission are also included in the most advanced variants, that were previously patented. Both structural and dimensional synthesis are presented, with a kinematic description of each system. A real implementation at a small scale is also described.

Introduction

Wheels are mostly suitable for motion on C^1 continuous surfaces (tangency continuity). In unstructured environments, obstacles may provide only C^0 continuity (contour continuity), as seen in Fig.1a. The worst case occurs for obstacles that do not even have C^0 continuity. They will be not treated in this work.

The conventional axis system in this paper uses *X* in the direction of longitudinal motion, *Z* in the ascending direction and *Y* oriented laterally so that *(X,Y,Z)* is direct. Generally, obstacles can be considered as shapes with a front surface being roughly vertical along Z, *i.e.* with a strong component of their normal vector along *-X* (Fig.1b). From this statement arose the concept of a suspension allowing also the longitudinal *X* damping motion for better obstacle-crossing.

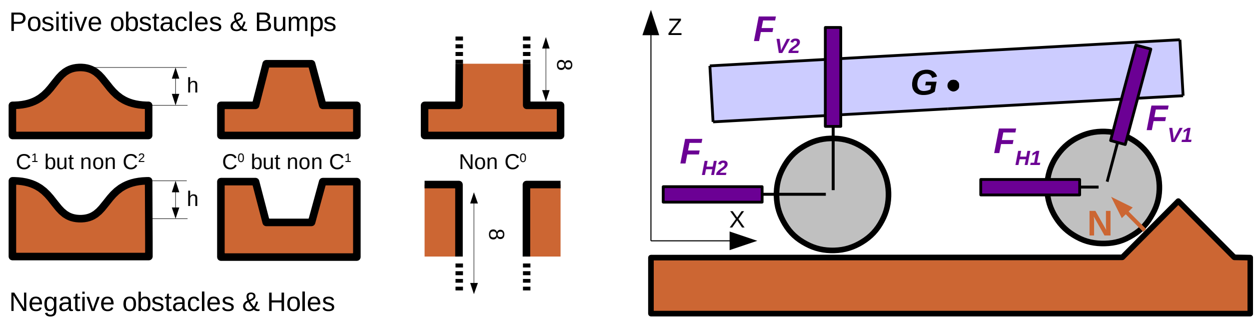

Fig.1. (a) Considered obstacles have a C^1 continuity and possibly only C^0. (b) Obstacles generally have a front surface whose normal vector N may have a strong horizontal component along -X.

A previous work [1] allowed to build a multibody simulation of a simplified 2D vehicle equipped by suspensions with 2 degrees of freedom (2DOF) : a vertical one along *Z*, like most of vehicle suspensions, and also a horizontal one along X, suitable to cross high obstacles with a quasi-vertical front surface. The preliminary results were encouraging and the model equipped by a 2DOF suspension could cross higher obstacles in simulation than the model with classical suspensions. Additionally, an experimental work [2] on a vehicle with classical 1DOF double-wishbone suspensions (Fig.2a) also showed that it exists a stability limit for obstacle-crossing, above which the vehicle tips-over (Fig.2b). The 2DOF *XZ* suspension is expected to extend the stability domain by allowing to cross higher obstacles at a given speed (or the same obstacles at a higher speed).

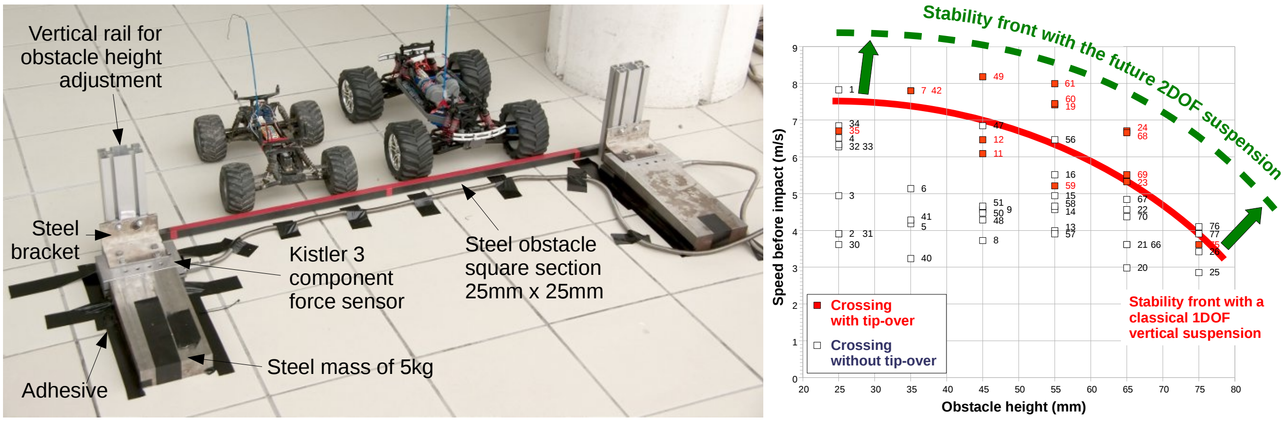

Fig.2. (a) Experimental obstacle-crossing with 1 DOF suspension. (b) Stability front of classical vehicles with dual 1DOF Z suspensions: impact speed vs. obstacle height. 2DOF XZ suspensions should extend the stability limit [2].

Existing all-terrain vehicles mostly rely on wheels that are of great diameter with respect to the obstacles to cross. Most of the fast commercial all-terrain vehicles such as cars, military trucks, ATVs or buggies rely on robust rigid axles or more comfortable double wishbone suspensions, (Fig.3a-d). Some original architectures can be noticed on small scale radio-controlled cars so-called “crawlers”, where each axle is guided by a parallel mechanism including generally four bars with double spherical joints, denoted *S* joints (Fig.3e). Some research platforms such as RobuROC 6 also have joints between axles but not necessarily suspensions (Fig.3f). Both crawlers and RobuROC 6 are suitable for high-obstacle crossing but only at slow speed. No commercial vehicle appears to allow a long-travel longitudinal suspension of its wheels.

The analysis of numerous vehicle suspension patents also confirmed that longitudinal suspension is an original concept. All the suspensions that use a trailing arm guide the wheel along a circular trajectory. The wheel centre motion has a minor longitudinal component, but it is coupled with the vertical component. Trailing arms represent the majority of motorbike rear suspensions, but they can also be transposed into leading arms for front suspensions (Fig.4a, [3]). Front and rear arms can also be coupled longitudinally (Fig.4b, [4]) to obtain a specific global behaviour of the vehicle. Because trailing arms can transmit a lot of the longitudinal contact force when the wheels impact against an obstacle, some suspensions use deformable trailing arms that

Fig.3. Existing all-terrain vehicles for fast (a-d) or slow (e-f) speed motion.

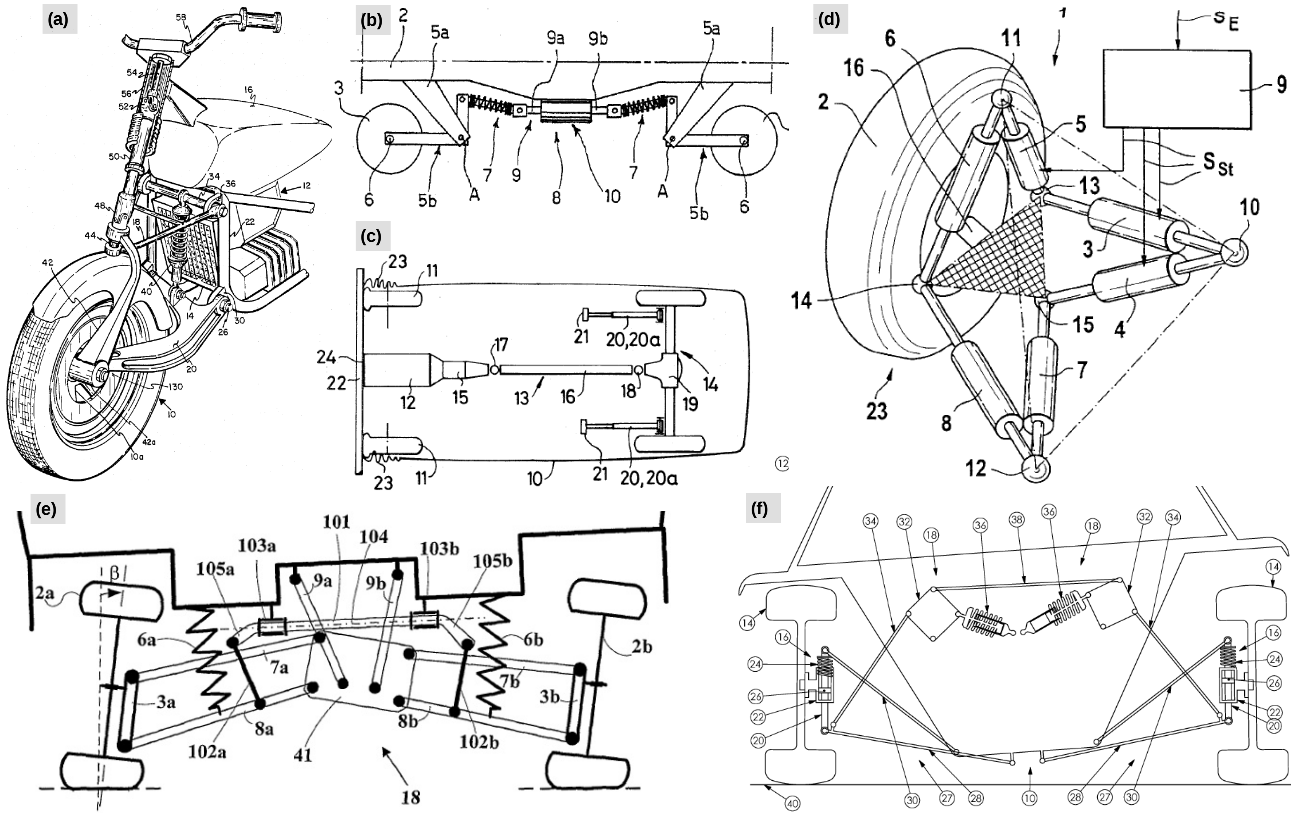

Fig.4. Selection of patents for suspension mechanisms.

help to absorb energy during accidental crashes, but absorption can be done only once (Fig. 4c, [5]). Other suspensions are capable to provide 6DOF, including a longitudinal translation of the wheel, by guiding the wheel with a Gough-Stewart parallel mechanism, but at the price of a complex control because of highly coupled motions (Fig.4d, [6]). Several devices such as the Michelin-IFMA Optimized Contact Patch suspension (Fig.4e, [7]) or the Sacli suspension (Fig.4f, [8]) take advantage of the centrifugal force applied to the vehicle towards the outside of the turns to adjust the camber angle of the wheels and improve adherence. These systems allow lateral but no longitudinal translation of the wheel contact-patch with respect to the vehicle frame. Although no suspension seems to allow wide longitudinal mobility of the wheels, most of the existing suspensions allow small motions, because they replace the classical spherical and revolute joints by compliant components made of rubber injected between metallic parts and also known as “bushings”. Bushings improve longitudinal comfort and vibration filtration for high frequencies. However, they only allow very short motion travel.

From this overview, it appears that new suspensions should be designed to absorb both vertical and horizontal components of reaction forces generated by contact shocks of the wheel against rough obstacles in all-terrain. The longitudinal and vertical motions should be of the same order of magnitude. For being used both on front and rear axles, this new mechanism should also include steering and transmission capabilities, so the so-called 2DOF-suspension could become a 4DOF mechanism according to the considered options. The prospect of a potential active suspension leads to the following constraint : the horizontal (respectively vertical) damper compression should correspond mostly to a horizontal (resp. vertical) motion of the wheel. This means that horizontal and vertical motions of the wheel should be as decoupled as possible, and also decoupled from the steering and transmission motions.

In the next section are introduced the very first 2D kinematics that were proposed for the required suspension. The following section focusses on 3D structural synthesis of realistic mechanisms providing good lateral guidance and managing the coupling phenomenon. Finally, dimensional synthesis allows to finalize the geometric setting and to build a demonstrator for the front suspension of an all-terrain car at scale 1:8.

2D structural synthesis

In the nine kinematics presented in the following sections, parts will always be designated according to a three-digit naming convention represented by *VSP* patterns, where *V* stands for the kinematics version, *S* for the considered system and *P* for the part in the system. A system *S* can be an assembly of components fixed together or a linkage. A *VS0* pattern designates the considered system *S* of a version *V*. Table 1 summarizes some examples of systems and parts pertaining to all the suspension versions.

V Suspension Version number (from 1 to 9)

Name *VSP*	System (assembly or linkage) *S*		Part *P*	
V00	*0*	Frame		
V10	*1*	Wheel assembly		
V11			*1*	Hub of the wheel
V20	*2*	Hub-carrier assembly		
V30	*3*	Linkage for lateral guidance		
V31			*1*	First bar
V32			*2*	Second bar
V40	*4*	Linkage for vertical suspension		
V41			*1*	Spring vertical suspension
V42			*2*	Damper vertical suspension
V50	*5*	Linkage for horizontal suspension		
V60	*6*	Steering linkage		
V61			*1*	Rudder bar
V70	*7*	Transmission linkage		

Table.1. Naming convention for the different parts.

The first evident idea is to create a 2DOF suspension by serial connections (Fig.5):

– first a vertical joint from the hub of the wheel 120 to an intermediate glider 130;

– second a horizontal joint from the glider 130 to the frame 100.

Putting the vertical joint closer to the wheel allows to avoid that lower parts collide with the ground. In Fig.5, dampers are not represented and sphere 101 is a lumped mass connected to the frame 100.

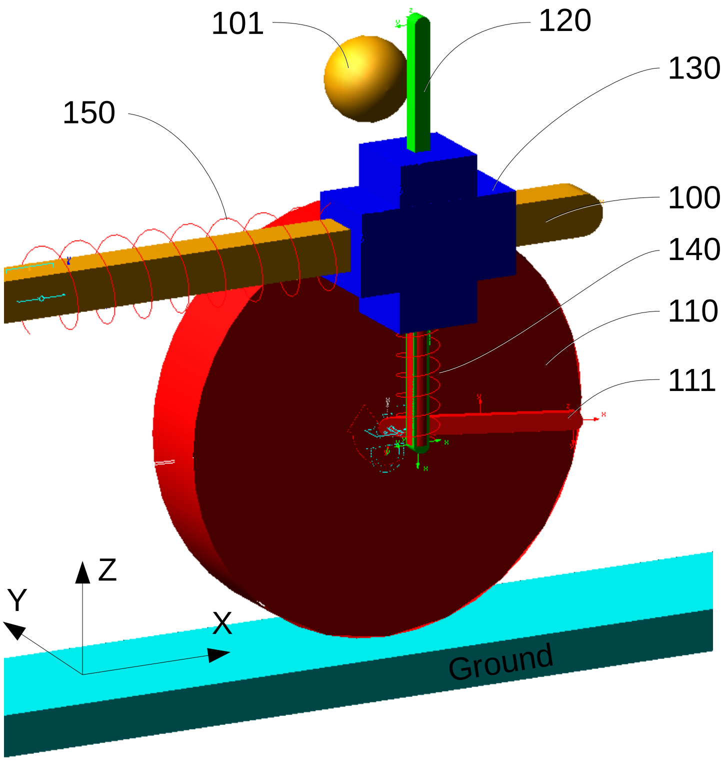

Fig.5. Version V1 of a 2DOF XZ suspension mechanism: 2D serial linkage [1].

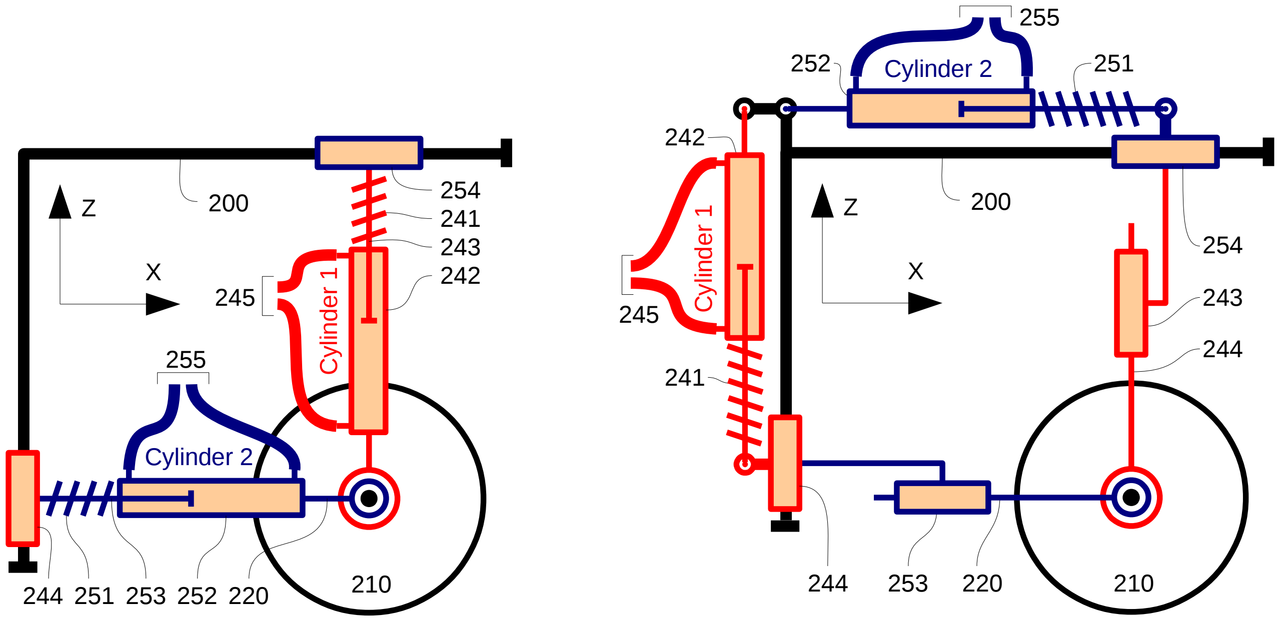

Fig.6. V2: (a) 2D parallel linkage; (b) Variant of V2 with cylinders attached to the frame.

Version V2 of the suspension (Fig.6) has a 2D parallel architecture and two limbs copying the serial structure of V1. The parallel architecture introduces a redundancy with respect to the serial architecture. Although the mobility of wheel 210 is the same, mechanism strength will be improved in case of shocks. For example in Fig.6a, horizontal shocks forces will be absorbed mostly by the compression of cylinder 2, instead of generating flexion of rod 243. In a variant of suspension V2 (Fig.6b), only passive joints are used in the limbs and the shock absorber are attached directly to the frame. This allows to decrease the non-suspended mass and provides better shock absorption.

Dual effect cylinders are represented in Fig.6 to show the possibility to actuate the suspension through hydraulic or pneumatic hoses 245 and 255. One interesting property of this parallel architecture is that it maximally regular. This means the Jacobian matrix is a unit matrix [9] and that pure wheel translation of distance d along the X or Z direction is achieved by actuating only Cylinder 2 or 1 of distance d respectively. On the contrary, the solution presented in Fig.4d has a coupled behaviour. A pure translation requires coupled control of several actuators. The cylinders of Fig.6b can also be replaced by passive dampers, possibly with adjustable damping coefficient. One should also note that the hub carrier 220 may be attached to body 252 or 242 with no significant influence.

Version V3 of the suspension (Fig.7) has also a parallel architecture but with coupled motions. This drawback is compensated by the elimination of the flexion moment around Y axis. Parts 342, 343, 352, 353 only bear traction-compression. The prismatic joints 244 and 254 in V2 are replaced by revolute joints 344 and 354 in V3. This will allow part downsizing for the same resistance.

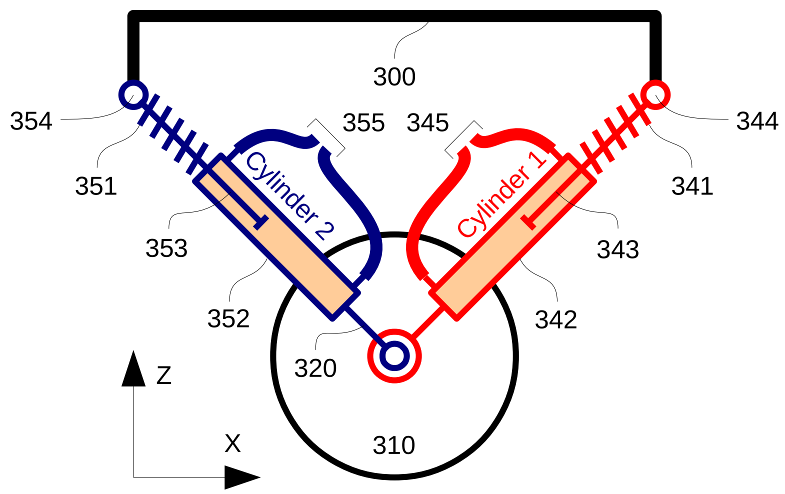

Fig.7. V3: 2D coupled 2(RPR) parallel mechanism.

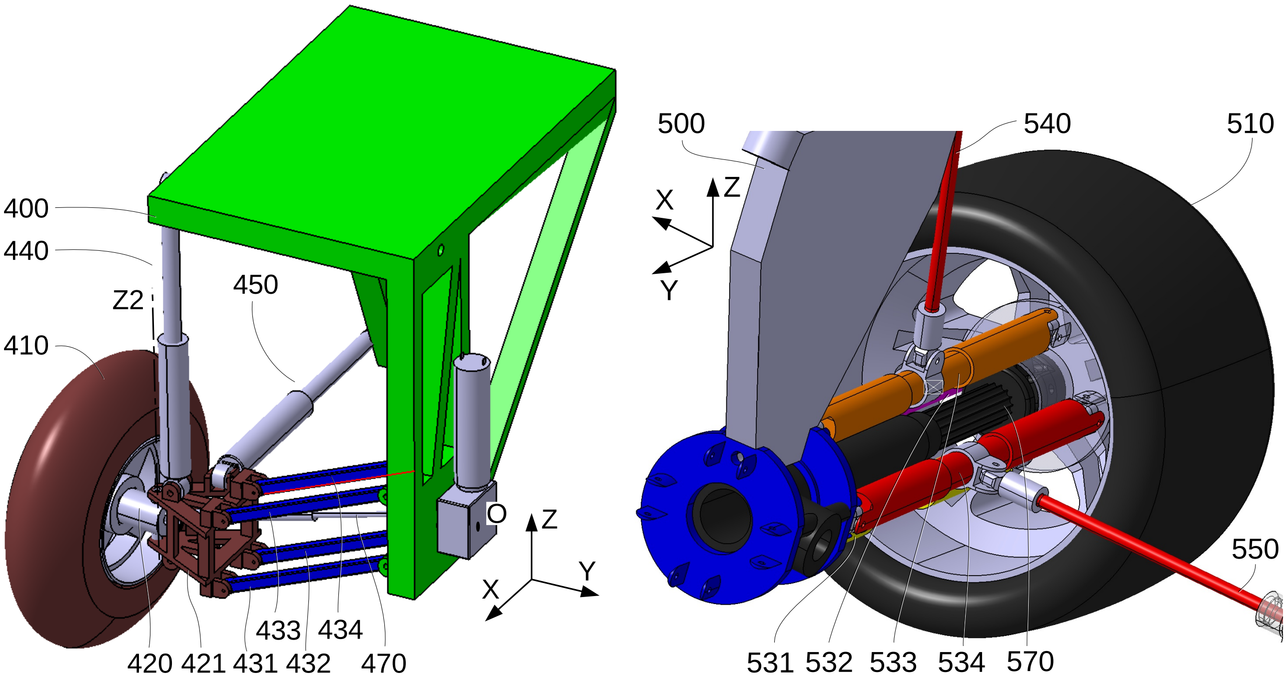

Fig.8. (a) V4: Spatial hybrid implementation [10] with four U-U parallel limbs for spherical translational suspension motions, locally uncoupled behaviour and steering capacity. (b) V5: with rhomboid layout of parallel limbs (without representation of steering device).

3D structural synthesis

All-terrain suspensions are subject to severe shocks, and the V2 solution is not acceptable because of flexion that would damage the gliders 244 and 254 (Fig. 6). This is why the 2D solution V3 (Fig. 7) will serve as the basis for improved 3D parallel architecture suspensions V4 (Fig. 8a, *[10]*). However, V3 suffers from several problems, that require solution in V4:

– *Lack of lateral stiffness*, which is critical during turns. Four parallel *U-U* limbs 431-434 were attached to guide the travelling platform 421 in a spherical translation. *U* joints are universal joints (two consecutive revolute joints with perpendicular crossing axes).

– *No steering capacity*. The hub-carrier 420 is now capable to steer thanks to a revolute joint around *Z2* direction between 420 and 421. The steering actuation-linkage is not represented.

– *Coupled motions*. The main axes of the cylinders 440 and 450 were rotated -45° around *Y* to become parallel to *Z* and *X* respectively. This setting allows a maximally regular behaviour ONLY in the central point of the workspace, where the links 431-434 are parallel to *Y* axis. Far from this position, extra-diagonal terms appear in the Jacobian matrix.

– *Variation of pitch angle* of the wheel hub 320 was important during suspension compression in V3. It was nullified in V4 by the use of *U* joints that resist to torsion around *Y*, contrary to *S* joints.

If a cross-section of suspension V4 is made in a plane normal to *Y* and cutting the four parallel bars 431-434, the four sections of the bars are located in a square layout. The attachments for the vertical and horizontal damping linkages 440 and 450 can be made directly to the travelling plate 421. However, most cars have deep recessed tyre-rims that prevent direct attachment to 421, unless curved bars are used, that would be submitted to intense flexion during shocks.

Version V5 of the suspension is represented in Fig. 8b, without any steering device, to extrapolate from V4. In the same cross-sectional view as previously, bars 531-534 are located in a rhomboid layout, resulting from a 45° rotation applied to bars 431-434 around *Y* axis. This brings bar 533 on top and bar 534 on the rear of the suspension, providing convenient attachment points for the vertical and horizontal damping linkages 540 and 550, even though the hub-carrier is hidden inside the wheel-rim.

Versions V4 and V5 use a lateral guidance linkage using four identical *U-U* limbs that allow a spherical translation of the wheel. Kinematically, any configuration of *N* parallel limbs with $N \geq 3$ may be used, although simplicity dictates to chose $N=3$ for solutions V6 to V9.

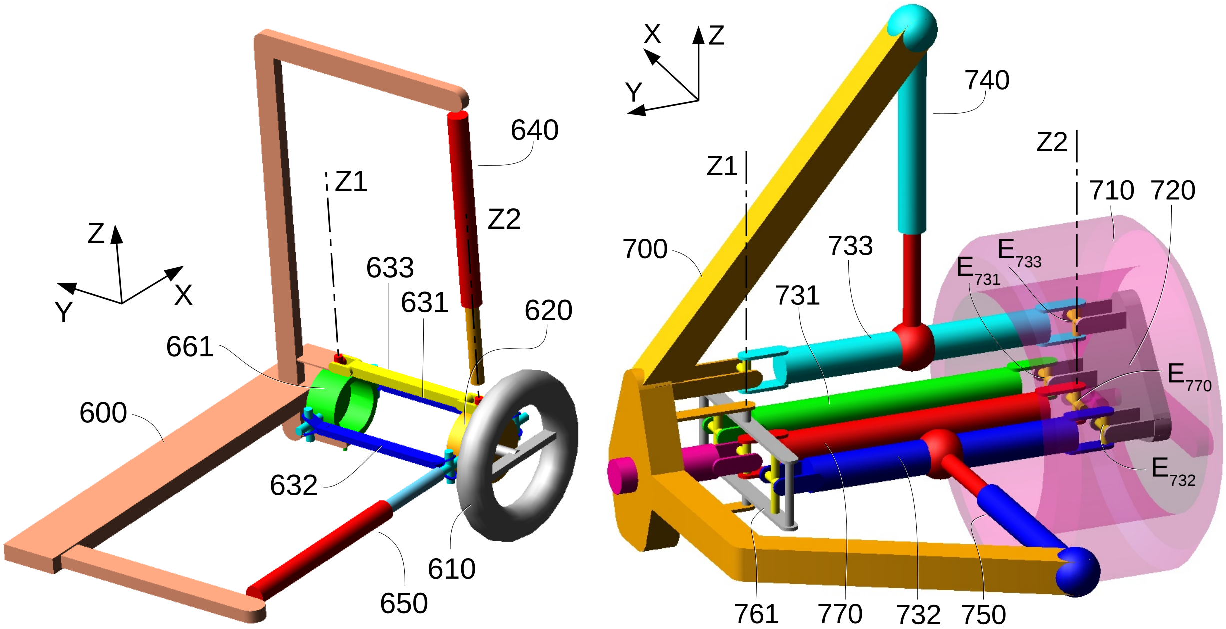

Fig.9. (a) V6: Parallel mechanism with 3DOF integrating the steering. (b) V7: the attachment points of the vertical and horizontal linkages 740 and 750 avoid any collision.

Solution V6 (Fig. 9a) offers a better integration of the steering linkage as it re-uses bars from the lateral guidance linkage. The two lower bars 631 and 632 are connected by *U* joints to a rudder-bar 661 on one side and to the hub-carrier 620 on the other side. Rotating 661 around *Z1* axis generates differential traction in 631 and 632 and makes 620 rotate around *Z2* axis of the same angle if bars 631-632-633 are parallel and of the same length.

Solution V7 (Fig. 9b) is an evolution of V6 where the attachment points for vertical and horizontal damping linkages 740 and 750 are located somewhere around the middle of bars 733 and 732 respectively. This layout avoids any collision between the tyre and the horizontal damper. Solution V7 also features a transmission line 770 with several shafts connected by *U* joints. The last important feature is about the steering axis *Z2* of wheel 710, that was designed to pass through the centre of the wheel contact patch with the ground in order to minimize the steering friction momentum. As *Z2* has to pass through E_{733}, end-point of bar 733, and also through the middle of E_{731} and E_{732}, the positioning constraint on *Z2* propagates to the positioning of bars 731-733.

Although V6 and V7 provide an interesting integrated steering system, they must deal with a disadvantage: a slight coupling remains between steering and horizontal damping motion.

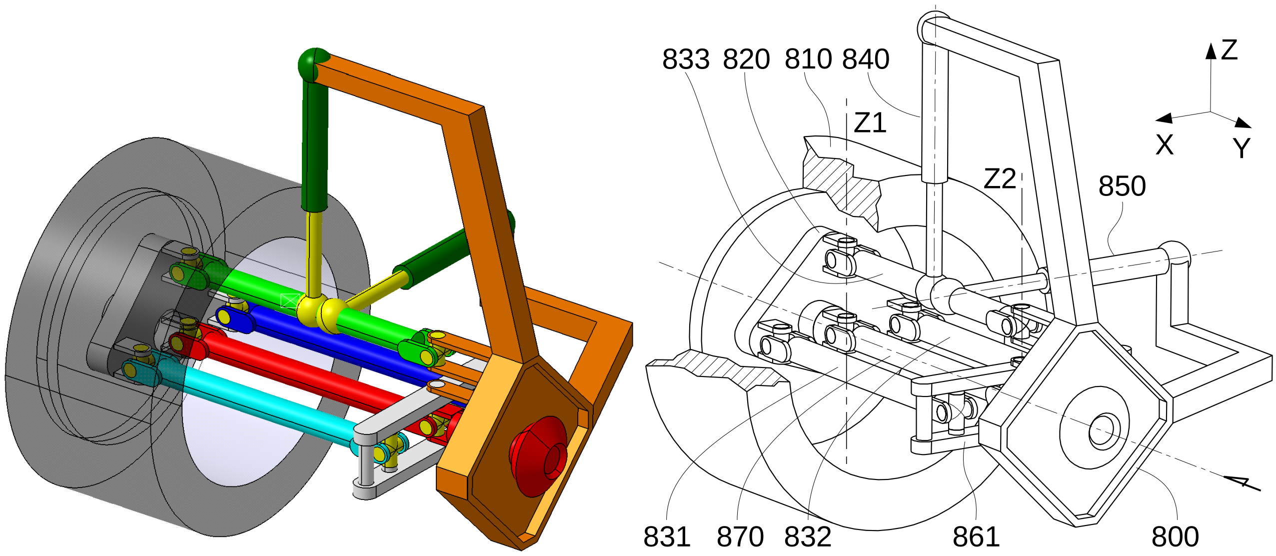

Fig.10. V8: Steering becomes decoupled from horizontal damping motion thanks to a change in the attachment point of 850 to the top bar 833.

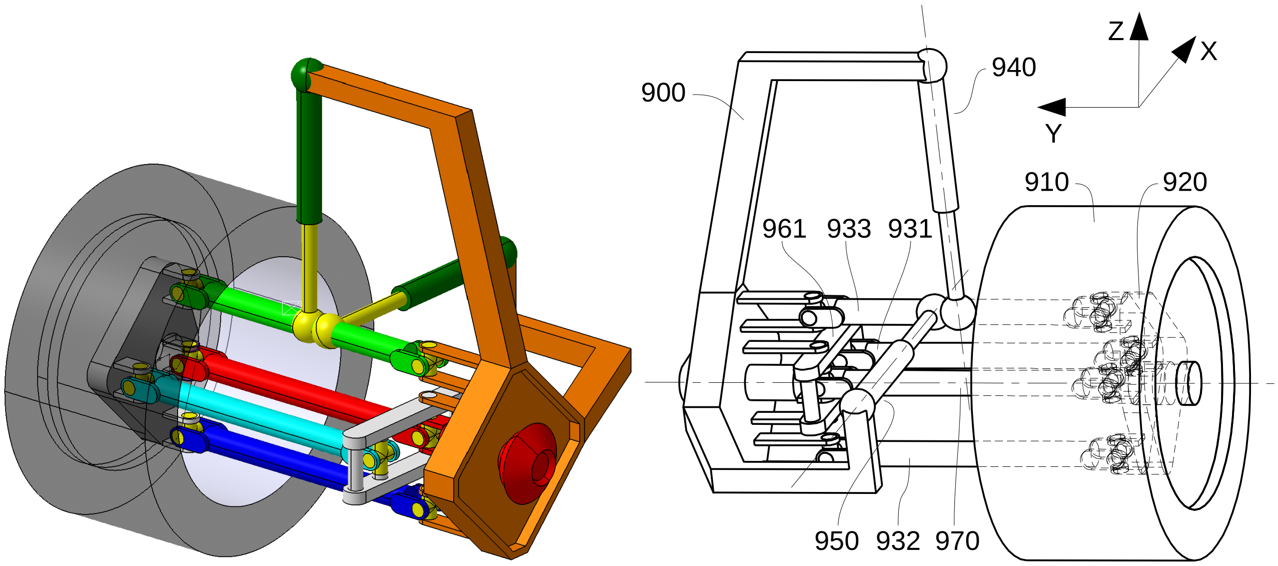

Fig.11. V9: Variation on V8. Steering is actuated by a single arm and two bars are directly connected to the frame. This structure is very close the double triangle architecture.

In solution V8 (Fig. 10), this coupling completely disappears as the attachment point of the longitudinal damping linkage 850 has been moved to a point that is not affected by steering: around the middle of the top bar 833, one of the few attainable points laying in the steering-invariant plane, which contains *Z1* and *Z2* axes. As the *S* joints that connect 840 and 850 to 833 cannot be easily merged, it was chosen to dissociate them enough to avoid any collision.

In solutions V6-V7-V8, the lower bar was eliminated among the four lateral guidance bars, mostly to clear out the space under the suspension. Solution V9 (Fig. 11) shows a variation on V8 where the rear bar was suppressed (it could be the front bar as well). The rudder-bar 961 now pulls only bar 931 to apply steering to the hub-carrier 920, whereas bars 932 and 933 are in charge of lateral guidance.

This choice is interesting as it is compatible with existing vehicles with double-wishbone suspension. Their frame already has upper and lower attachment points for the triangles that can be re-used here for V9 suspension.

Dimensional synthesis

In solutions V8 and V9, the top bar 833 or 933 has a critical role, as it transmits both the horizontal and vertical components of impact forces on the obstacle. Consequently, this bar is submitted to combined flexion momentums around *Z* and *X* and should be reinforced compared to 831-931 and 832-932, that undergo only pure traction and could be downsized.

Some dimensions result from a compromise in order to avoid interference between parts in different positions of the suspension. The distance between bars 831 and 832 should be as large as possible for better steering stiffness but is limited by the non-interference between the bars and the tyre-rim, particularly at extreme steering position and horizontal compression. It must also be large enough to let enough space for the transmission line 870, whatever the position. Another interference may occur between the internal part of the tyre and linkage 850, mostly at maximum steering. This is why 850 is attached to 833 at point slightly translated in *Y* direction.

Bars 831-833 should also be taken as long as possible for getting a larger spherical translation radius and for allowing an approximate planar motion of the wheel in plane *XZ*.

In the real implementation of suspension V8, a Traxxas E-Maxx radio-controlled all-terrain car of scale 1:8 was chosen. Re-using the double cylinder dampers required to refine the 850 linkage (Fig. 12). Cylinders 852 and 852' were rigidly paired to intermediate part 853, that was connected by a *U* joint (part 854) to car-body 800. At the other end, the damping assembly was connected to 833 by a *S* joint. The real implementation (Fig. 13) required some adjustments in the steering mechanism and also a stronger servomotor. The car was successfully tested.

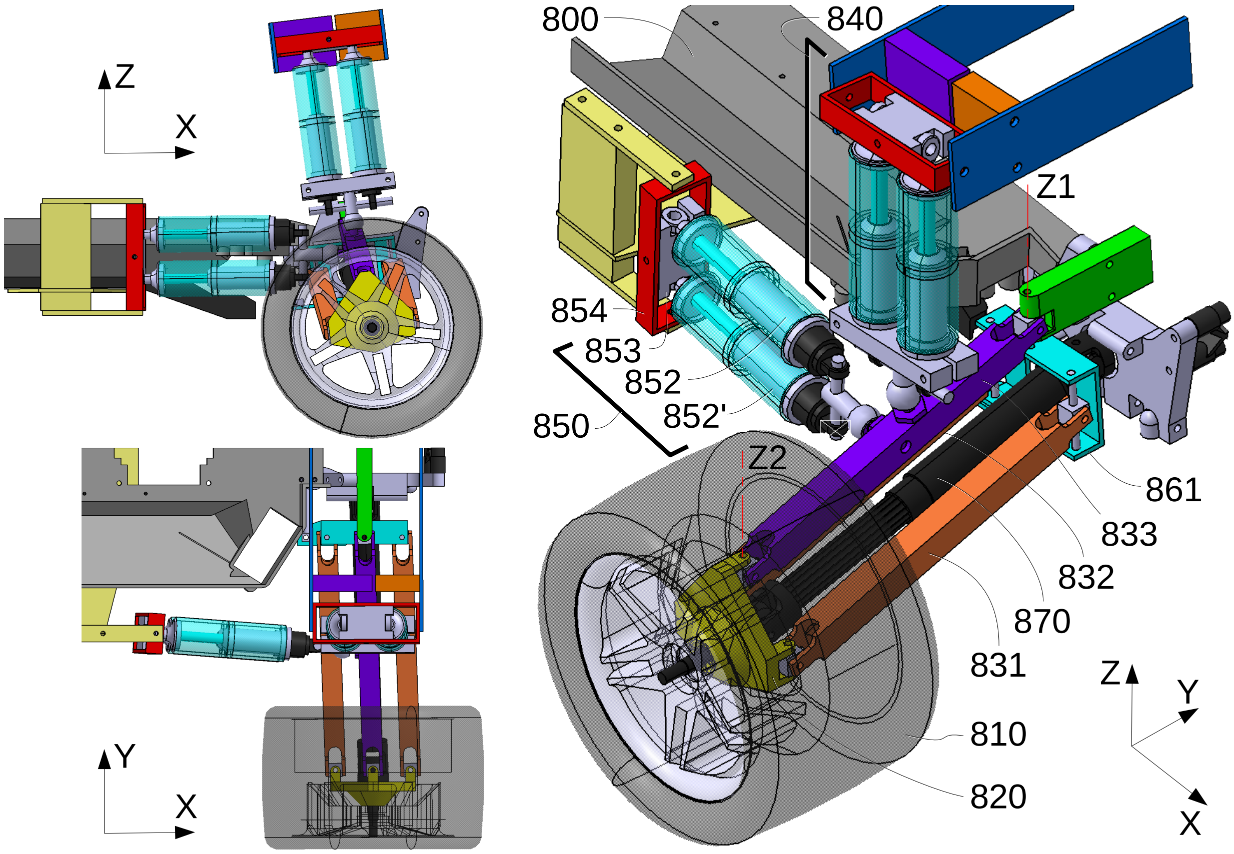

Fig.12. CAD model of V8 integrating real parts and double-damper sub-assemblies [11,12].

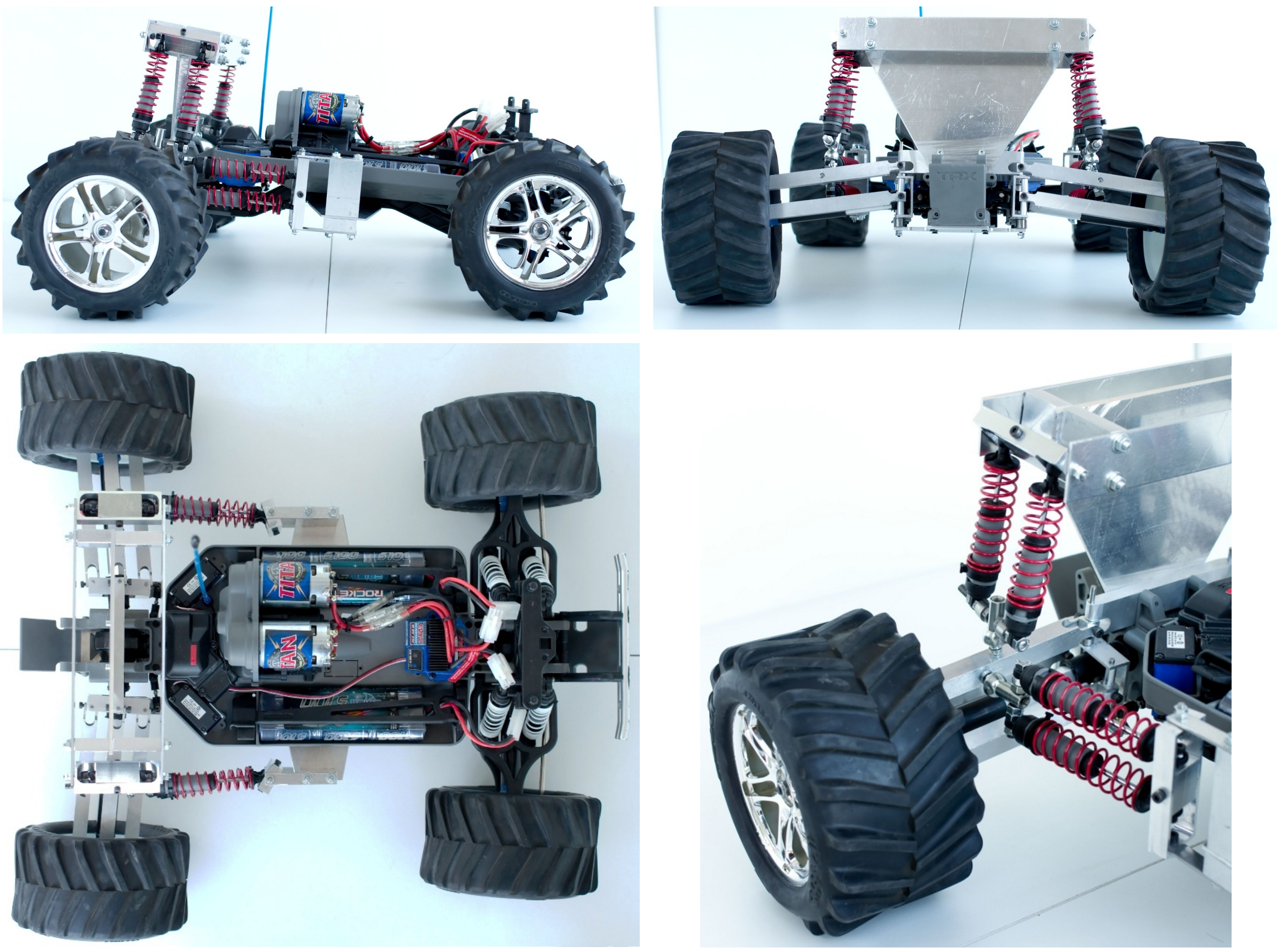

Fig.13. Mechanical implementation [11,12] of suspension V8 as the front suspension of a Traxxas E-Maxx 1:8 scale model.

Conclusion

This paper presented the structural & dimensional synthesis of nine suspension mechanisms designed to provide vertical and longitudinal all-terrain suspension. Eight solutions have parallel kinematics and six of them are spatial mechanisms. Most of them were patented in [13]. Solutions V8 & V9 integrate all the constraints and are capable to provide longitudinal & vertical motions of the wheel, with an uncoupled behaviour in the centre of the workspace. They also allow steering and power transmission. A new campaign of dynamic obstacle crossing experiments will reveal the achievable improvements from this 2DOF vertical & longitudinal suspensions with respect to classical 1DOF vertical suspensions and also will allow to design the associated control strategies.

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