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Development of an innovative test machine for tyre, wheel and suspension systems for automotive and industrial vehicles

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Abstract

Reliability and efficiency of automotive and industrial vehicles are strongly affected by the behaviour of the components involved in the interaction with the ground. For this reason, components like tyres, wheels and suspension systems are physically tested, for performance and durability assessment. Anyway, standard test machines are generally designed for testing only one specific component.

The aim of this work is the design of an innovative test machine, able to perform different kinds of tests, for tyre, wheel and suspension system components, on a unique bench. The machine is a flat track type, better fit than traditional drum types in simulating real tyre-to-ground contact conditions. The core of the machine is a Gough-Stewart platform, where the flat track is fixed, which is moved by six hydraulic actuators to reproduce multiple work configurations, for extended time periods or numerous test block repetitions. After the preliminary design phase, the machine components were subjected to topology optimization and modal analysis by FEM, to reduce weights and avoid any issues related to the expected working frequencies. Given the quality of the results achieved, currently, the software design for closed-loop control configuration is under development, for the machine prototype next construction.

Author Keywords. Fatigue, Test Machine, Wheel, Topology Optimisation, Modal Analysis.

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1. Introduction

Both in the automotive and off-highway industry, vehicles and their components are subjected to several tests before serial production release and, later, to guarantee that product quality standards are met. In particular, components involved in the interaction with road or ground, like tyres, wheels or suspension systems are very important because they have great impact on vehicles efficiency and durability. For this reason, for many years, specific machines for testing these components in the laboratory have been designed and produced both by research groups or institutes (Grubisic and Fischer 1983; Cabrera et al. 2003) and industrial manufacturers (Langer and Potts 1980). For example, dynamic tests are done on tyres and rims to check both tyre performance and rim fatigue strength, sometimes in combination with numerical models that simulate the effect of test loads on the specific components (Cheng et al. 2020; Ballo et al. 2020). Two different kinds of machine constructions are generally used for that: drum type machine and flat track type machine. The former is based on a rotating steel drum, where wheels are pushed against, internally or externally, with a certain load and angle (**Figure 1(a-b)**). The load is usually generated through one or two hydraulic actuators in order to create, respectively, only radial or radial and lateral loads (biaxial test machine (Grubisic and Fischer 1983)). The latter is based on a flat track system, usually consisting in a closed-loop flexible belt, driven by two smaller rotating drums, to form a flat surface on which the wheel is free to rotate at an imposed load (**Figure 1(c-d)**). In industry and research labs, the drum type machine is much more common, due to higher simplicity of construction and lower associated costs. However, the curvature of the contact surface of the steel drum is not always the best option to simulate real contact and pressure conditions between tyres and ground. For that reason, in test standards for specific components, a minimum drum diameter is recommended in relation to the tyre diameter under test load, to minimise this issue (see for example standards from ('EUWA – Association of European Wheel Manufacturers' 2022) or (ISO 28850:2018 2018)). Nonetheless, flat track type machines are still preferable in tests where tyre-to-ground contact must be as real as possible.

Other components are tested on different machines, such as cornering test machines for wheel discs or specifically designed machines for suspension systems and shock absorbers. Also, in the automotive industry, more complex customised machines have been developed by industrial systems manufacturers for full scale testing of vehicles, recreating the load history of the vehicles in service through the action of actuators linked to the wheel or hub parts (Ramirez Ruiz et al. 2012). Since the understanding of real working conditions on components is fundamental to replicate proper tests in labs, field tests are more and more common, historically in the automotive market (see for example (Sonsino et al. 2021)), and now also in the off-highway market (Cima and Solazzi 2021; Mazzoni and Solazzi 2022; Stellmach et al. 2021; Marques, Solazzi, and Benasciutti 2022). During field tests, devices like wheel force transducers (WFT), or data acquisition systems (DAQ) connected to sensors, are used to measure, respectively, forces and moments, or stress and strain histories acting on components in real service conditions. Although field tests are very useful and, sometimes, necessary to understand operating conditions, they are often quite expensive and difficult to do, because of the limited availability of vehicles for testing purposes, in terms of time and variety of tracks.

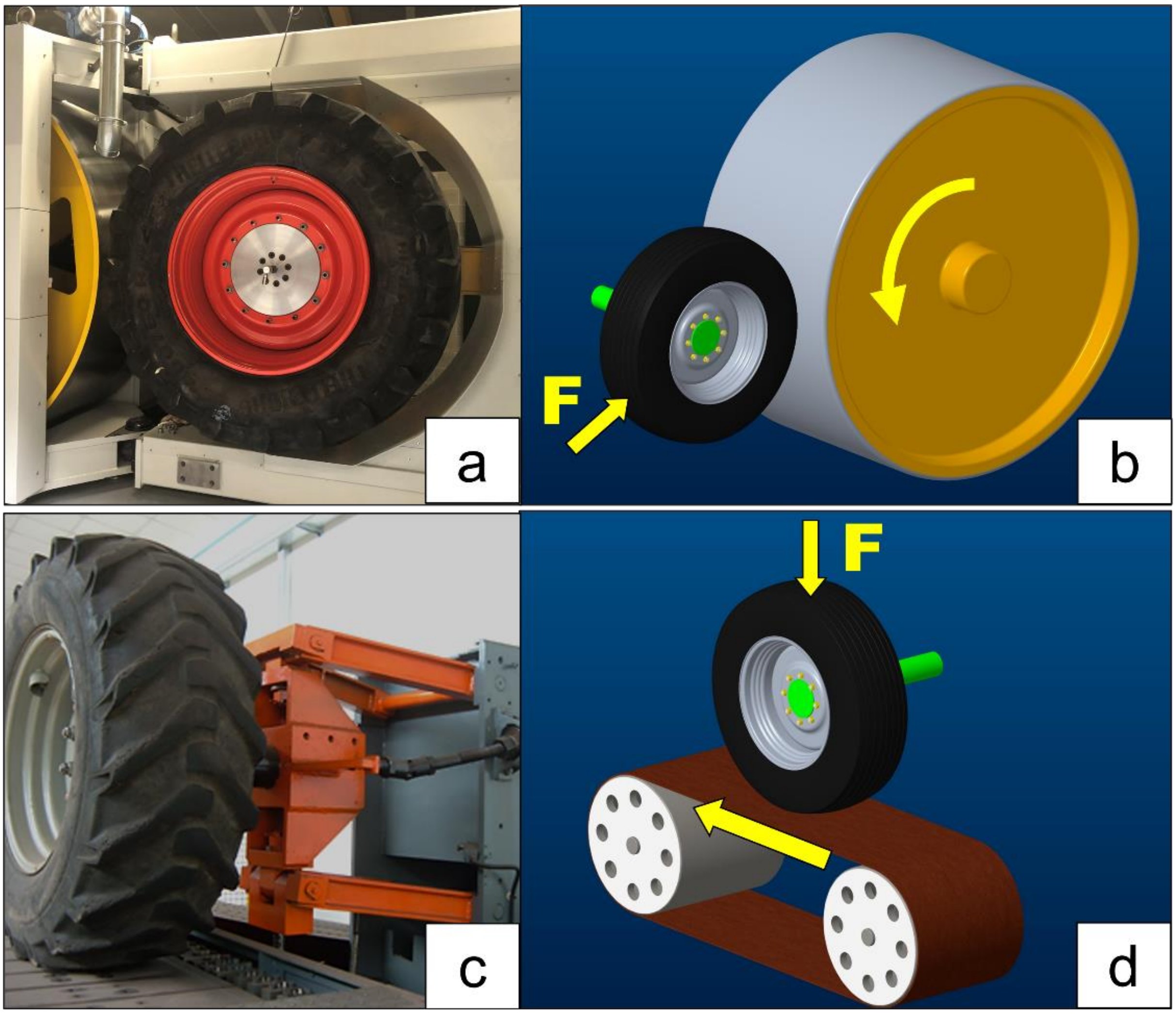

Figure 1: Example of machines for tyre and rim test: drum type machine (a, with simplified working scheme b) and flat track type machine (c, with simplified working scheme d)

This paper presents the design of an innovative multipurpose test machine, able to perform different kinds of tests on the main components involved in the interaction between road and vehicles, such as tyres, wheels and suspension systems. The machine is based on a flat track system mounted on a hexapod table, also known as Gough-Stewart platform, which has six degrees of freedom, thus giving the possibility of recreating all the possible load directions acting on the components. The load itself is generated by a hydraulic press, with a proper frame for the vertical guide, and transmitted to the tyre and wheel assembly through a suspension system. The degrees of freedom of the hexapod table can be used to impose variable slip or camber angles, to simulate real working conditions or to recreate load histories measured in the field and repeat them for a longer period or a defined number of test blocks. The machine is completed with sensors to control functioning parameters and acquire test data. A cooling system to avoid tyre overheating is also present, together with proper safety protections, in view of the possible machine construction. The next step of the design process will be the development of the software for the motion control of the table.

2. Machine description

The machine concept is represented in **Figure 2**.

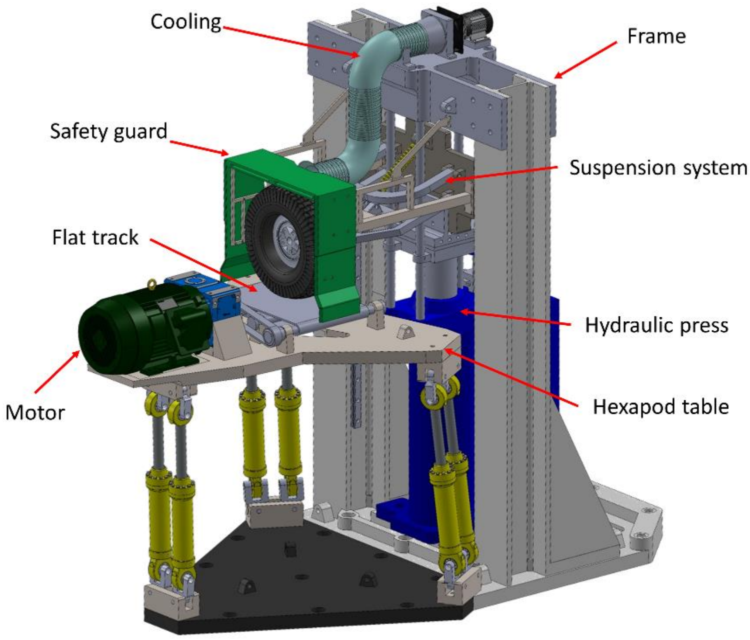

Figure 2: 3D model of the test machine

It is made of two main parts: a hydraulic press and a hexapod table. The hexapod table, also named as Gough-Stewart platform, is the core of the machine, since, thanks to its six degrees of freedom, it allows a wide range of movements and rotations to recreate many possible contact configurations between tyres and road. The Gough-Stewart platform is a mechanical system that works as a parallel robot and is formed by six linear actuators, paired together with both the machine base and the upper platform through prismatic joints between axial ball joints. The combined action of the actuators gives to the system the possibility to sustain high loads in relation to its overall mass, with a very low rate of bending loads, which is important for the precision of the movements. For these characteristics, similar systems are widely used in flight simulators. Additional applications and studies can be found, for example, in (Qazani, Pedrammehr, and Nategh 2018; Butzhammer, Müller, and Hausotte 2023; Pedrammehr et al. 2014; Friedrich and Ihlenfeldt 2022).

On the top of the platform, the flat track system is mounted. It is moved by a 45 kW electric motor, coupled with a proper reducer, which allows an equivalent speed range from 50 km/h to 130 km/h for the wheel assembly.

The vertical load on the tyre-wheel assembly is produced by the hydraulic press, which is designed to achieve 20 kN maximum load. However, the platform actuators are designed to sustain a maximum load of 25 kN, which also considers the weight of the above components. The press is fixed to a proper structure, with a base made of ductile iron EN-GJS-400-15 (EN 1563) and two commercial steel beams of type HEB (DIN EN 10365), which work as vertical guides for the sliding plate that allows the transmission of the load from the press to the suspension system. This is another important part of the machine and it is represented in **Figure 3**. Simulating a quarter of a car suspension system, it is designed as a double wishbone scheme, where both arms and shock absorber are directly connected to the sliding plate for load transmission. If needed, customised suspension systems could be adapted to the machine for specific tests on their components, thus widening the versatility of the machine. A proper design of this system is also important for testing the tyre-wheel assembly, since the load acting on these components is generated by the reaction force in the contact between the tyre and the flat track system. Therefore, the load transmission between the press and the tyre-to-track contact area must be as realistic as possible.

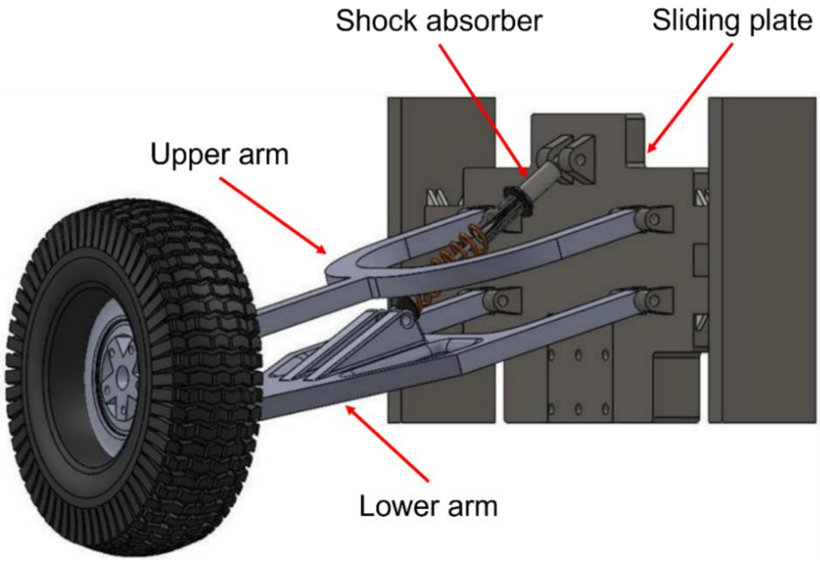

Figure 3: 3D model of the suspension system

To guarantee the correct functioning of the machine, a series of sensors is needed. They are used both for systems control and data acquisition. In particular, temperature sensors are used to monitor the tyre temperature during the test, with a double purpose: on the one hand, to guarantee a tyre temperature range consistent with typical values of real usage, on the other hand, to avoid rubber overheating that could lead to premature tyre damage or even explosion, with related problems for safety. A tyre cooling system is present to reduce that risk. As temperature sensors, infrared pyrometers are chosen. They are mounted on the fixed structure of the machine, to avoid any contact with moving parts, and focus on the tyre surface to check its temperature continuously. Potentiometers are installed on the shock absorber and on the hexapod table actuators for feedback control of the system, while a series of strain gauges must be applied to the structural components, like the HEB beams and the sliding plate, to monitor their state of stress and strain and assure that it remains inside the design range. Finally, rotary encoders are used on the motor shaft, on the track rollers and wheel hub, to measure their speed and control possible slippage phenomena of the track belt and any related motor issues. Linear encoders, instead, are used on the hexapod table actuators to check their relative position and guarantee the correct functioning of the control system responsible for the table movement. The positions where strain gauges, rotary encoders and linear encoders will be installed are indicated in **Figure 4**.

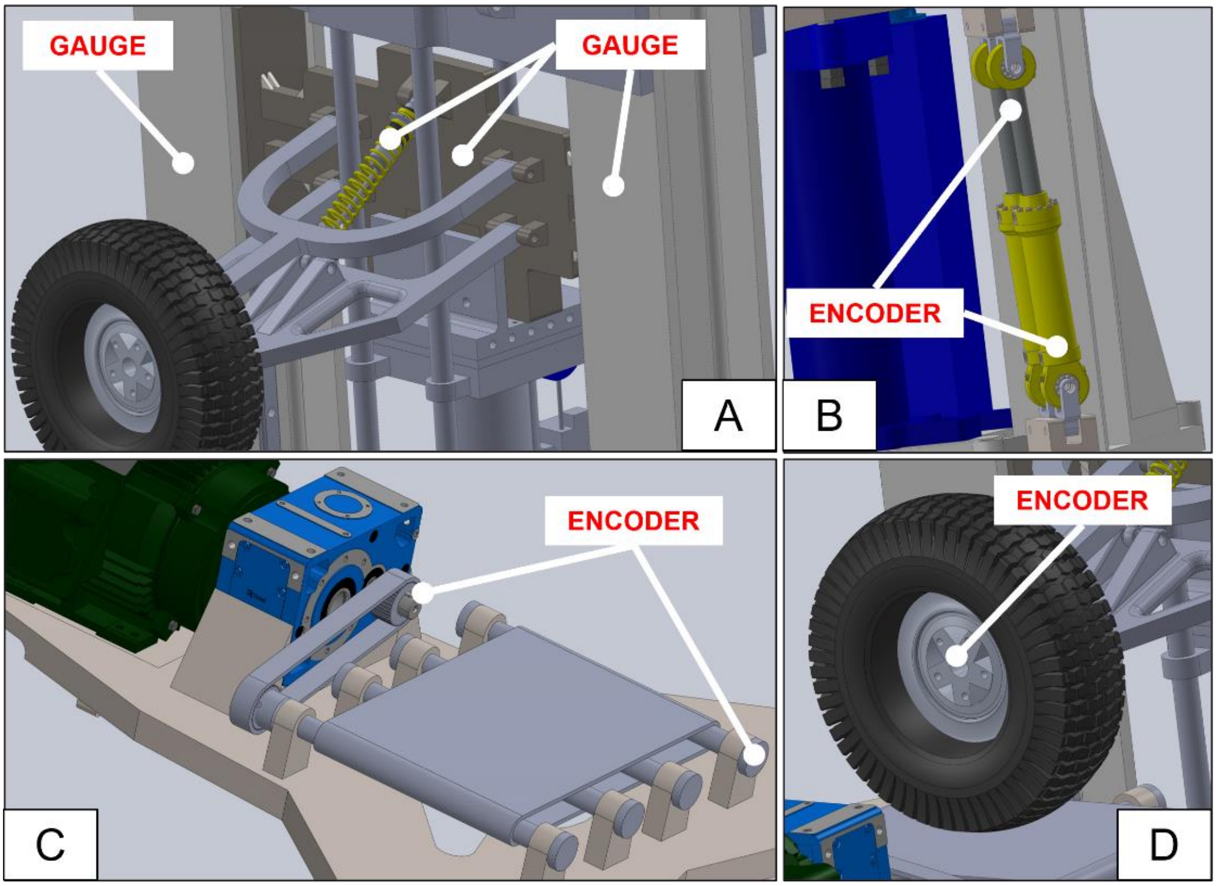

Figure 4: Positions of strain gauges (A), linear encoders (B) and rotary encoders (C-D)

Details of machine design methodology and calculation are discussed in section 3, while the general machine specifications are reported in **Table 1**. The materials used for the structural components of the machine, which will be mentioned in the following sections, are summarised in **Table 2.**

Tyre outer diameter range	350 ÷ 1000 mm
Tyre and wheel nominal diameter range	from 12” to 22.5”
Maximum tyre width	440 mm
Equivalent road speed	50 ÷ 130 km/h
Maximum vertical load on wheel	20 kN
Maximum lateral load allowed	15 kN
Overall dimensions (LxWxH)	3380 x 1700 x 2900 mm
Overall weight (approx.)	12000 kg

Table 1: General machine specifications

Material	Young Modulus [MPa]	Mass Density [kg/m^3]	Tensile Strength [MPa]	Yield Strength [MPa]	Component
EN-GJS-400-15 (EN 1563)	120,000	7,200	400	250	Frame base
S275J0 (EN 10025)	210,000	7,850	410 to 560	275	Frame beams
S460J0 (EN 10025)	210,000	7,850	550 to 720	460	Hexapod table, sliding plate

Table 2: Material specifications of the structural components of the machine

3. Machine design

3.1. Preliminary study

The design of the machine started from a preliminary analysis of the layout, based on the main components, namely the hexapod table, the hydraulic press and the frame, mainly made up of the base and two vertical HEB beams. For cost related reasons, the actuators of the table will be chosen from commercial products available on the market and already suitable for connection with hardware and software devices. However, these actuators are designed to bear a total axial force of 25kN, with maximum rod stroke of 250 mm at 200 bar working pressure. The total force considers the maximum vertical load that can be produced by the hydraulic press and the weight of the upper table, with its main components (electric motor and flat track system). Design verifications were done following standard analytical methods to choose the actuators size, preventing possible buckling occurrence.

The preliminary dimensioning of frame, hexapod table and sliding plate (interface between press and suspension system) was verified by FEA and the main results are shown in sections 3.1.1, 3.1.2 and 3.1.3. General design target was keeping the maximum deflection of the structures below 1.5 mm, with maximum stress value one order of magnitude below the yield stress of the materials, in operating conditions. Then, a modal analysis was conducted both on the main components and the complete machine, to verify that the natural frequencies of the system would not create resonance issues in relation to the working frequency range. Afterwards, topology optimisation was done on the most massive components, in order to reduce the overall weight of the machine with negligible change on its performance. Finally, a final verification of the stress state and the natural frequencies of the structure, after topology optimisation, was done to finalize the project. Details of topology optimisation and final modal analysis are discussed in section 3.2 and 3.3 respectively.

SolidWorks Simulation software has been used throughout the design phases described above.

3.1.1. Design verification of the machine frame

The frame of the machine is made up of a ductile iron base, where two HEB steel beams are welded. Two load cases have been verified, in relation to the vertical position of the sliding plate for the transmission of the force between the press and the wheel assembly:

1. 800 mm distance of the sliding plate from the top section of the HEB beams.
2. 1000 mm distance of the sliding plate from the top section of the HEB beams.

Both in load case 1 and 2, a vertical load of 25 kN is applied to the frame base, where the press is fixed, and a lateral load of 25 kN is applied to HEB beams, thus considering the unlikely extreme condition of wheel assembly close to the horizontal position and pushing on the beams with the total reaction force. FEA results are shown in **Figure 5** and **Table 3**, both for load case 1 and load case 2.

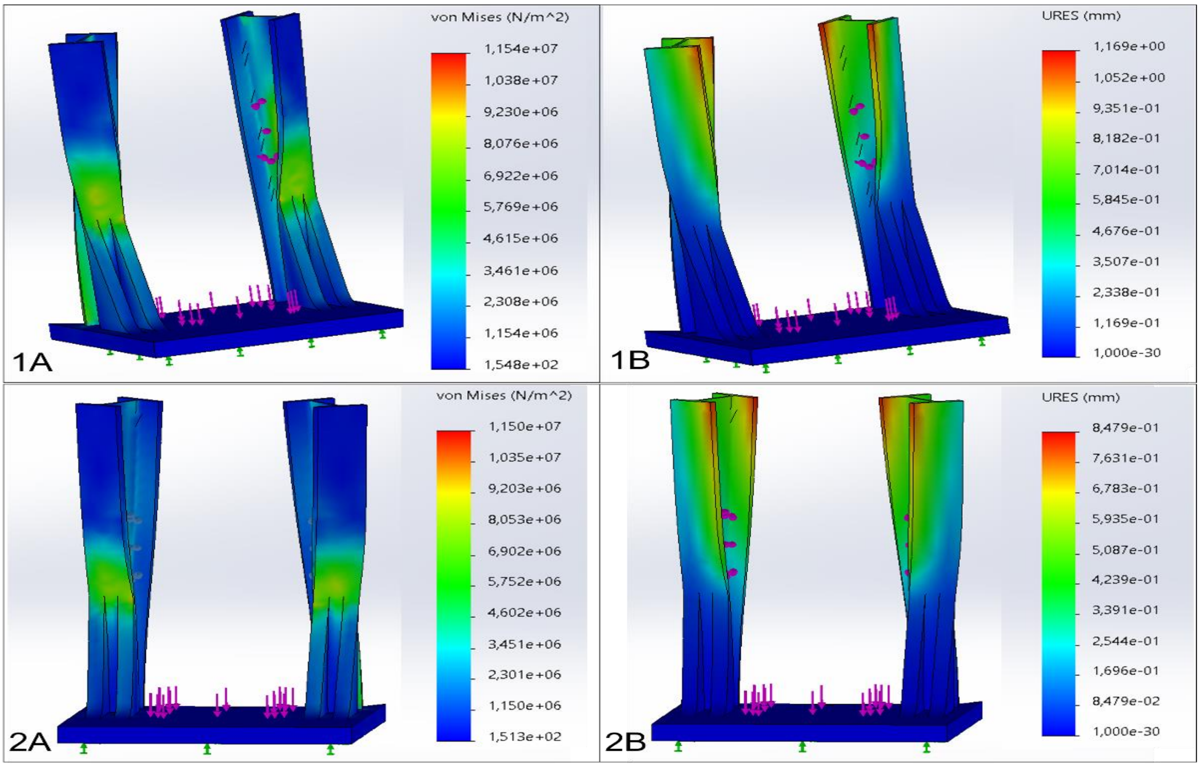

Figure 5: FEA results for the machine frame: von Mises stress (1A) and maximum displacement (1B) for load case 1 and load case 2 (2A-2B)

Load Case	von Mises Maximum Stress [MPa]	Maximum Displacement [mm]
1	11.6	1.17
2	11.5	0.85

Table 3: FEA results for the machine frame

Values in **Table 3** are in line with design targets, since von Mises stress is well below the yield stress of the beams material (275 MPa for steel S275J0) and the maximum displacement is around 1 mm in the worst theoretical condition, unlikely to occur in real use.

3.1.2. Design verification of the hexapod table (Gough-Stewart platform)

The hexapod table is the core of the machine. Its design needs to meet the target of proper stiffness, for the loads and accelerations involved, together with low weight to reduce the power required to the actuators and allow energy saving. The material chosen for its production is steel S460J0, with minimum yield strength of 460 MPa.

Also for the hexapod table verification, two load cases have been analysed:

1. Table in horizontal position, loaded with one vertical force F_1 = 20 kN, to simulate the maximum load condition, and an additional vertical force F_2 = 5 kN, to simulate the weight of the electric motor and connected parts.
2. Table positioned at 45° angle respect to the horizontal plane and subjected both to the vertical forces of load case 1 and to a lateral force F_{1L} = 15 kN, which is due to the inclination of the table. The same value of the vertical forces, as per load case 1, was kept just as an extra safety factor in the calculations. The lateral component of F_2, F_{2L}, has low influence both on stress and displacement, because it acts in the midplane of the table, which is designed to be properly stiff.

A simplified scheme of both load cases is represented in **Figure 6**, while FEA images are shown in **Figure 7**. Detailed results are shown in **Table 4**.

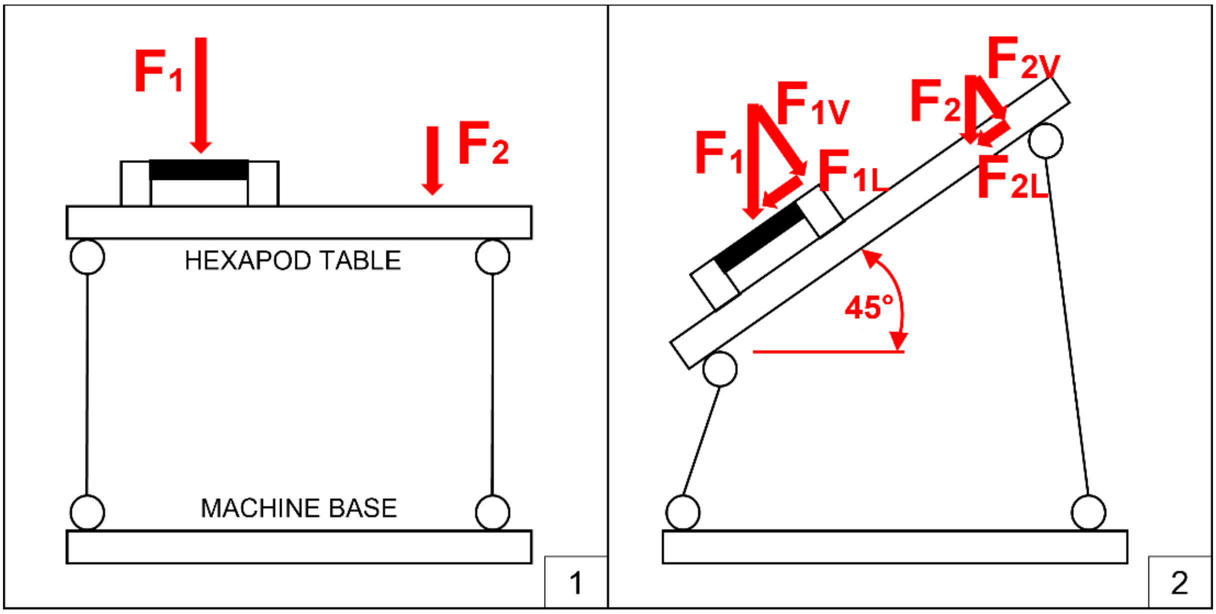

Figure 6: Simplified scheme of load case 1 and load case 2 for the hexapod table. F_1 is the vertical force due to the press load, F_2 is the vertical force due the motor weight

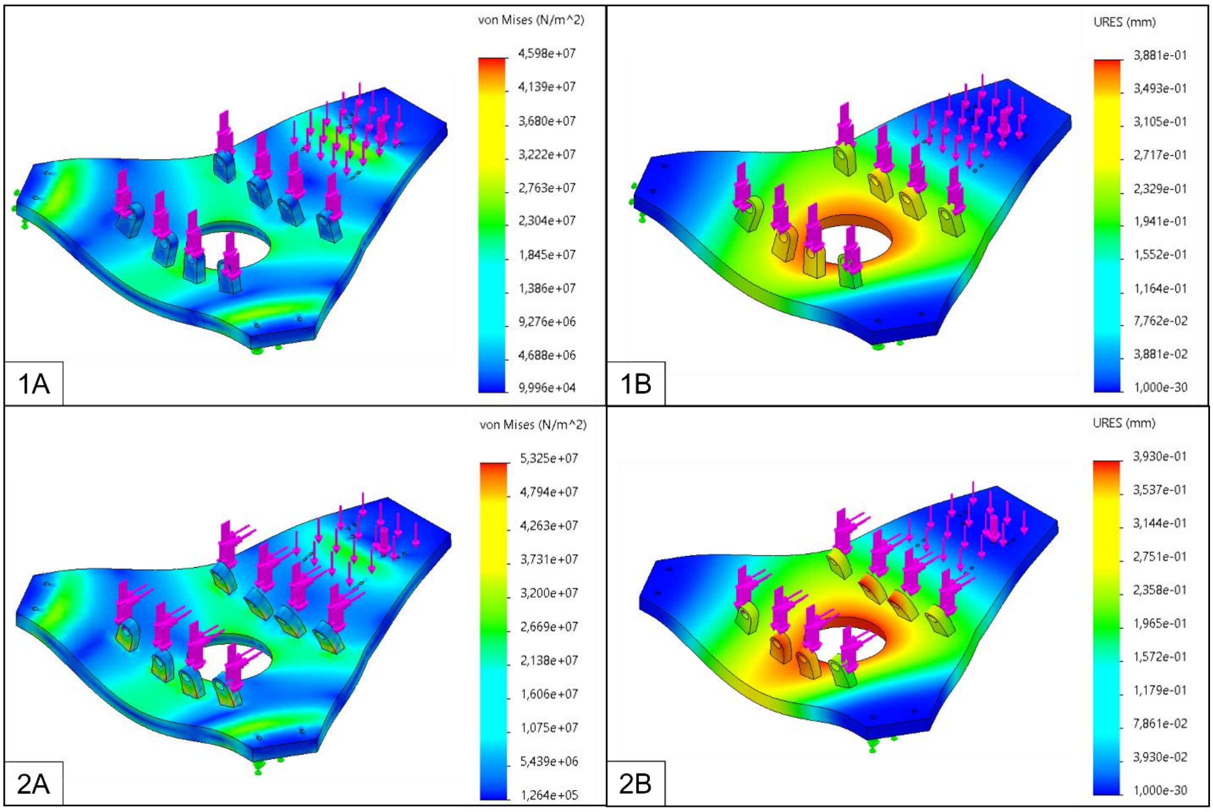

Figure 7: FEA results for the hexapod table: von Mises stress and maximum displacement for load case 1 (1A-1B) and load case 2 (2A-2B). Maximum full scale values: 53.3 MPa for stress (2A) and 0.39 mm for displacement (2B)

Load Case	von Mises Maximum Stress [MPa]	Maximum Displacement [mm]
1	46.0	0.38
2	53.3	0.39

Table 4: FEA results for the hexapod table

Values in **Table 4** are in line with design targets, and the additional lateral force, which is applied in load case 2, has limited effect both on stress and displacement.

3.1.3. Design verification of the sliding plate for load transmission between hydraulic press and suspension system

The sliding plate that allows the load transmission between the hydraulic press and the suspension system is one of the most massive components of the machine. The material chosen for its production is the same of the hexapod table, steel S460J0, with minimum yield strength of 460 MPa. The first step of its design led to a simple and approximate shape, which was subsequently revised and improved through topology optimisation. This second design step is discussed in detail in section 3.2.3, with related changes in the geometry of the component shown in **Figure 10**. Instead, FEA results of **Table 5** are related to the preliminary shape and dimensions. Only the most severe load condition was analysed, which considers the reaction forces acting on the plate both along vertical direction and horizontal direction, due to 45° inclination of the hexapod table as per load case 2 of section 3.1.2.

von Mises Maximum Stress [MPa]	Maximum Displacement [mm]
82.6	0.10

Table 5: FEA results for the sliding plate

According to the values in **Table 5**, the design target is met also for the sliding plate, with safety factor more than 5, in reference to the yield stress of the material.

3.2. Topology optimisation

After the preliminary study, the design of the machine was refined through a process of topology optimisation, by means of SolidWorks Simulation software. The analysis focussed on the three most massive components presented in the previous sections of this paper: the machine frame, the hexapod table and the sliding plate for load transmission. As a final result of the topology optimisation, a theoretical overall weight reduction of 1971 kg was achieved, with negligible effects on the mechanical strength and functionality of the machine. In the following sections, details of this design phase are shown.

3.2.1. Topology optimisation of the machine frame

The frame is the most massive part of the machine. It is important both for the mechanical strength and for the general stiffness of the machine. The target of its topology optimisation was to maximise the ratio between strength and weight, keeping, as a constraint, the maximum allowable displacement below 1.5 mm. For structural reasons, the HEB beams were not object of any design change. The modifications for weight reduction were limited to the base and are shown in **Figure 8**, where the geometry of the component before and after the topology optimisation is represented. As a final result of the analysis, a theoretical weight reduction of 1496 kg is achieved, corresponding to 30% of the initial value. A further FEA, done on the optimised shape, shows no major effects on the maximum stress and displacement of the structure, in comparison with the preliminary study of section 3.1.1, as presented in **Table 6**. Only the worst load condition is verified.

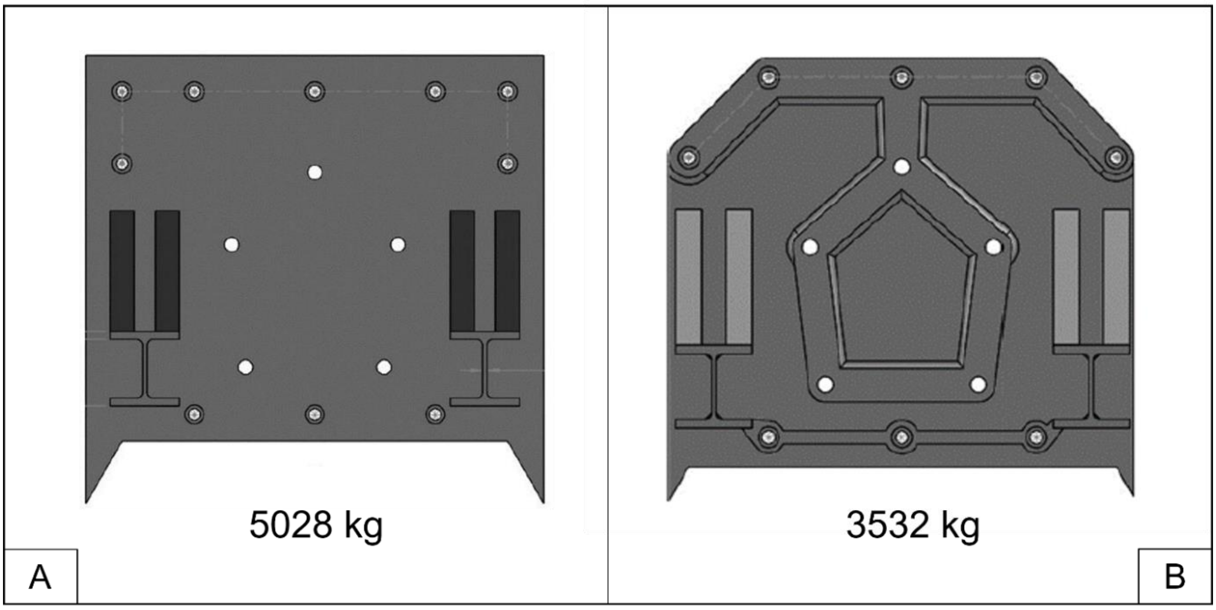

Figure 8: Geometry of the machine base: preliminary design (A) and after topology optimisation (B)

Design phase	von Mises Maximum Stress [MPa]	Maximum Displacement [mm]
Preliminary	11.6	1.17
After optimisation	90.3	1.45

Table 6: FEA results comparison for the machine frame before and after topology optimisation

The expected increment in the maximum von Mises stress of the frame is still acceptable, considering that the safety factor is of around 3, with reference to the minimum yield stress of the material (250 MPa for ductile iron EN-GJS-400-15). The maximum displacement is also acceptable, since the load case verified is quite severe, in reference to the expected working conditions of the machine.

3.2.2. Topology optimisation of the hexapod table (Gough-Stewart platform)

The hexapod table has a key role for the proper functioning of the machine. The ratio between its stiffness and mass is an important factor to guarantee a quick and precise configuration of tyre-to-ground contact condition, keeping inertial phenomena under control. Therefore, a constraint of maximum deflection of the table less than, or equal to 1 mm was imposed. The result of the analysis is a weight reduction of 167 kg, corresponding, approximately, to 15% of the initial value. Also in this case, von Mises stress and maximum displacement are fully acceptable for the new geometry obtained after topology optimisation. Detailed results are highlighted in **Figure 9** and **Table 7**. Only the worst load case is analysed, with the hexapod table having 45° inclination respect to the horizontal plane.

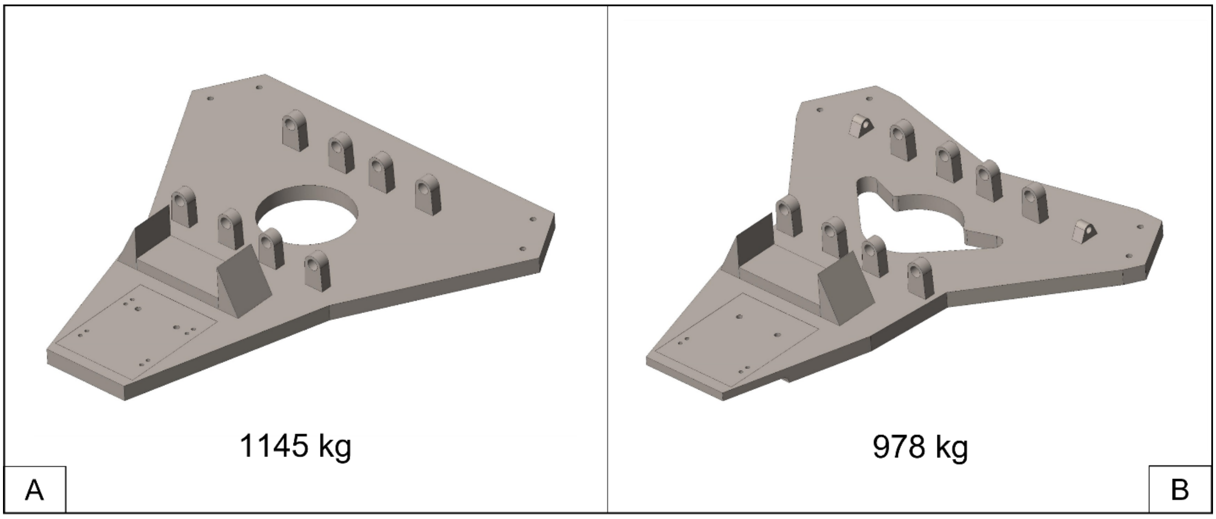

Figure 9: Geometry of the hexapod table: preliminary design (A) and after topology optimisation (B)

Design phase	von Mises Maximum Stress [MPa]	Maximum Displacement [mm]
Preliminary	53.3	0.39
After optimisation	42.2	0.50

Table 7: FEA results comparison for the hexapod table before and after topology optimization

3.2.3. Topology optimisation of the sliding plate for load transmission between hydraulic press and suspension system

The preliminary design of the sliding plate led to a very simple and rough shape, which was then optimised just imposing, as a constraint, no mass removal from those regions directly involved in the connection both with the HEB beams of the frame and with the suspension system. This process was done in two steps. In fact, despite the fact that the first step of the optimisation gave a very good result in terms of weight reduction, an intermediate modal analysis suggested a stiffness increase of the structure. We must remind that a modal analysis was conducted after each design step of this work, in order to check the accuracy of the solutions adopted, which are summarized in section 3.3 for clearer presentation.

The final design solution consists of two smaller lateral arms, instead of just a bigger one, welded to the plates that are connected to the HEB beams. The weight reduction is 308 kg, corresponding to 49% of the initial value, without affecting the strength and functionality of the structure. In **Figure 10** and **Table 8**, geometry and FEA results of the topology optimisation are shown. The safety factor is of around 3.5, with reference to the yield stress of the material used for the sliding plate (460 MPa for steel S460J0).

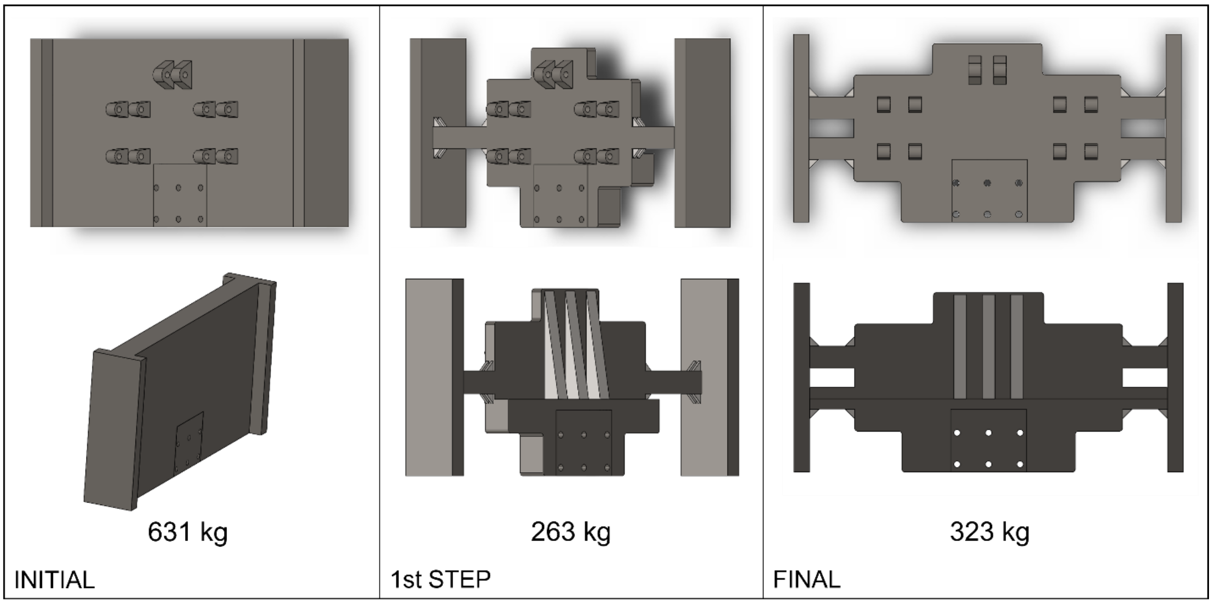

Figure 10: Geometry of the sliding plate: from preliminary design to final optimisation

Design phase	von Mises Maximum Stress [MPa]	Maximum Displacement [mm]
Preliminary	82.6	0.10
1st step optimisation	133.1	0.61
Final optimisation	130.9	0.68

Table 8: FEA results for the sliding plate before and after topology optimisation

3.3. Final modal analysis

The modal analysis of a system is an important method to investigate its natural frequencies of vibration, thus indicating the various periods at which the system would resonate, due to its mass and stiffness. The frequency of load application, during working conditions, must be far from the natural frequencies of the system, to avoid the possible occurrence of increasing amplitude of structure vibrations and displacements, which would lead to failure. For that reason, in this work, a modal analysis was done on the main machine components after each design step. Indeed, the radial and axial runout of the tyre and wheel assembly would generate reaction forces in the structure, directly related to the rotational speed of the system. The frequency of these forces, f, occurring in the machine during working conditions, was calculated according to **Equation 1**, considering a linear speed range between 50 km/h and 130 km/h for the tyre and wheel assembly, based on a standard 16” diameter size (nominal diameter *d* = 406.4 mm):

$$v = \omega * r = \omega * (d/2) = \pi * f * d, \qquad f = v/(\pi * d) \tag{1}$$

where v is the linear speed in millimetres per second [mm/s], $\omega = 2\pi f$ is the angular speed in radians per second [rad/s] and $r = d/2$ is the wheel nominal radius in millimetres [mm].

The resulting range of frequencies is reported in **Table 9**, with the corresponding linear speed indicated in kilometres per hour [km/h], because this is the unit commonly used for ground vehicles speed. The frequency values of **Table 9** were used just as a reference for comparison, since the tyre diameter directly influences these values, decreasing them for bigger diameters and increasing them for smaller ones. Considering the fact that speed has also a direct impact on tyre temperature during test, the maximum speed condition will probably only be used for specific test purposes, while the majority of tests will run at an intermediate speed range. Therefore, the frequency range which the machine will be submitted to, during standard work, is estimated between 15 Hz and 20 Hz.

Speed [km/h]	Frequency [Hz]
50	10.9
70	15.2
100	21.8
130	28.3

Table 9: Estimated work frequency range of the machine, related to the equivalent linear speed of a 16" wheel

All the main components of the machine were subjected to modal analysis, keeping the same constraints used for the static structural FEA discussed in sections 3.1 and 3.2. An example of the shape of the first natural frequency for frame, hexapod table and sliding plate is shown in **Figure 11**.

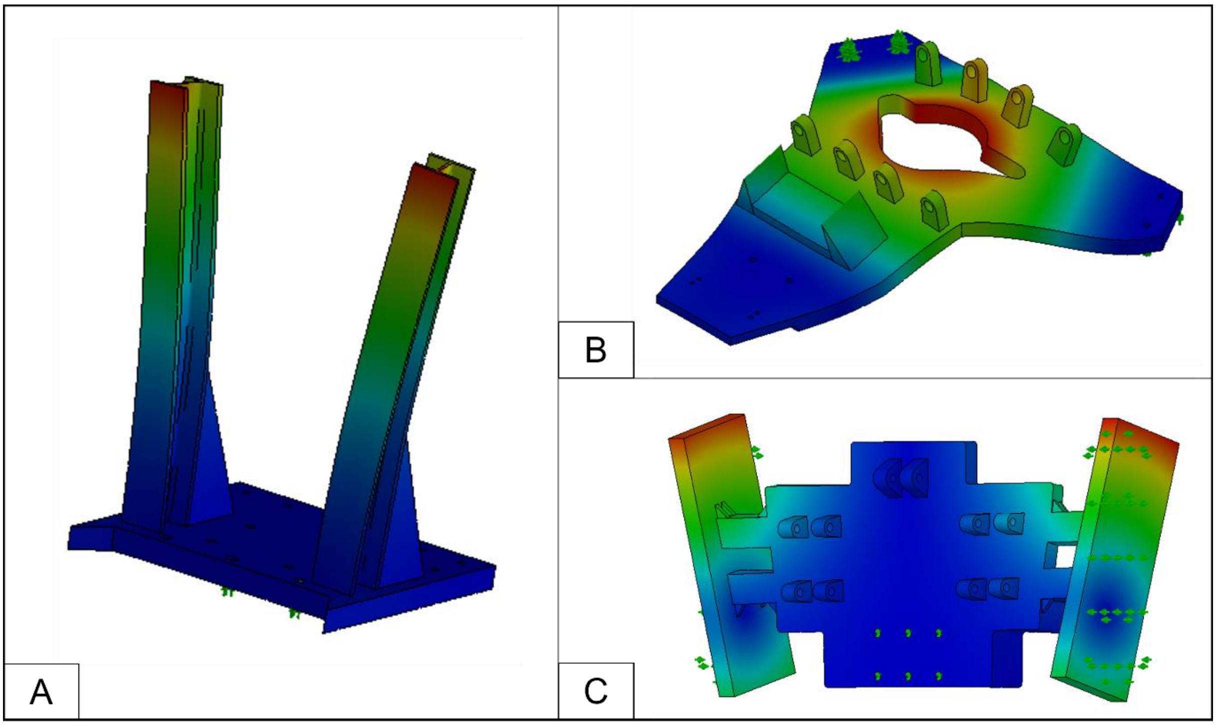

Figure 11: Modal analysis of the machine components: example of first natural frequency shape for frame (A), hexapod table (B) and sliding plate (C)

Detailed results for mode 1 to 4 are represented in **Table 10**, in [Hz] unit. A comparison is also made between the results of preliminary design and final geometry after the topology optimisation process, highlighting how that process has no major effects on the dynamic response of the components, in relation to the expected work frequency range of the machine.

	Frame		Hexapod table		Sliding plate	
Mode	Preliminary design	After optimisation	Preliminary design	After optimisation	Preliminary design	After optimisation
1	34.2	34.0	128.5	155.7	253.6	156.0
2	34.6	34.7	209.0	246.7	286.2	158.9
3	63.0	69.6	214.2	278.0	479.5	281.0
4	63.2	70.0	423.6	375.6	547.5	284.8

Table 10: Natural frequencies values of machine components: comparison between preliminary and optimised design. Unit of measurement [Hz]

For further check, a modal analysis was also conducted on a simplified model of the whole machine. The simplified model was used for computational reasons. The shapes of the natural frequencies mode 1 to 4 are shown in **Figure 12**, while analytical values are reported in **Table 11**.

Mode	Frequency [Hz]
1	30.5
2	32.1
3	38.7
4	41.7

Table 11: Natural frequencies values of the complete machine (simplified model)

Analyzing the results, the lower natural frequency value for the whole structure is 30.5 Hz, which is also similar to the minimum of the machine frame alone (34 Hz, see **Table 10**).

Since the machine is supposed to work in the frequency range between 15 and 20 Hz for most of the tests, the safety factor is of around 2 and it is considered acceptable for the machine purposes.

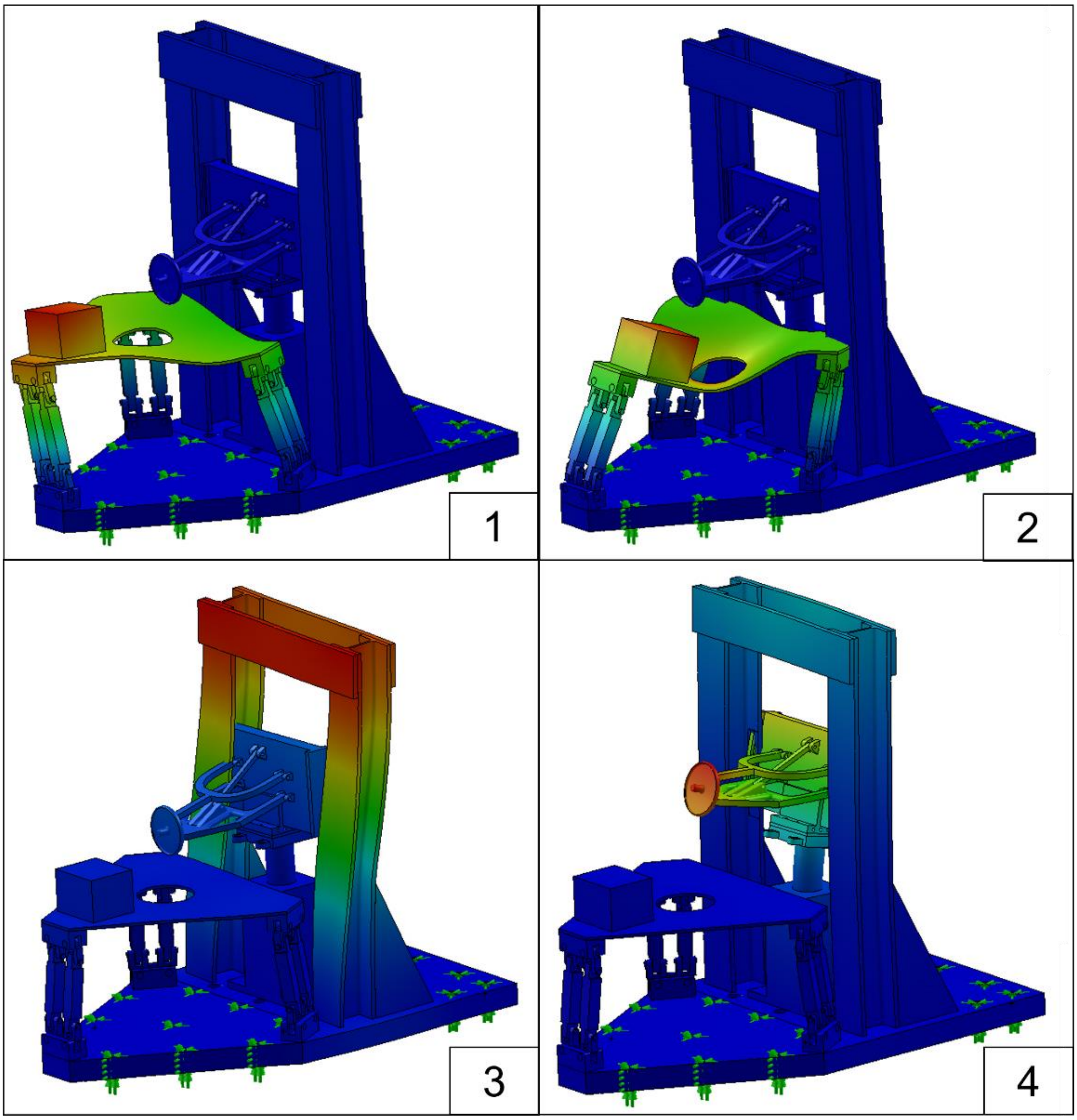

Figure 12: Modal analysis for the complete machine (simplified model): natural frequencies mode 1 to 4

3.4. Safety devices

Safety devices were also designed to guarantee both people safety and machine proper functioning. A series of controls, based on the sensors briefly discussed in section 2, needs to be implemented during the software development, which will be the next step of this work. Of course, in case of manufacturing, the machine must be compliant with the norms required by the Machine Directive 2006/42/EC. However, for the aim of this paper, only two main parts are briefly described in this section: the safety guard and the cooling system. They are both represented in the machine 3D model of **Figure 2**.

The safety guard is made of a steel frame and a plastic screen. It is designed to prevent tyre debris from hitting people or machine parts during tests. The steel frame is composed of hollow section bars, assembled through welding, in steel S460J0, while the plastic screen is made of ABS. Even during the design phase of the safety guard, modal analysis were done both for the two separate components and for the assembled structure. Detailed results are shown in **Table 12**, in [Hz] unit. Again, these values are acceptable for the design target.

Mode	Guard frame	Guard screen	Assembled structure
1	54.5	37.6	34.8
2	83.0	68.9	35.5
3	87.9	69.0	47.3
4	95.4	94.9	53.6

Table 12: Natural frequencies values of safety guard: separate components and assembled structure. Unit of measurement [Hz]

Finally, a cooling system is provided, in order to reduce the tyre temperature in test conditions. We must remember that tyre temperature will be constantly measured by an appropriate sensor during test duration, giving the possibility of suspending tests or reducing speed in case of overheating. The fresh air flow driven by the cooling system will help both to increase the overall tyre life and to keep lab test conditions more similar to real use on road or ground.

The cooling system is quite simple. It is made of three components: three-phase asynchronous motor, axial fan and pipe for air flow. The motor has a power of 2.2 kW with 2750 rpm maximum speed, while the axial fan is formed by a rotor of 200 mm-diameter mounted in a galvanized steel cage, which prevents the system from corrosion. The pipe is made of PVC over a steel spiral structure, which increases its strength without affecting the overall weight of the assembly.

4. Conclusions

Components involved in the interaction with road and ground are fundamental for the functionality and safety of the vehicles. A number of lab tests have historically been developed in the automotive and off-highway industry for the validation of their design, in terms of performance and durability. The aim of this work is the development of an innovative concept of test machine for tyres, wheels and suspension systems, which could be used for testing all these kinds of components, in several conditions, just on a single test rig. The core of the design is a hexapod table that gives the opportunity to recreate many possible contact configurations between tyres and road, thanks to its six degrees of freedom. The general design of the machine started from a preliminary study that was subsequently refined by FEA, regarding both the static structural and the dynamic behaviour of single components and complete assembly. The main design targets are achieved and safety devices are also included. The next step of this work will be the development of the software to drive and control the whole system, using a closed-loop feedback configuration for the first machine prototype, now at implementation stage.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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