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Design of electro-mechanical height adjustment system for multi-link suspension

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Abstract

Electro-mechanical height adjustment suspension system is a promising automotive technology. It works on the ground clearance of the vehicle, reducing the aerodynamic drag with not negligible benefits in terms of fuel consumption and CO_2 emissions; besides these advantages, it improves also the versatility in the use of the vehicle.

The present work examines in a systematic way the state of art of the existing solutions, summarizing the main features, the working principle and the architectures, with particular attention to the electro-mechanical systems, which represent the best compromise between functionality, high reliability and low costs.

The aim of this master thesis is the design of a reversible eccentric electro-mechanical height adjustment system for a multi-link suspension (Alfa Romeo Stelvio rear suspension), that permits to obtain the best results with references to the power consumption and actuation speed.

The optimized design has been realized by running numerical simulation, using parametrized models to evaluate performances and considering the constraints due to the implementation of the actuator in an existing vehicle.

Furthermore, the different components of the device have been refined and then detailed 2D and 3D drawings are obtained.

The last step of the project consists in an experimental validation of the numerical model and the set-up of a suitable electronic control unit.

Acknowledgements

I would like to express my great gratitude to Prof. Nicola Amati, supervisor of this thesis, for the opportunity of taking part to this project, for his availability and for the constant advices.

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1. Introduction

The aim of this chapter is to provide an introduction to the dissertation in order to understand the background and the goals of the project.

The environmental benefits and an overview on the state of art of the height adjustment suspension system will be provided along with the structure of the thesis.

1.1 Environmental benefits

Road transport is one of the most consumers of fossil fuel and air pollution. According to the data of the International Energy Agency, private cars and vans were responsible for around 10% of global energy-related CO_2 emissions in 2022. [1]

Since 2010, rapid electrification, accompanied by technologies that improve fuel economy (including hybridization) in vehicles that run on combustion engines, led to the largest improvement in the specific fuel consumption.

However, the rise of heavier and less efficient cars such as SUVs continues to slow progress on vehicle efficiency improvements; in 2022, SUVs accounted for around 46% of global car sales and so their combustion-related CO_2 emissions increased by nearly 70 million tons in 2022.

In 2023 the European Union announced stricter CO_2 emission to ensure carbon neutrality by 2050; so, on 19 April 2023, the European Parliament and the Council adopted Regulation (EU) 2023/851 to strengthen the CO_2 emission performance standards for new passenger cars in line with the EU's increased climate ambition. The EU fleet-wide CO_2 emission targets set for new vehicle are as follow:

- From 2020 to 2024, new cars have to emit less than 95 g CO_2/km;
- From 2025 to 2029, 15% reduction is planned;

- From 2030 to 2034, 55% further reduction;
- From 2035 onwards, 100% additional reduction.

These target levels refer to the New European Driving Cycle (NEDC) emission test procedure. [2]

Technologies that are widely used to decrease fuel consumption and exhaust gas emissions are based on improving the internal combustion engine (ICE) and the drive train efficiencies, on electrification, on lowering vehicle weight and on using active chassis components, like as height adjustment suspension system.

The latter works on the ground clearance of the vehicle, optimizing the aerodynamic shape and reducing the frontal area, decreasing aerodynamic resistance D:

$$D = \frac{1}{2} * C_x * \rho * A * V^2 \tag{1}$$

where C_x is the aerodynamic drag coefficient, ρ is the air density, A is the reference area and V is the vehicle speed.

In Figure 1, the graph represents the two main resistances (rolling and aerodynamic) to the motion of the vehicle as function of its velocity.

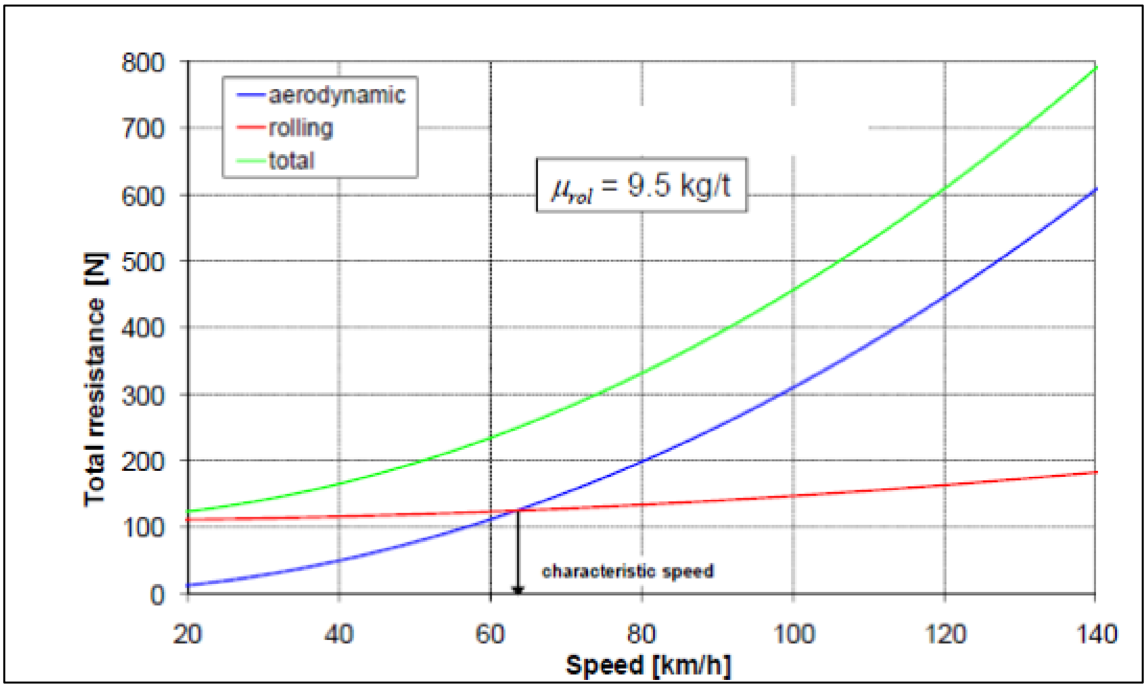

Figure 1: **Total, rolling and aerodynamic resistances as function of speed for a mid-size passenger cars** [3]

Since the lowering of the vehicle height could give benefits in both the aerodynamic drag coefficient C_x and frontal area A, it is then considered a promising feature to be used on a modern car to reduce fuel consumption and CO_2 emission.

By referring to the paper "Design of electromechanical height adjustment system", Amati et al. analyze the benefit of adjusting ground clearance by using a vehicle model including the longitudinal dynamics; for the study, an average A-segment car was chosen and the engine torque and the specific fuel consumption maps implemented in the model are experimentally measured by the manufacturer. [4]

The aerodynamic drag coefficient C_x of the vehicle at nominal road clearance is 0.35; it has been measured numerically and experimentally, that decreasing the vehicle height by 20 mm and 40 mm, this coefficient will reduce respectively to 0.33 and 0.31.

The car parameters used in the simulations are reported in Table 1.

Parameters	**Value**
Vehicle mass [kg]	1090
Engine volume [cm^3]	900
Nominal power [kW]	600
C_x at nominal vehicle height [-]	0.35
C_x at 20 mm lowered vehicle height [-]	0.33
C_x at 40 mm lowered vehicle height [-]	0.31

Table 1: **Vehicle parameters for fuel consumption simulations** [4]

Running the vehicle simulator at different ground clearances, fuel consumption and CO_2 emission values for NEDC and Worldwide Harmonized Light Vehicles Test Procedures (WLTP) are obtained.

During the simulations, the value of the aerodynamic drag coefficient C_x is decreased from nominal to one of the reduced values only on extra urban parts of the homologation cycles, in which the vehicle speed V is high and the aerodynamic resistance D is predominant.

Fuel consumptions obtained by the simulations are then converted in CO_2 emission by using the conversion factor suggested in EC Regulation No 443/2009 for gasoline engine (i.e. 2330 g CO_2 for 1 L of petrol).

The fuel consumption reduction at different road clearances compared to the nominal value is in the range 0.037–0.22 L/100 km (0.88–4.41 %), which corresponds to a decreasing of CO_2 emissions by 0.87–5.17 g/km. The results are depicted in the following Figure 2.

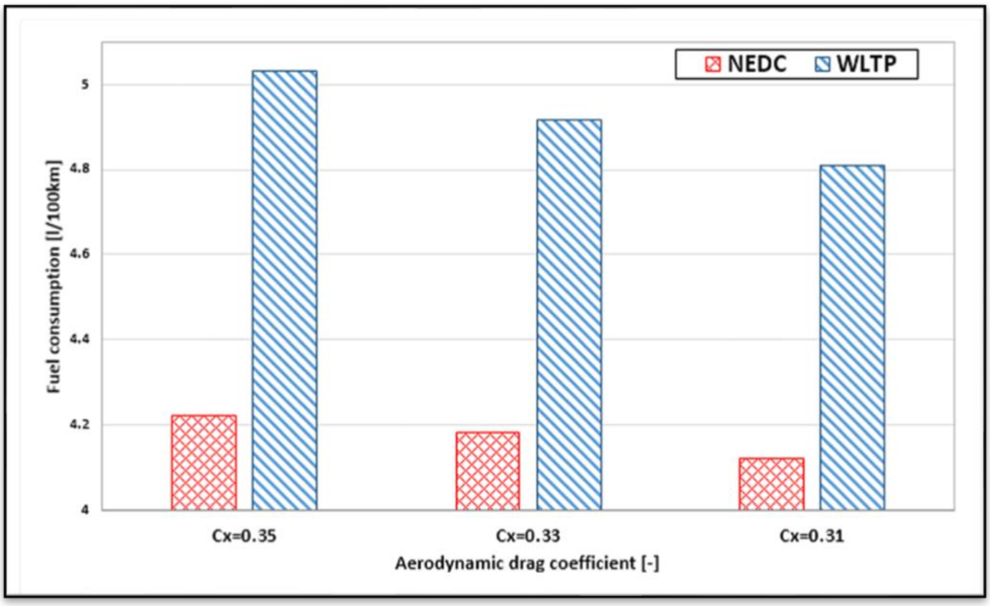

Figure 2: **Fuel consumption in NEDC and WLTP homologation cycles at different value of C_x** [4]

These outcomes justify the use of height adjustment systems on cars to achieve lower CO_2 emissions.

Besides that, added benefits are present, including possible reduction of roll body, pitch adjustment to load distribution, improved accessibility and also higher adaptability of the vehicle to different type of roads with different roughness, considering also off-road.

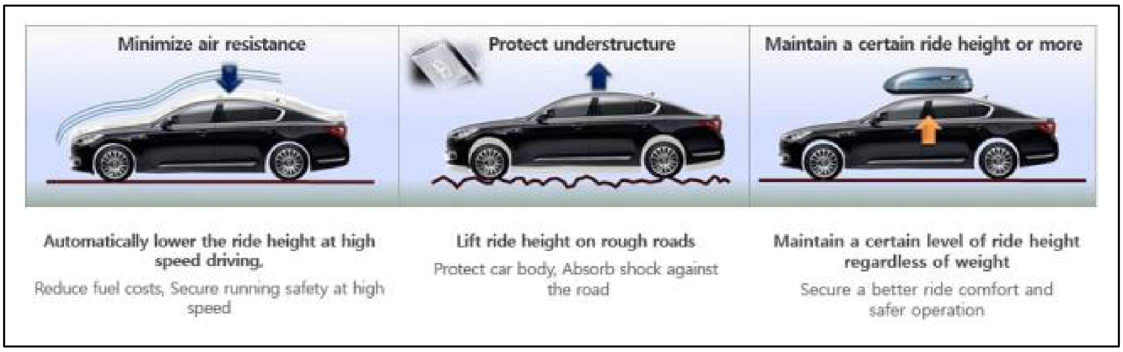

Figure 3: **Main benefits to use height adjustment suspension system** [5]

Thus, height adjustment suspension system is able to give a perfect compromise between fuel consumption and versatility in the use of the vehicle.

1.2 State of art

A suspension is a mechanical system, that is essential to obtain certain dynamic performances in terms of handling, road holding and comfort.

It links the wheel to the body of the vehicle and it is composed mainly, making reference for instance to the front suspension of Lancia Thesis represented in Figure 4, by primary elastic elements (springs – green elements), secondary elastic elements (bump-stops, rebound-stops and bushings – red elements) and damping elements (shock-absorbers – blue elements).

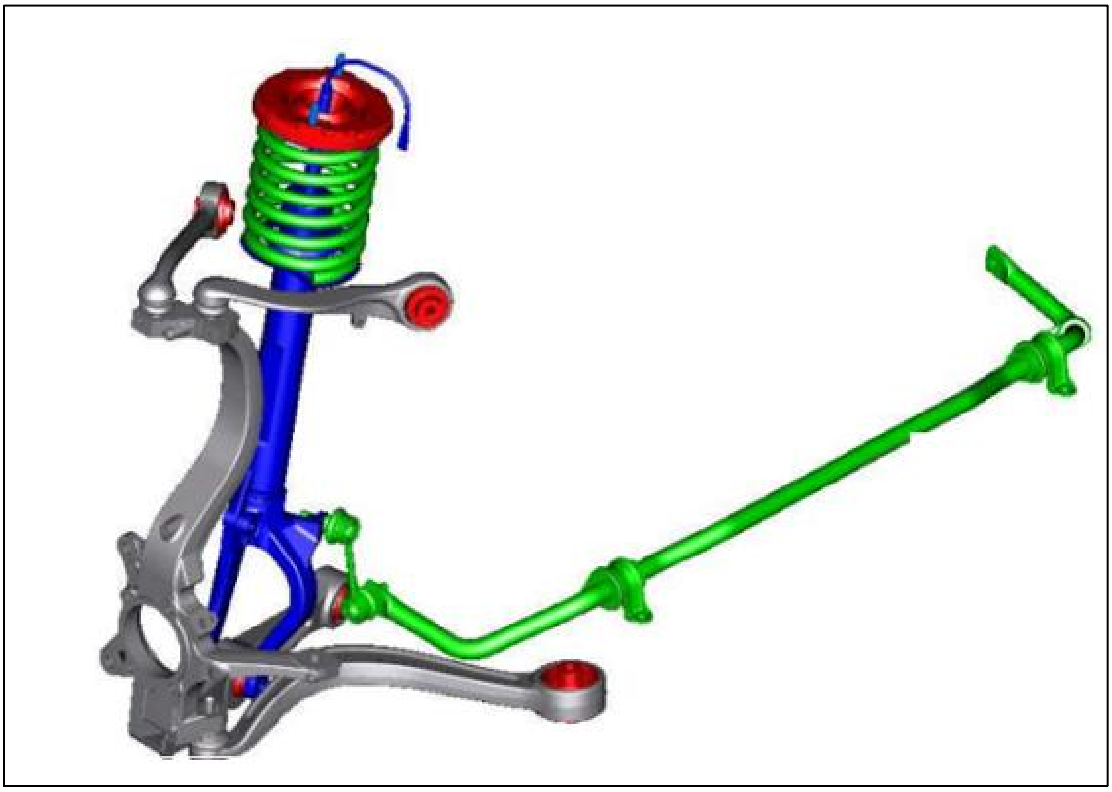

Figure 4: **Front suspension Lancia Thesis**

The main aims of an automotive suspension are:

- Create a mechanical filter to the stress derived from the road and transmitted to the vehicle body to guarantee comfort, reducing the effect of road irregularities on the passengers and on the carried load;
- Ensure the right tyre-road contact in the different working situation to improve road holding capability and handling.

Based on the working principle of damping elements, vehicle suspension can be divided into: passive, semi-active and active.

Passive suspensions haven't adjustable characteristics, while in semi-active and active ones, the value of damping can be varied based on input signals measured by vehicle sensors, according to an appropriate control logic.

Active and semi-active suspensions differ from one another in the amount of energy, that has to be injected into the system from other sources, for example electric motors, to provide regulation.

Active suspensions are used to continuously control in real time the vehicle dynamics; they are more complex and require higher energy than semi-active ones, which only actuates some electronically controlled elements, such as control valves.

Vehicle height adjustment suspensions are classified into active suspension with small actuation bandwidth.

Basically, the modification of the ground clearance is fulfilled by increasing or decreasing the relative distance between the vehicle body and the wheel hub; this can be realized by acting on different parts of the suspension structure, like as: upper spring seat, lower spring seat or shock absorber tube.

On the basis of the type of actuation, height adjustment device can be classified as hydraulic, pneumatic, hydro-pneumatic or electro-mechanical.

1.2.1 Hydraulic height adjustment system

Hydraulic height adjustment systems are in general composed by linear or rotary actuators (respectively called hydraulic cylinders or hydraulic motors), hydraulic pumps, pipes, control valves, sensors, tanks, accumulators and hydraulic fluids.

Evaluating the different solutions present as patents, based on the part of the suspension on which is realized the actuation, it's possible to distinguish two main categories:

- The height adjustment is performed actuating the upper spring seat, modifying the compression of the spring;

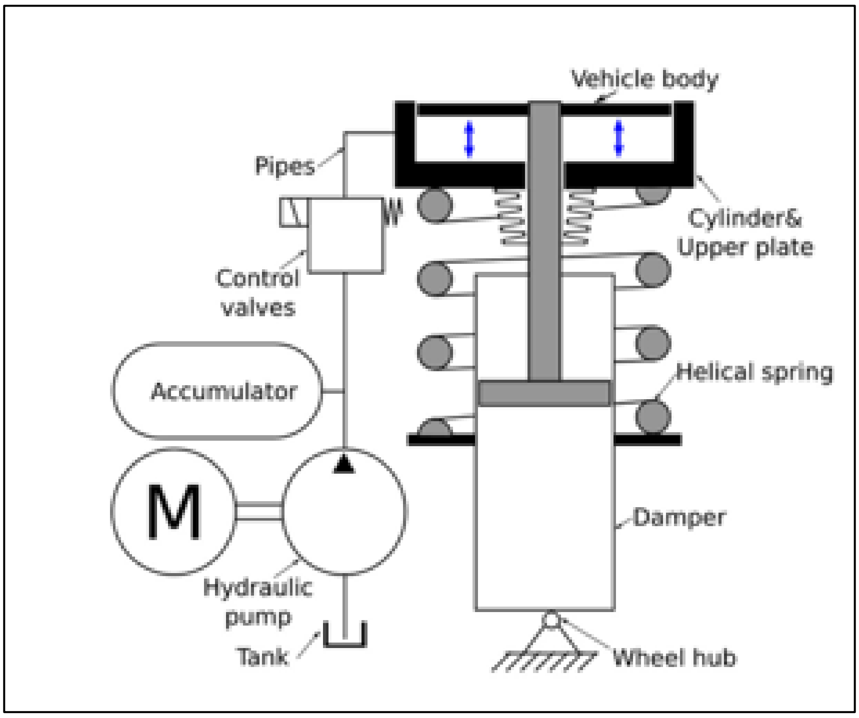

Figure 5: **Hydraulic height adjustment system with upper spring seat actuation** [4]

- The height adjustment is performed actuating directly the piston rod of the hydraulic damper.

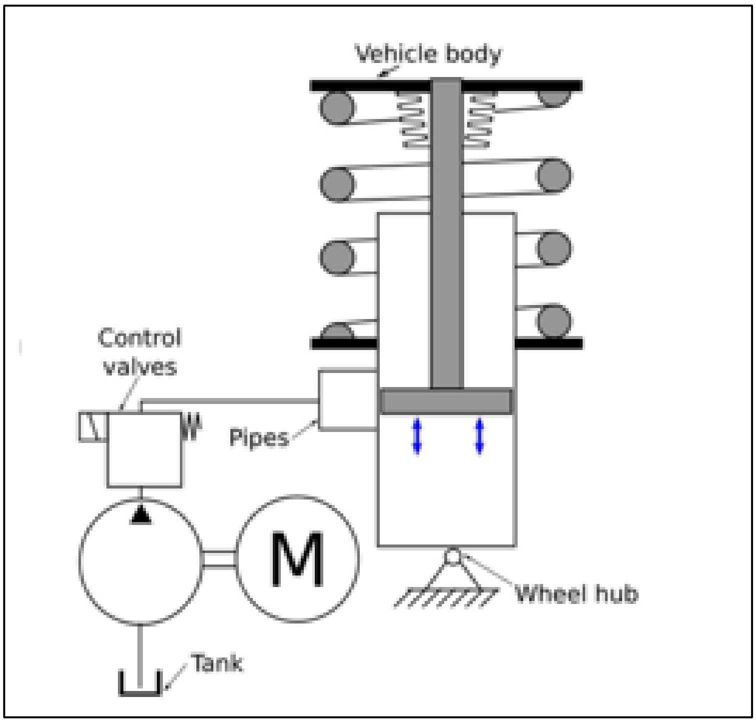

Figure 6: **Hydraulic height adjustment system with damper actuation** [4]

Referring to the first typology, Mercedes Benz AG has patented a solution to have height adjustment feature, but also to improve ride comfort and stability, controlling yaw and rolling movement and compensate for inclined positions of the vehicle body. [6]

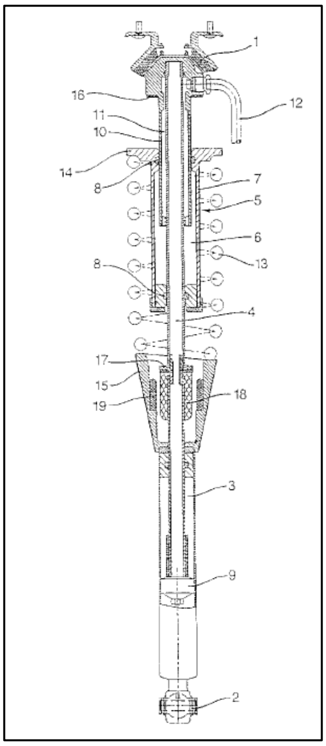

Figure 7: **Mercedes Benz AG hydraulic height adjustment system** [6]

This invention relates a traditional coaxial spring (13) - shock absorber (composed by a damping cylinder (3) and a piston rod (4)) suspension structure, like for example Figure 4, disposed between the vehicle body and the wheel hub by means respectively of a rubber support (1) and a joint (2).

The height adjustment device (5) is a linear actuator mounted in series to the helical spring (13) and in parallel to the damper and it is composed by a cylindrical hollow element (7) with an annular collar (14), that forms the upper spring seat. It is supported on the piston rod (4) so as to be axially movable by means of changing the volume of hydraulic fluid in the operating space (6), realized between the piston rod (4) and the cylindrical component (7).

So, to modify ground clearance, this system actuates the cylindrical part (7) with an engine driven pump, that feeds or discharges the operating space (6), through an annular gap (11) between the piston rod (4) and a sleeve structure (10); the latter (10) connects both piston rod (4) and the cylindrical element (7) to the vehicle body. The annular gap

(11) is in communication in a controlled manner with a hydraulic medium storage space via a line (12).

Hence, displacing the cylindrical housing (7) and as a result the upper spring seat, the compression of the helical spring (13) is modified, causing a change in the vehicle height.

Always related to the first type of hydraulic height adjustment unit, Daimler-Benz AG has patented a different solution to modify ground clearance, but also to counteract pitching movements, rolling movements and tilted position of the vehicle body. [7]

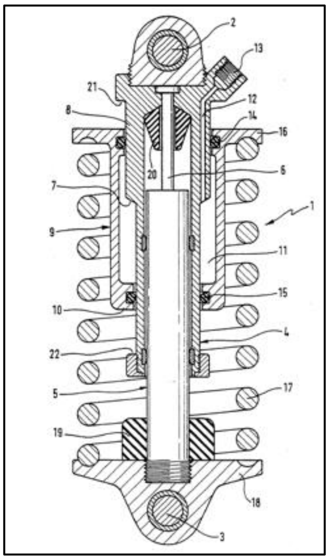

Figure 8: **Daimler Benz AG hydraulic height adjustment system** [7]

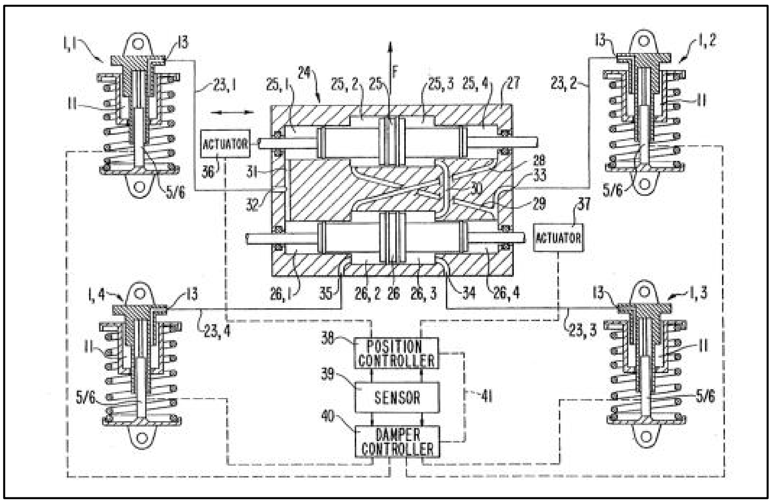

Figure 9: **Daimler Benz AG schematic representation of the overall hydraulic height adjustment system** [7]

This device, also, is related to a traditional coaxial spring (17) – shock absorber (composed by a damping cylinder (5) and a piston rod (6)) suspension structure, which arranged in a cardanically pivotable manner to the vehicle body by way of a body-side joint (1) and to the wheel-hub by means of a wheel-side joint (2).

This system is a linear actuator that works in series with the helical spring (17) and in parallel with the damper. It consists of two hollow cylindrical elements: the first one (4) is arranged on the body-side joint (1) and accommodates in a telescope fashion the damping cylinder (5), which is settled on the wheel-side joint (2), while the second one (9) is mounted on the outside of the first cylinder (4) and is guided in an axially displaceable way. This last (9) has in addition a flange (16), that forms the upper spring seat.

So, to change the vehicle height, the cylindrical element (9) is moved by way of an controlled actuating hydrostatic feed unit (24), that feeds or discharges hydraulic fluid into the annular gap (11) between the two cylindrical parts (4,9) via a line (13) through an axial hole (12) and an hydraulic connection (13).

The feed unit (24) has two-side stepped pistons (25,26) of symmetrical configuration which are arranged moveably in correspondence of stepped bores of housing (27), obtaining separated working spaces (from 25,1 to 25,4 and from 26,1 to 26,4), that are connected to one another by means of lines (from 28 to 31) and to the annular space (11) of the spring strut (from 1,1 to 1,4).

The pistons (25,26) are actuated by actuating units (36,37) which are controlled by a position controller (38). The latter (38) receives input signals from body/wheel sensors (39) and from that, it can determinate how the vehicle is attempting to move.

Thus, when the hydrostatic feed unit (24), moving the two piston (25,26), changes the volume of hydraulic medium in the operating space (11) of the spring structure (from 1,1 to 1,4), the cylindrical element (9) with the upper spring seat (16) moves axially, modifying the compression of the helical spring (17) and so the vehicle height.

Instead, referring to the second category of hydraulic height adjustment system, Hyundai Motor Company has patented a solution to adjust ground clearance with an actuation directly in the damper. [8]

This invention relates to a monotube shock absorber, that permits to obtain a height adjustment device with advantages in the vehicle layout such as decrease of weight due to reduction of size, diminution of manufacturing costs and securing more installation space respect to a twin-tube type cylinder one.

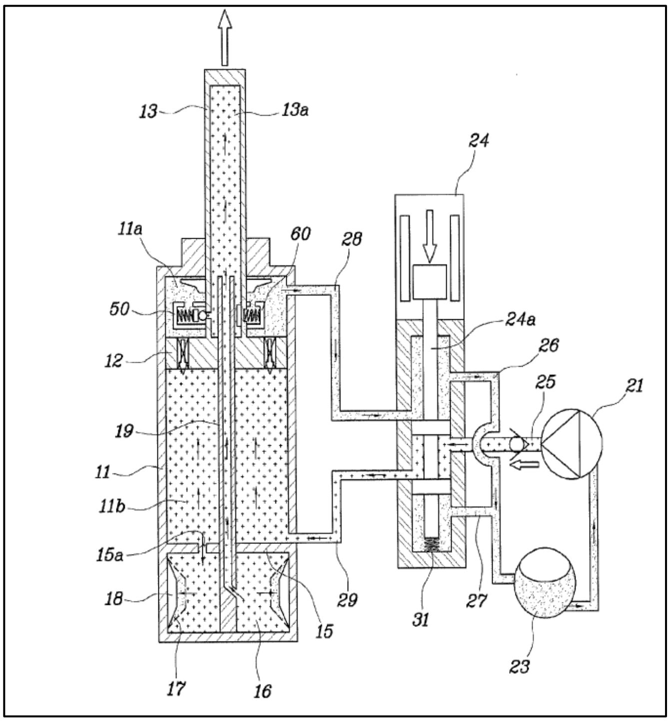

Figure 10: **Hyundai Motor Company hydraulic height adjustment system: ground clearance increased** [8]

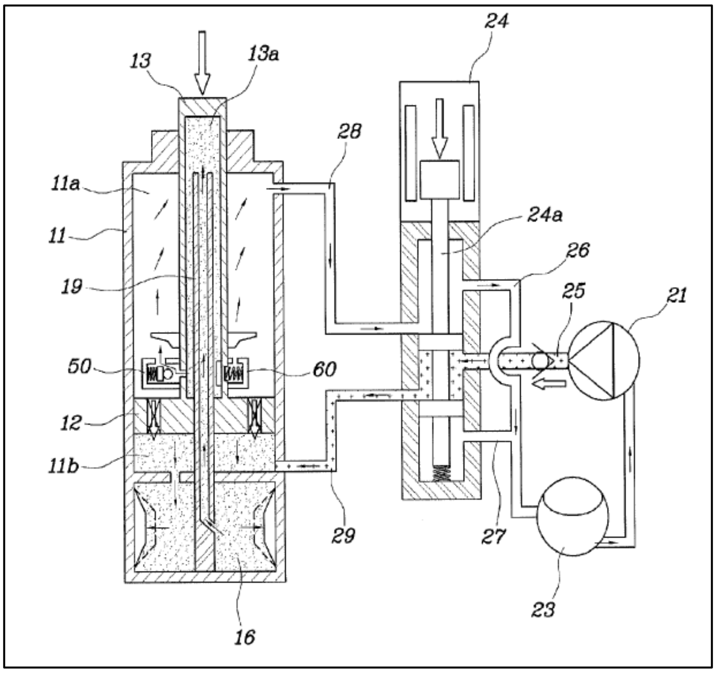

Figure 11: **Hyundai Motor Company hydraulic height adjustment system: ground clearance decreased** [8]

The shock absorber apparatus mainly is composed by a monotube type cylinder (11), a piston (12) installed within the cylindrical element (11) to move along a lengthwise direction of it, dividing it in an upper (11a) and a lower (11b) chamber, a piston rod (13), having an oil flow space (13a), that is connected on one end to the piston (2) and on the opposite end to the vehicle body passing through the cylinder (11), a support chamber (16) separated from the lower one (11b) by means of a partition wall (15) with an oil passage (15a), an high pressure chamber (18) divided from the support one (16) by a diaphragm (17). Further there are two orifices (12a) in the piston (12) to unite the upper (11a) and the lower (11b) chamber, a valve mechanism (14) placed in the piston rod (13) to link the oil flow space (13a) and the lower chamber (11b) and a pipe member (19) to connect the high-pressure chamber (18) with the oil flow space (13a).

The height adjustment unit is a linear actuator integrated with the damper, that includes a controlled engine driven oil pump (21) linked to an accumulator (23) via pipe line (22) to generate hydraulic pressure and a controlled solenoidal valve (24). This last (24) is connected with the pump (21) by means of a supply pipe line (25), with the accumulator by two return pipe lines (26 and 27), with the upper chamber (11a) of the cylinder (11) via a first pipe line (28) and with the lower one (11b) via a second pipe line (29).

Therefore, changing the volume of the hydraulic fluid inside the upper (11a) and the lower (11b) chamber of the monotube cylinder (11) by means mainly of the oil pump (21) and the solenoidal valve (24), the piston (12) is actuated and moved relative along to a lengthwise direction of the cylindrical tube (11), that is fixed on the wheel hub. So, the piston rod (13), connected to the vehicle body, is displaced modifying ground clearance.

Another damper actuation solution has been patented by Bayerische Motoren Werke Aktiengesellschaft (BMW); along with the height adjustment feature, this invention has the possibility of control actively the damping force of the hydraulic shock absorber and counteract possible rolling movement of the vehicle body, introducing additional force between the latter and the wheel. [9]

Making reference to the following Figure 12, this suspension system is composed by a spring element (2) in parallel to a hydraulic piston (1b) - cylinder (1a) unit (1) acting both as damper and liner actuator; the piston rod is connected to the wheel while the cylinder is mounted on the vehicle body (16).

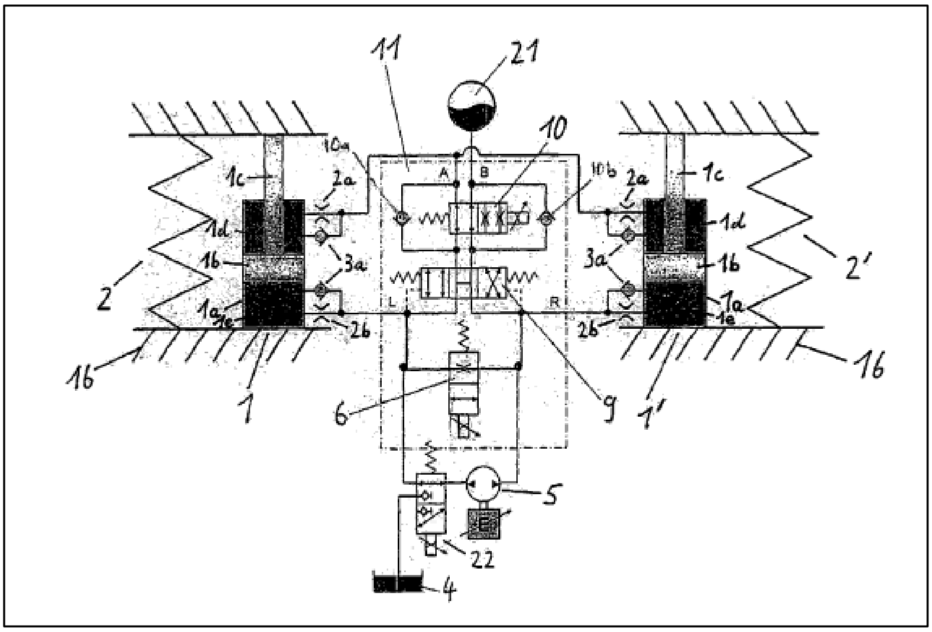

Figure 12: **BMW hydraulic height adjustment system** [9]

The height adjustment device (1) has two working chambers (1d,1e) and to each of them an own hydraulic supply is assigned to introduce or remove hydraulic medium. Moreover, its arrangement primarily contains a hydraulic tank (4), an engine driven pump (5), several controlled valves (6,9,10,22), a pressure accumulator (21), pressure sensors and an electronic control unit. Then, a damper valve (2a and/or 2b), that can be passive or regulable, in parallel with a non-return valve (3a), permits to the actuator (1) to work effectively simultaneously as a damper, with respectively constant or modifiable damping characteristic.

So, to modify ground clearance, the pump (5) in a controlled manner, delivers a quantity of hydraulic fluid in the appropriate working chamber (1d and/or 1e) to reach the desired height.

Hydraulic height adjustment suspension systems have some advantages, such as:

- Fast actuation, related to the use of high-pressure hydraulic medium;
- Compactness, in fact the main components of the device can be fixed on the vehicle sprung mass, leading to slight modification of the unsprung one.

However, these devices have some drawbacks, related mainly to the usage of high-pressure hydraulic fluid:

- High cost of manufacturing, concerning the request tight tolerances;
- Corrosion of metallic parts due to moisturization.

1.2.2 Pneumatic height adjustment system

Pneumatic suspension is mainly used in public transport, heavy-duty and premium cars where the aim is to improve passenger comfort.

The main components consist of air spring (usually, a reinforced rubber bag with pressurized air), high pressure tank, air compressor, air dryer to avoid the problem of the moisturization and a control system including valves and sensors.

The adjustment of the vehicle height is obtained by pumping air into or out of the air spring bag.

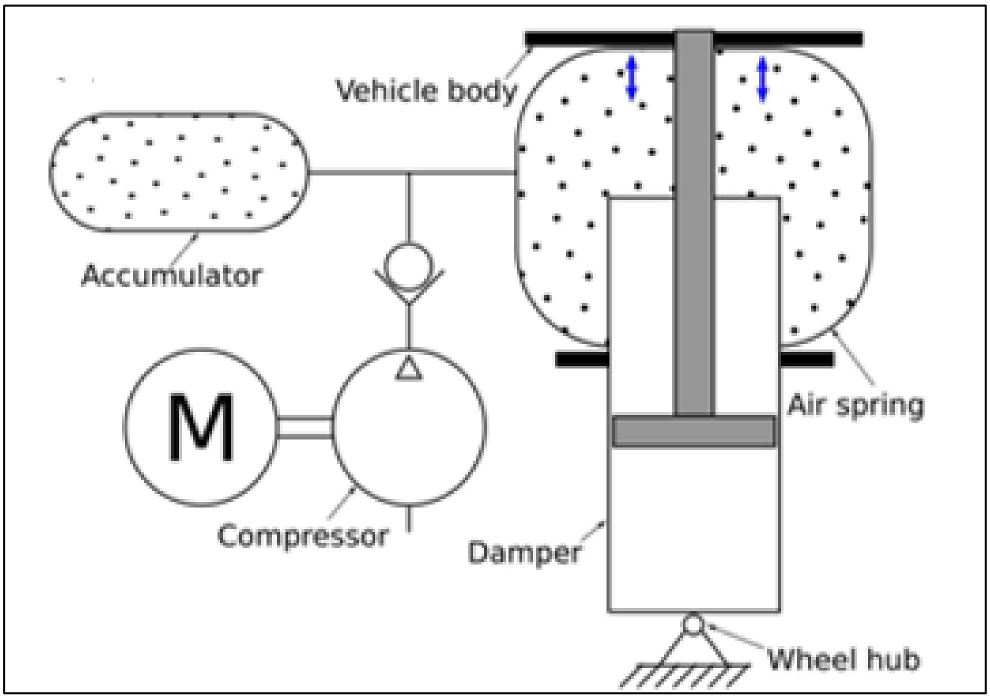

Figure 13: **Pneumatic height adjustment system** [4]

In the paper "Toyota Electronic Modulated Air Suspension System for the 1986 Soarer", Hirose et al. describe Toyota's electronic modulated air suspension (TEMS). [10]

Toyota has developed this solution to have height adjustment feature but also to control spring rate and damping force to permit soft riding and stable maneuverability.

The system is mainly composed by an electronic control unit (ECU), sensors, including height ones, single cylinder reciprocating type compressor driven by dc motor, solenoidal height control valve and controlled air spring and shock absorber unit.

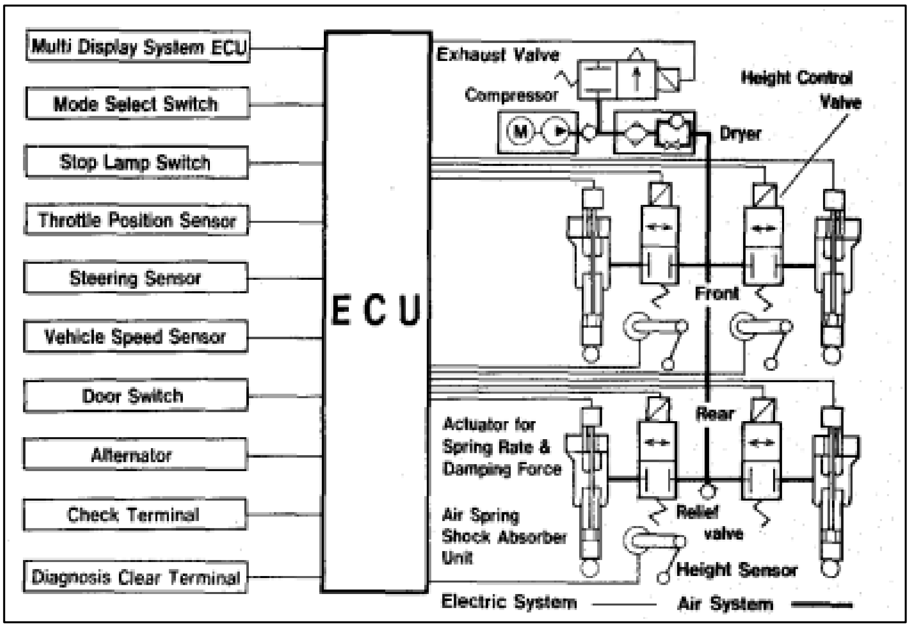

Figure 14: **Toyota air suspension scheme** [10]

The modification of the ground clearance is provided by means of the compressor and the solenoidal height control valve that opens and closes the air passage from the air lines to the air spring to change the amount of air and the pressure inside it.

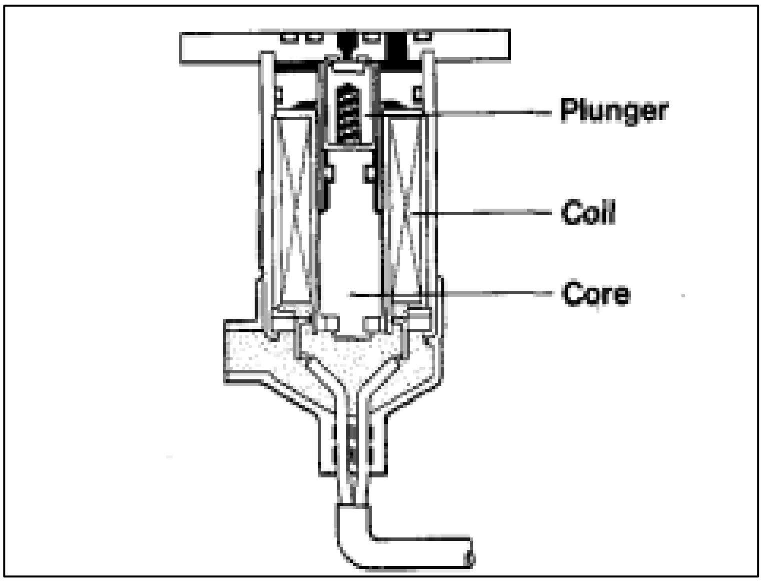

Figure 15: **Toyota pneumatic height adjustment system: height control valve** [10]

The vehicle height, together with the spring rate and damping resistance, can be adjusted actively by way of the ECU on the basis of sensors signals; however, they can be also modified by the driver selecting via a mode select switch the desire control mode for these ride parameters.

Another solution is described by Tener in the paper “Overcoming the Ride/Handling Compromise – A Cockpit Adjustable Suspension System”. [11]

This pneumatic suspension relates to a MacPherson structure and has the aim to adjust the vehicle height but also to control the spring rate to improve both comfort and handling, monitoring the amount of air in the air spring.

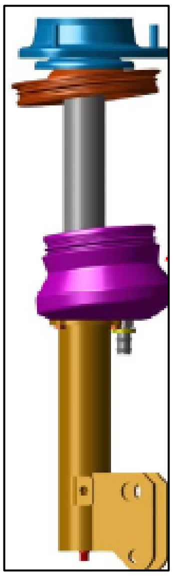

Figure 16: **MacPherson suspension with air spring** [11]

From the constructive point of view, the device consists of a rolling lobe air spring, which is characterized, as shown in Figure 17, by a cylindrical rubber sleeve sealed at one end by a “cap” and equipped with a piston at the other end, position and acceleration sensors, electric switch used for command input, computerized control unit, valve block, air compressor, desiccant and pneumatic accumulator.

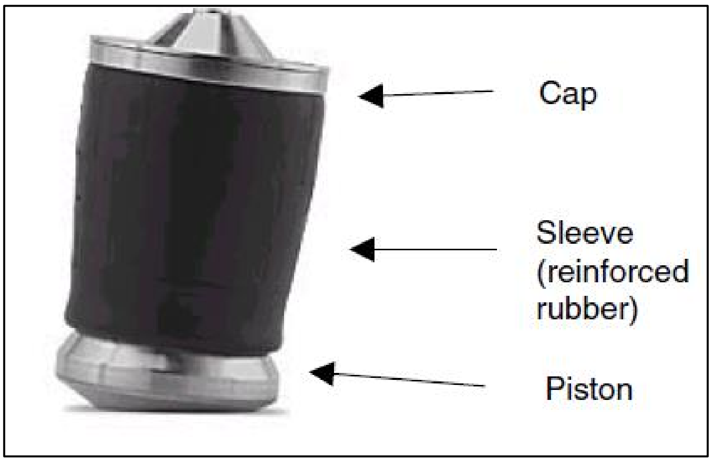

Figure 17: **Rolling lobe air spring** [11]

The ground clearance adjustment, as well spring rate one, can be made directly by the driver using a dashboard switch and selecting the wanted calibration.

The main pros of using a pneumatic height adjustment suspension system are:

- Intrinsically owning of height adjustment feature, without adding specific pneumatic components;
- Possibility of controlling, along with the ground clearance, the spring rate to obtain a good compromise between handling performance and ride comfort;
- Fast actuation in presence of stored pressurized air.

But, these solutions have also some disadvantages, such as:

- High costs related primarily to the expensive pneumatic components;
- Reduced reliability and robustness mainly due to failure of the rubber air spring;
- High maintenance requirements;
- Loss of efficiency when significant changes of the vehicle loading conditions take place or fast and frequent actuations are required (especially, when the pressurized air tank is depleted).

1.2.3 Hydro-pneumatic height adjustment system

Hydro-pneumatic suspensions use elements of both hydraulic and pneumatic systems; the hydraulic part, composed mainly by a piston-cylinder unit, a tank and an engine driven pump, is responsible for providing damping and it can be used for height adjustment feature, while pneumatic one, which consists on a high-pressure accumulator that works as air spring, is related to the elastic function.

The modification of the ground clearance is obtained with actuation directly inside the operating chambers of the hydraulic piston-cylinder unit, that behaves both as damper and linear actuator.

The first solution, schematized in Figure 18, was invented by Citroen in 1950 on DS model and nowadays is still used in luxury model with some upgrades in control logic and components. It permits to control, in addition to the vehicle height, the spring rate of the suspensions. [12]

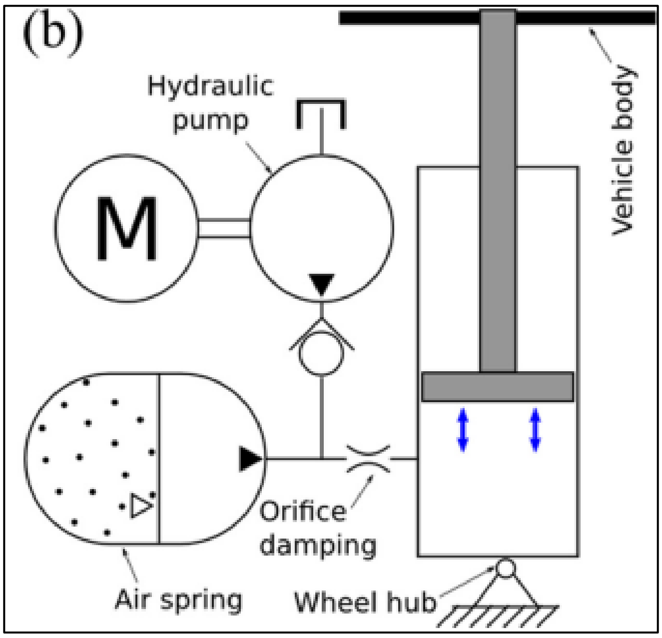

Figure 18: **Hydro-pneumatic height adjustment system** [4]

In fact, the stiffness element is a high-pressure accumulator filled with nitrogen gas (to avoid corrosion) that works as pneumatic spring; this one is attached to the end of the hydraulic actuator and has a membrane to separate gas and liquid.

Hence, changing in a controlled manner the volume of hydraulic medium inside the chambers of the actuation unit, it's possible to adjust both vehicle height and spring rate.

Hydro-pneumatic height adjustment suspension systems combine the characteristics of hydraulic and pneumatic ones, taking the pros and cons; so, the main advantages, respect to only pneumatic, are:

- Faster actuation;
- Higher height adjustment speed;
- Higher reliability and robustness of the components;
- More compact size
- Improved performance.

But, this solution has also some drawbacks, such as:

- High production costs;
- Need of specialized service of the components.

1.2.4 Electro-mechanical height adjustment system

Starting from the mid-2000s, car manufactures have attempted to find low cost height adjustment systems applicable also on low cost cars segment; so, new solutions started to be studied, especially electro-mechanical ones.

In principle, all the electro-mechanical solutions make use of linear translation part, usually by means of threaded screw-nut mechanism connected to the suspension components that are to be actuated for providing height adjustment feature. Moreover, in general, these devices are moved by an electric motor directly or through a speed reducer. In Figure 19, different architectures are schematized.

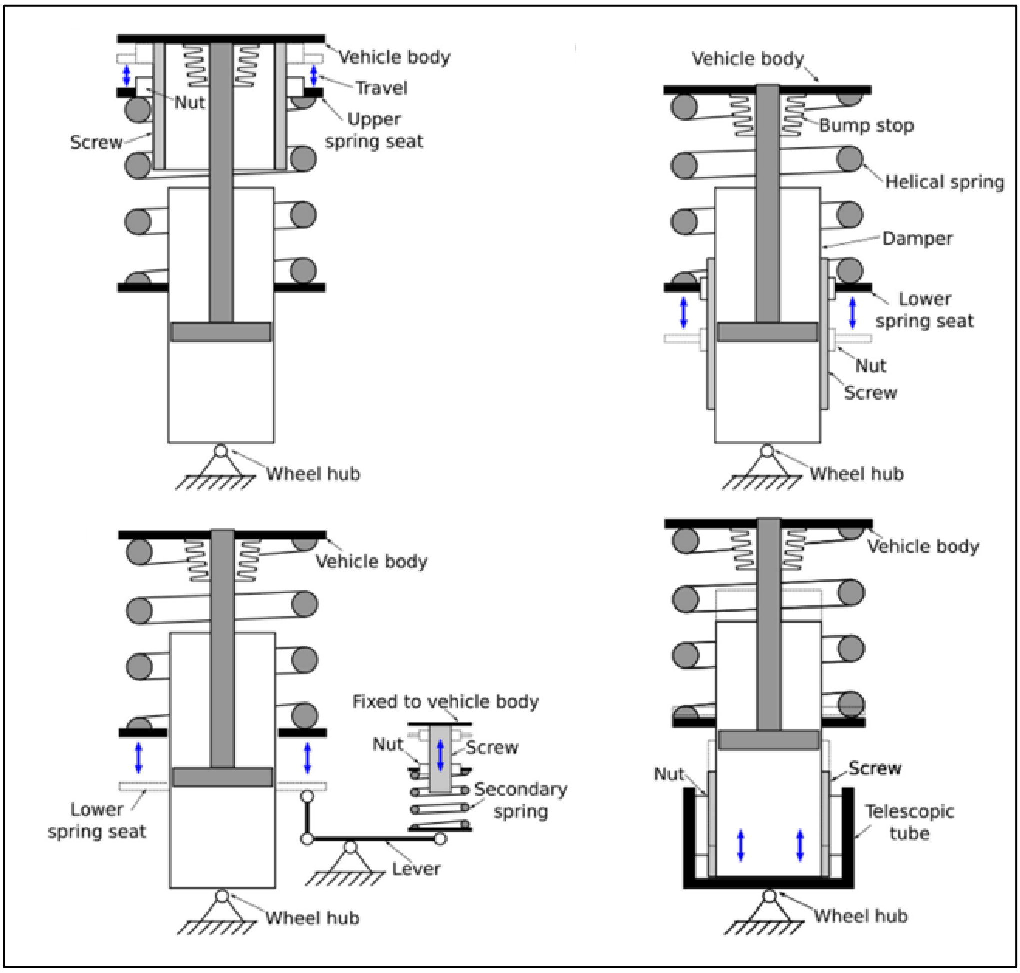

Figure 19: **Electro-mechanical height adjustment systems** [4]

Audi AG patented a solution, as shown in the following Figure 20, related to a wheel suspension, comprising a conventional shock absorber (16) with a helical compression spring (20) which surround it. [13]

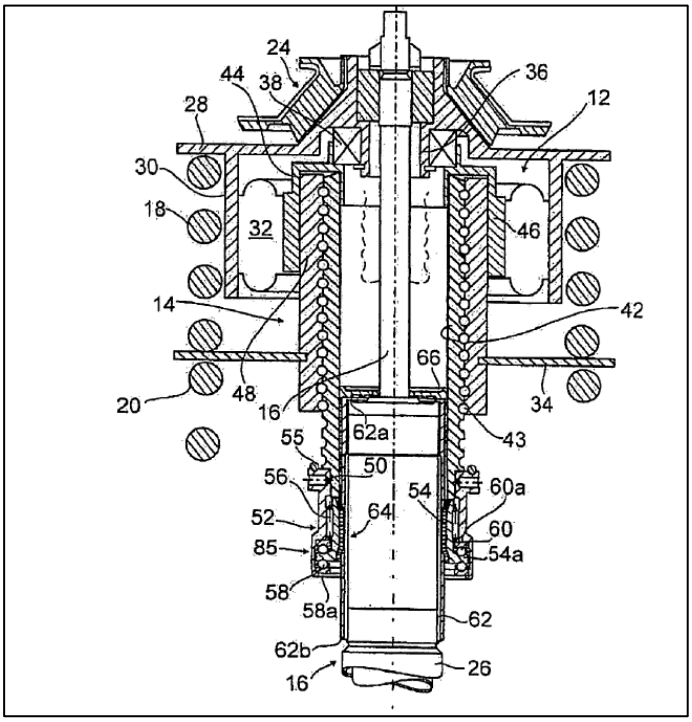

Figure 20: **Audi AG electro-mechanical height adjustment system: concentric and upper spring seat actuation** [13]

The device is essentially composed by an electric motor actuator (12) and a ball screw drive (14). In more detail, there is an axially adjustable spring plate (34), on which the aformentioned supporting spring (20) and a compensating one (18) act, linked to the ball screw nut (48).

The latter (48) is located around a hollow cylindrical threated spindle (42), that in turn is supported to a body-mounted sleeve part (36) by way of bearing arrangement.

This threated spindle (42) is driven using an ad-hoc electric motor via a flange part (44) connected to the rotor (46).

So, by turning the threated spindle (42), the ball screw nut (48) is displaced along the axis of the shock absorber; thus, the upper spring seat is actuated, changing the compression of the main spring (20) of the suspension, modifying the ground clearance. The compression spring (18) is used conventionally to reduce the actuating forces and to prevent actuator components from shock loads.

Using a reversible ball screw to increase overall efficiency, when the desired vehicle height is reached, it's necessary to lock the system to avoid overhauling due to the vehicle body weight; this function should be done by additional locking device, that are not mentioned in the patent.

Another solution, in which there is the actuation of the upper spring seat, has been patented by Honda Motor Co. [14]

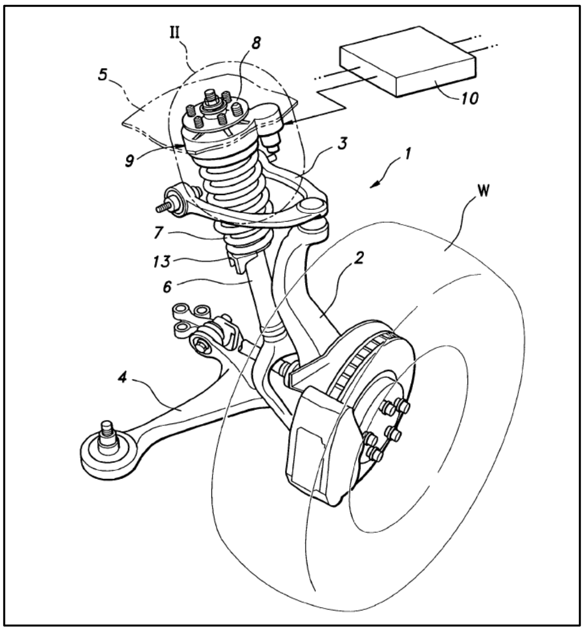

Figure 21: **Honda Motor Co. double wishbone suspension** [14]

It relates to a double wishbone suspension, that comprises a knuckle (2), which supports the wheel (W) by hub bearing, an upper (3) and a lower (4) arm, that connects the knuckle (2) to the vehicle body, a damper (6), a coil spring (7) substantially coaxial with the damper (6) and a height adjustment device (9), controlled by a drive control unit (10) and arranged between the upper end of the spring (7) and the mount plate assembly (8).

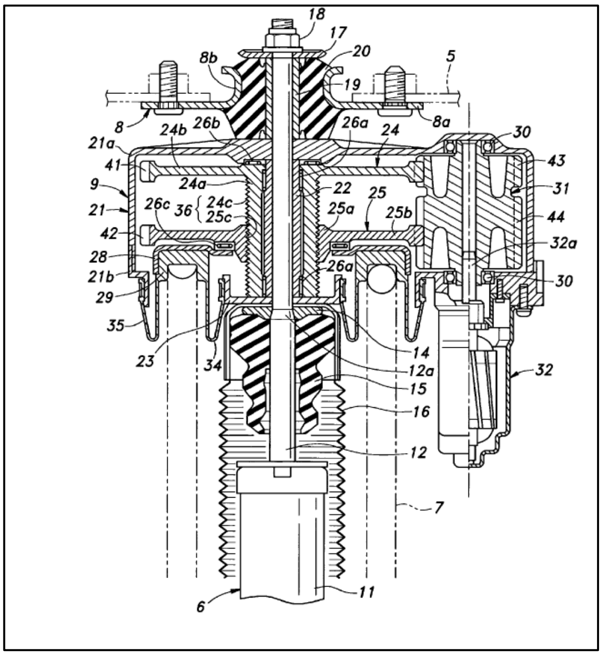

Figure 22: **Honda Motor Co. electro-mechanical height adjustment system** [14]

The height adjustment actuator (9) is composed by two coaxial and mutually rotatable rotors (24,25): the first one (24) consists of a cylindrical shaft (24a) rotatably fitted on a central collar (22) via needle bearings (26a) having a male thread (24c) on the external surface, while the second one (25) is shaped as a disk (25b) with a central boss (25a) defining a bore with a female thread (25c) and it is connected to the upper spring seat retainer by means of thrust bearings (26c). These elements (24,25) are engaged with a screw mechanism and, to providing a modification in the ground clearance, are actuated by an electric motor (32) and two spur gear connections with different transmission ratio (respectively 43-41 for the first rotor, 44-42 for the second one).

So, when the engine (32) is running, a relative rotation of the rotors (24,25) is obtained and it is converted with the threaded mechanism into a relative axial linear displacement of the second rotor respect the first one, that is axially fixed. Therefore, this produces a modification in the position of the upper spring holder (29), changing the compression of the coil spring (7) of the suspension and so the height adjustment feature is obtained. This device can be also adapted to a multi-link suspension.

A drive control unit (10), comprising an electronic control unit and a mode selection switch, controls the system on the basis of the signals derived from driver manual selection or from sensors, that detect the drive conditions.

A different solution, in which, instead, is actuated the lower spring seat, has been patented by Audi AG. [15]

Making reference to Figure 23, the invention relates to a suspension, having a shock absorber (11), with a cylinder (12) and a piston rod (13), coaxial with a helical spring (20), which is arranged between a stationary upper (22) and adjustable lower (21) spring plate.

The height adjustment unit is mainly based on an electric motor (16) with a toothed gearing (40) and a ball screw mechanism, consisting of a nut part (23), connected to the lower spring seat (21), and an inner rotatable coaxial cylinder (24), working as a screw.

In the adjustment process, the spindle part (24) is turned around the axis of symmetry (37) and so the nut (23) with the lower spring plate (21) changes its axial position, modifying the compression of the main spring (20). Moreover, it’s possible to adjust also simultaneously the position of the shock absorber cap (18), by means of a claw coupling

(34,35) and a thread connection (36/38), to avoid over-compression of the spring (20) during bumps.

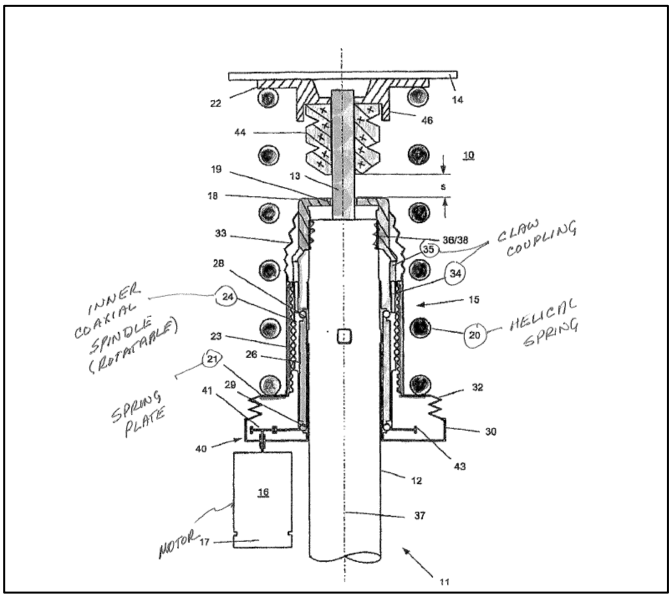

Figure 23: **Audi AG electro-mechanical height adjustment system: concentric and lower spring seat actuation** [15]

Also in this actuator, using a ball screw to improve efficiency, a locking mechanism to prevent overhauling due to the vehicle body weight is necessary (not mentioned in the patent).

Moreover, Audi AG has patented an additional solution, in which the lower spring seat is actuated by means of leverage mechanism. [16]

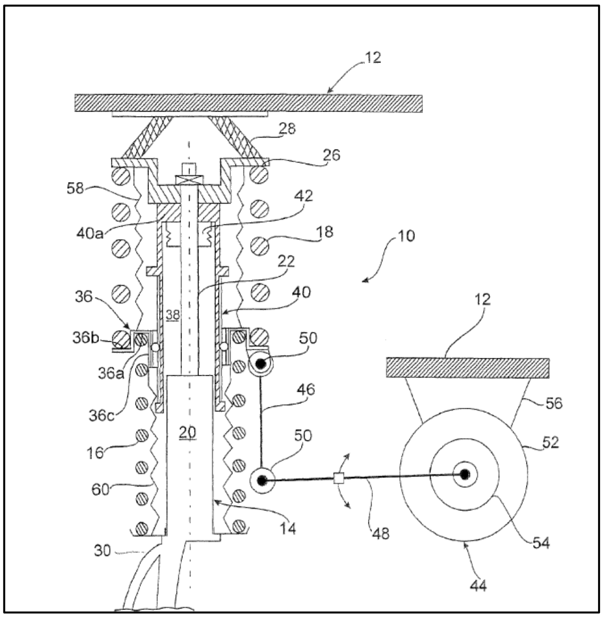

Figure 24: **Audi AG eccentric electro-mechanical height adjustment system** [16]

The suspension strut, that connects the vehicle body (12) to the wheel, is composed essentially by a telescopic shock absorber (14) (with a cylindrical tube (20) and a piston rod (22)) and two helical compression spring (one works as bearing spring (16) and the other as a preload spring (18)).

Between these last, there is an axially movable spring plate (36), that has a radially inner hub surface (36c) which is guided, by an electro-mechanical actuator (40), to move on a guide sleeve (40) via linear antifriction bearing (38) with several rows of balls.

The actuation device (40) has a hollow-shaft electric motor (52), a gearing (54) and a leverage mechanism, that is arranged radially outside the suspension on the vehicle body (12) and comprises a connection rod (46), a lever (46) and the corresponding joints (50).

Hence, pivoting the leverage mechanism, it’s possible to change the compression of the main spring (16) and then to control the ground clearance. The preload spring (18) is useful mostly to reduce the actuation forces.

To avoid overhauling due to the vehicle body weight, self-locking characteristic is obtained intrinsically in the reduction gear stage (54) on the electric motor shaft (52) or using the electric motor braking.

A different solution in which the actuation is related to the shock absorber tube has been patented by Hyundai Motor Company. [17]

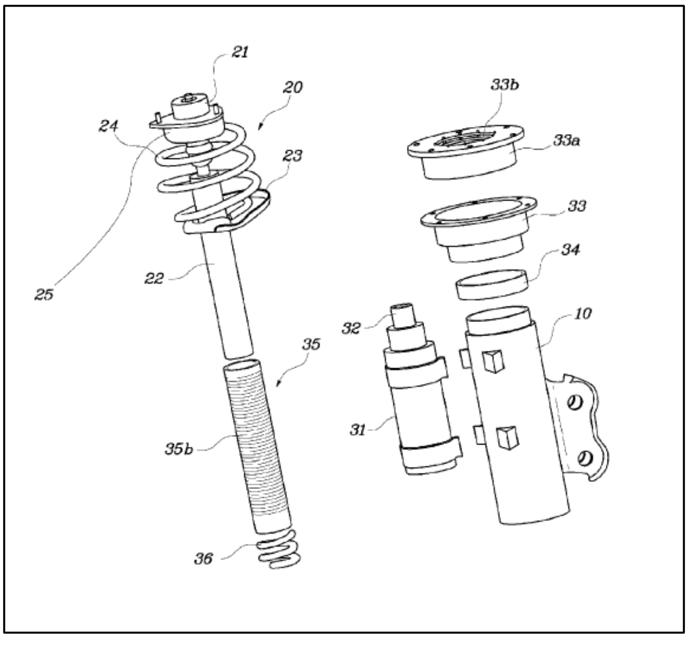

Figure 25: **Hyundai Motor Company electro-mechanical height adjustment system** [17]

The invention includes a housing (10), coupled to the wheel, a strut assembly (20), arranged to the vehicle body, and a vehicle height controller (30).

The latter (30) is composed by a motor (31), a first rotary gear (32), a lifting shaft (35), having on its outer surface a ball screw section (35b) and interference-fitted to the lower end of the shock-absorber (22) and a second rotary gear (33). This last is engaged on the outer surface to the first rotary gear (31) and on the inner one is coupled with the lifting shaft (35) by means of a ball screw section (33b).

So, when the system is actuated by the motor (31), by way of the reduction stage and a ball screw connection, the lifting shaft (35) is moved axially with the shock absorber (22), providing height adjustment.

The system is actively controlled in real time by way of a controller, that receives signals from different sensors to have info about the driving conditions.

Interpretation of how overhauling is avoided in the system is not discussed in the patent.

A similar solution has been always patented by Hyundai Motor Company, where the only difference is in the use of a grooved shaft and roller element in the groove respect of a ball screw mechanism.

Moreover, also Nissan Motor Co. has patented an electro-mechanical height adjustment system and the invention relates to a torsion beam type suspension, which includes a torsion beam and a shock-absorber arranged separately. [18]

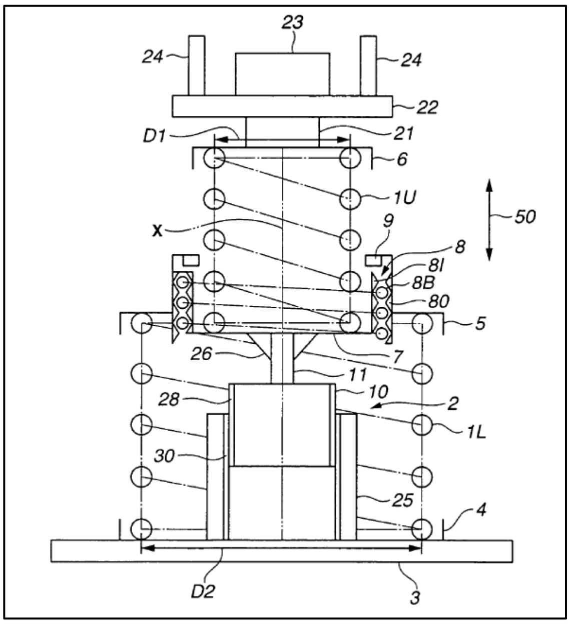

Figure 26: **Nissan Motor Co. electro-mechanical height adjustment system** [18]

The apparatus is composed mainly by an upper (1U) and a lower (1L) coil spring and an actuator (2), that is disposed within the inside space defined by the two springs (1U,1L) and operates to move relatively them (1U,1L).

The latter (2) has a motor (10) with a rotating shaft (11) connected to the lower spring seat (7) for the upper spring (1U) through reinforcing rib (26), a motor support (25) which by means of a splined connection permits the motor to move axially along the main direction (50), and a ball screw assembly (8) disposed between the upper (1U) and the lower (1L) spring.

The screw assembly (8) includes a threaded inner member (8I) mounted on the lower spring seat (7) for the upper spring (1U) and an outer one (8U) arranged on the upper spring seat (5) for the lower spring (1L).

Thus, when the motor (10) with the shaft (11) is rotating, the lower spring seat (7) for the upper spring (1U) translates along the axial direction (50), changing the compression of the two springs (1U,1L) and modifying the vehicle height.

Using a ball screw mechanism (8) to reducing energy losses, the locking device is essential but not describe in the patent.

Additional solution, in which the modification of the ground clearance is obtained actuating the lower spring seat, has been patented by Marelli Suspension System Italy. [19]

As shown in the following Figure 27, the invention relates to a suspension unit composed by a damper (10), having a cylinder (14) and a rod (16), with a cylindrical coil spring (12) fitted to it.

There is a controlled electro-mechanical adjustment device (20) interposed between the cylinder (14) and the lower spring seat (16); this comprises mainly an electric motor (22) with a speed reducer and a screw (24) - nut (26) motion conversion mechanism from rotational to linear. This last is mounted eccentrically around an axis (z’) extending parallel to the main axis (z) of the damper (10) - spring (12) unit.

The nut is carried by an upper support arm (34) and to prevent from rotating relative to it, an anti-rotational member (36) is used.

The upper support arm (34) is connected to the bottom end of a guide sleeve (38), slidable along the cylinder (14) by means of anti-friction bushes (40). The lower spring seat (16), instead, is mounted on the top end of the guide sleeve (38).

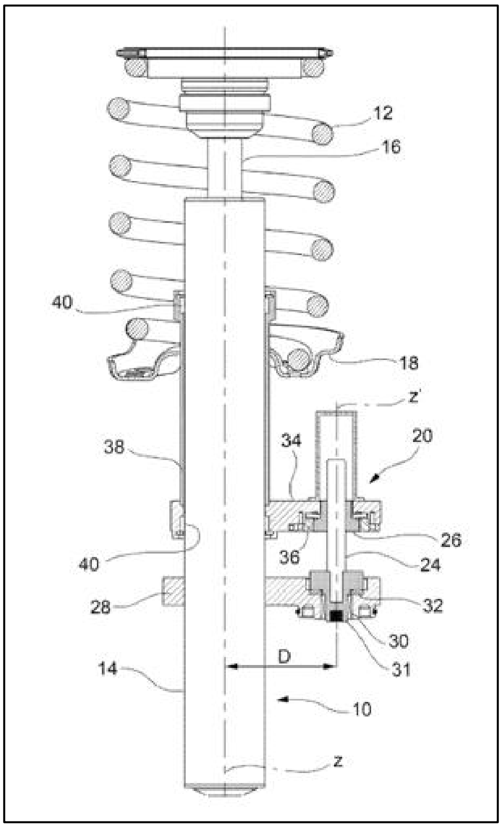

Figure 27: **Marelli Suspension System ITA electro-mechanical height adjustment system** [19]

The upper support arm (34) is connected to the bottom end of a guide sleeve (38), slidable along the cylinder (14) by means of anti-friction bushes (40). The lower spring seat (16), instead, is mounted on the top end of the guide sleeve (38).

Hence, when the motor (22) is rotating, this motion is transmitted to the screw (24) using an adapter (31), causing the linear displacement of the nut (26) and thus, by way of the upper support arm (34) and the guide sleeve (38), the lower spring seat (36) is shifted, changing the compression of the spring (12), having so the height adjustment feature.

In this solution, the conversion mechanism is irreversible and so it's not necessary a locking mechanism. To improve the overall efficiency of the system, it's possible to use a ball screw-nut connection to convert the motion from rotational to linear. Therefore, to avoid overhauling due to the vehicle weight, a locking mechanism, like an electro-magnetic clutch, is provided. [20]

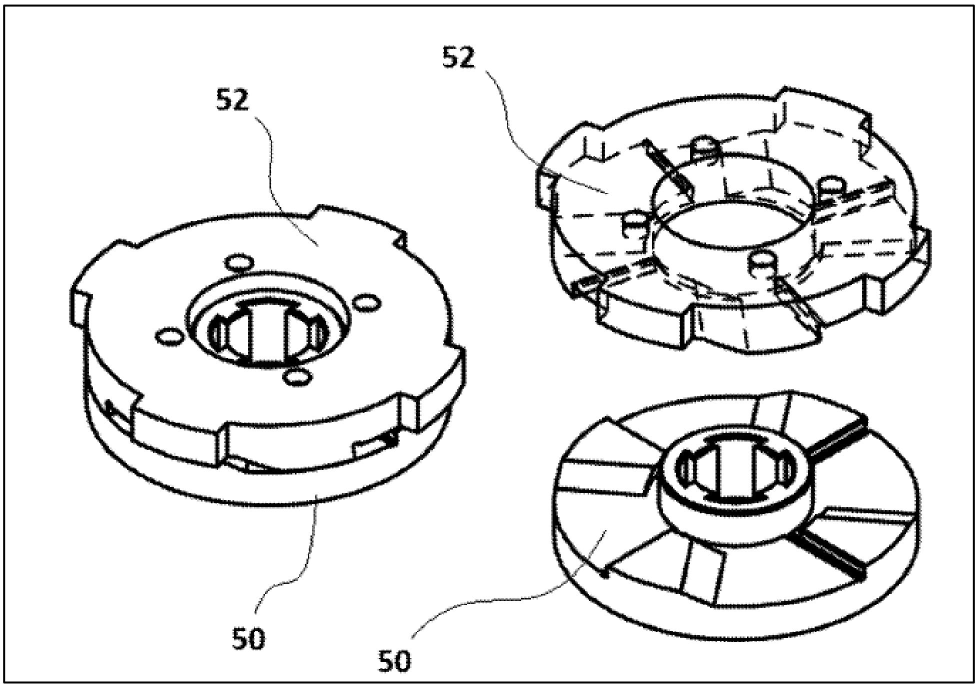

Figure 28: **Electro-mechanical clutch** [20]

This includes two discs (50,52) with profiles, which can mesh one another and it is controlled by an electromagnet (58), excited with electric current, and resilient means (56).

The lower disk (50) rotates together with the screw, while the upper disk (52) is non-rotatably fixed but is able to translate axially along the axis of the screw by means of the forces derived from the electromagnet (58) and the resilient means (56).

So, considering the relative motion of the two discs (50,52) is possible to engage or disengage the profiles, respectively locking or unlocking the system.

Electro-mechanical height adjustment systems have some advantages respect the other solutions, such as:

- Higher reliability and robustness (when failure of the system occurs, the system behaves much like traditional suspension system);
- Low costs;
- No special experience needed for the service;
- Oil free;
- Overall compact size of the components;
- Possibility of having modular design and being retrofit to the traditional suspension.

But, they have also some drawbacks, like:

- Lack of ability to vary elastic behavior;
- Lower actuation speed (if the height adjustment is required only when the driving conditions are changed, for example, from low-speed city conditions to high-speed highways or from smooth to rough terrains, the actuation speed is not a critical parameters).

Therefore, electro-mechanical solutions represent the best tread-off between high reliability, low costs and compact size.

1.3 Thesis outline

After an introduction about the environmental benefits in the use of this technology through sustainable mobility and an overview about the state of art, the structure of the dissertation in the following chapters will be:

- *Chapter 2 - Concept definition*: explanation of the motivation behind the choice of using a reversible eccentric electro-mechanical height adjustment system with upper spring seat actuation for a multi-link suspension;
- *Chapter 3 – Design and modeling of height adjustment system:* computer aided design of the main components of the actuator with detail 3D and 2D drawings along with the numerical modeling with a free body diagram approach to evaluate performances in terms of power consumption and actuation speed;
- *Chapter 4 – Experimental validation of the numerical model:* experimental data are obtained to validate the numerical model, using prototypes already realized based on a MacPherson suspension structure.
- *Chapter 6 – Conclusion and future development:* conclusion to the dissertation discussing the future steps and the possible developments of the project.

2. Concept definition

The present chapter analyzes in detail all the motivation behind the definition of the concept of the height adjustment system; before that, to fully understand, it's necessary to briefly describe the architecture of a multi-link suspension.

2.1 Multi-link suspension

The multi-link suspension has been introduced in the series production by Mercedes on 190 Model at the beginning of 1980s; before that, it has been used only in competition cars.

This solution is more complex and expensive, but it permits to achieve excellent results in terms of comfort and handling.

In the general architecture, it connects the wheel group to the vehicle body by means of five levers to control in the wanted manner the motion of it, eliminating five degrees of freedom of the six typical of a rigid body moving in the space, as can be consider the wheel. For the same functionality of the suspension system, only the vertical displacement is left free.

Moreover, it is characterized by a non-coaxial arrangement between the compressing spring and the shock absorber.

Because the aim of the present work is the design of a height adjustment system for Alfa Romeo Stelvio's rear suspension, it' necessary to underline that this vehicle model adopts on the rear axle a particular patented technology of multi-link suspension, called Alfa Link rear suspension, shown in the following Figure 29.

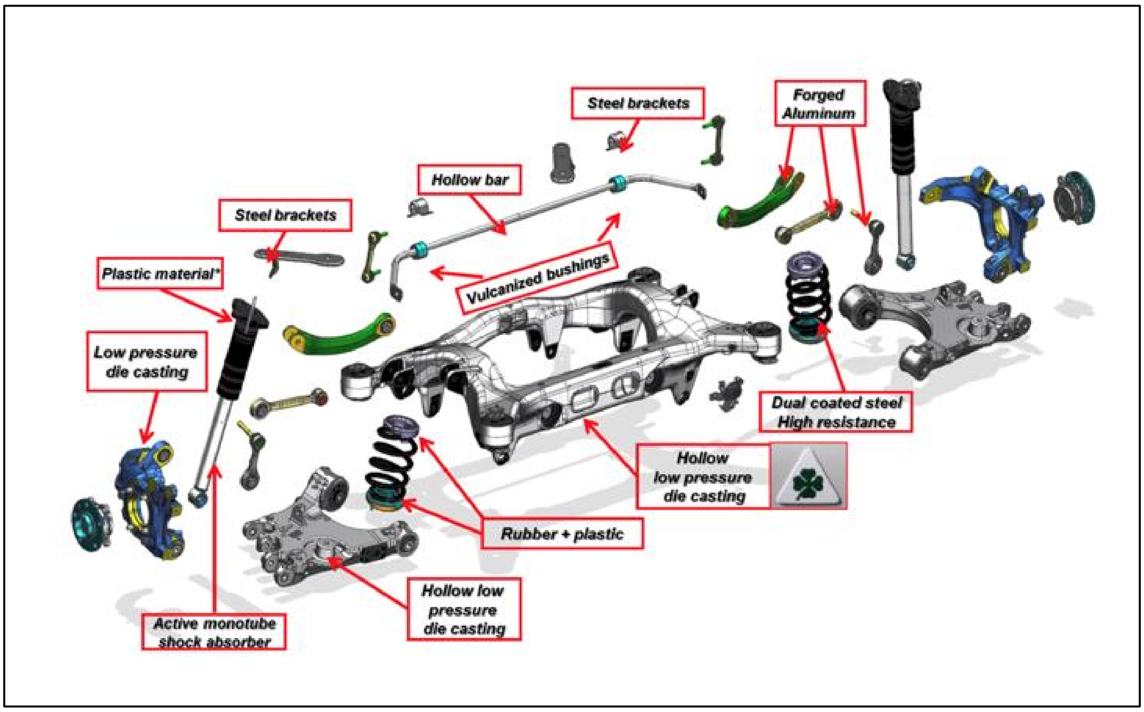

Figure 29: **Exploded view of Alfa Link rear suspension** [21]

2.2 Choice of the height adjustment system

As described in the Sub-chapter 1.2 – *State of art,* it's possible to distinguish four type of actuation mode: hydraulic, pneumatic, hydro-pneumatic and electro-mechanical.

For this application, the height adjustment is only required when the ride conditions are changed, like as for example off-road or highway situations; so, fast actuation is not a priority. Moreover, with the aim to produce the system and implement it on an existing vehicle, low costs, compactness and high reliability are important features.

Thus, the best compromise is an electro-mechanical actuator, in fact mechanical components such as threaded screws-nuts and gear trains offer high reliability and reasonable responsiveness, while the possibility of using commercial electric motors for automotive applications could lead to an acceptable balance between cost and benefit.

As can be seen from the different patents presented previously in the Section 1.2.4 – *Electro-mechanical height adjustment system*, the electro-mechanical height adjustment suspension systems primarily work on the modification of the compression of the supporting spring, actuating the lower or upper spring plate to change ground clearance.

These inventions can be classified into two main categories:

- Concentric solution;

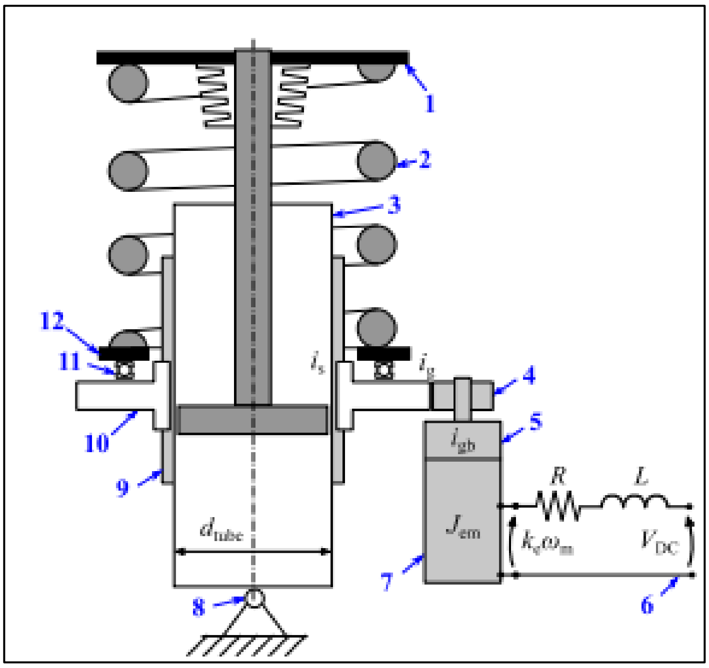

Figure 30: **Electro-mechanical concentric height adjustment system** [22]

- Eccentric solution.

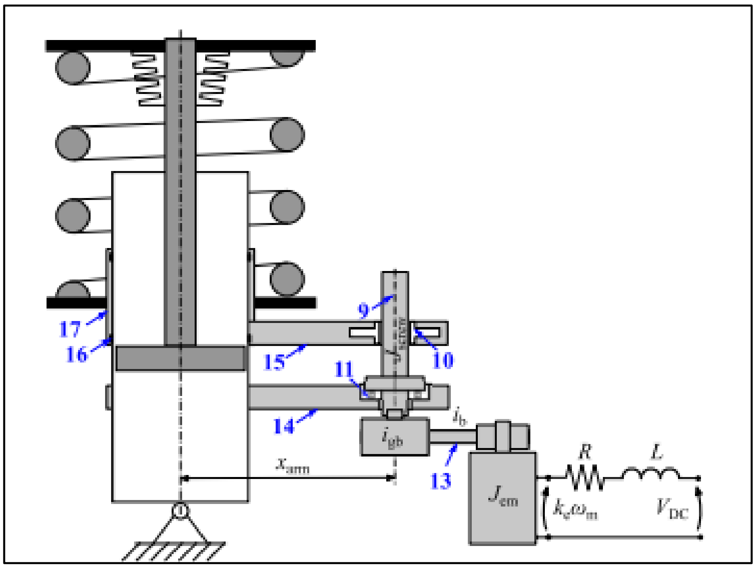

Figure 31: **Electro-mechanical eccentric height adjustment system** [22]

The concentric solution is characterized by the same axis between the screw-nut mechanism and the main spring.

In a typology of suspension in which the spring-damper unit is coaxial, as MacPherson, the cylinder tube of the damper is used to fit the power screw around it, while in a non-coaxial arrangement, like multi-link, it's necessary to introduce a specific guiding element.

The main advantage of this architecture is that it permits to utilize the space inside the compressing spring, by limiting the radial direction bulk; however, there are also some drawbacks, such as:

- The sizing of the screw and the nut is dependent by the external diameter of the cylinder of the shock absorber or the internal guide, in fact:

  $$d_{min} = d_{tube} + s \tag{2}$$

  where s is the thickness of the screw and depend on the mechanical strength and fatigue life calculation.

  Moreover, increasing the diameter brings to a reduction of motion conversion efficiency in lifting and lowering phase:

  $$\eta_{s,lift} = \frac{tan\lambda}{\tan(\lambda + \phi)} \tag{3}$$

  $$\eta_{s,lower} = \frac{tan\lambda}{\tan(\lambda - \phi)} \tag{4}$$

  where ϕ is the friction angle ($\phi = arctan(\frac{f_s}{cos\alpha})$, f_s is the friction coefficient between the threads and α is the thread angle) and λ is the screw lead angle ($tan\lambda = \frac{p}{\pi * d}$).

- Need of an anti-rotational device to prevent unwanted rotative motion of the spring plate actuated by the nut, which rotates and translates; this element increases the power loss via friction.

Instead, in the eccentric solution, there is a certain distance between the axis of the spring and that of the screw-nut mechanism; so, the actuator bulk in the radial direction is increased. However, there are a series of advantages, like as:

- The size of the diameter of the screw and the nut is minimized, based only on mechanical strength and fatigue life calculation; this allows to have better motion conversion efficiency.
- Modularity, which means that the same sizing of the power screw and nut may be applied to several solutions.
- The eccentricity gives intrinsically spring plate anti-rotation feature.

Moreover, in terms of construction simplicity and cost, the eccentric architecture has several pros respect to the concentric counterpart:

- Smaller volume of the screw, which makes the substantial part of the system cost;
- The sliding surface of the shock absorber tube does not need a strict tolerance;
- Fewer and smaller bearings, replaceable by bronze bushings;
- Ease of maintenance.

By applying these considerations to the Alfa Link rear suspension, shown in Figure 26, the eccentric height adjustment system is preferable; in fact, despite the increment in radial direction bulk, it permits to achieve overall better performances and the best tread-off between cost and benefit. The results of the two solutions are summarized in Table 2.

Performances	Concentric solution (d=27, p=8) DC electric motor 48V	Eccentric solution (d=15, p=3) DC electric motor 12V
Current to lift – [A]	15.85	15.46
Current to lower – [A]	6.29	4.83
Lifting speed – [mm/s]	2.04	2.03
Lowering speed – [mm/s]	2.86	3.28
Power to lift – [W]	190	185
Power to lower – [W]	75	58
Lifting efficiency - [%]	1.5	6.1
Lowering efficiency - [%]	5.3	31.6

Table 2: **Comparison between concentric and eccentric solution**

For the diameter and the pitch of the screw-nut mechanism, values that ensure to guarantee the appropriate mechanical resistance minimizing the power consumption and keeping an actuation speed reasonable have been used. It’s important to highlight that in the concentric solution, DC 48V electric motor must be used to manage to lift the system, greatly increasing the costs.

At this point, due to configuration of the suspension unit, for the ease of implementation, being able to exploit a substantial free zone from other vehicle components, it has been opted for an upper spring seat actuation

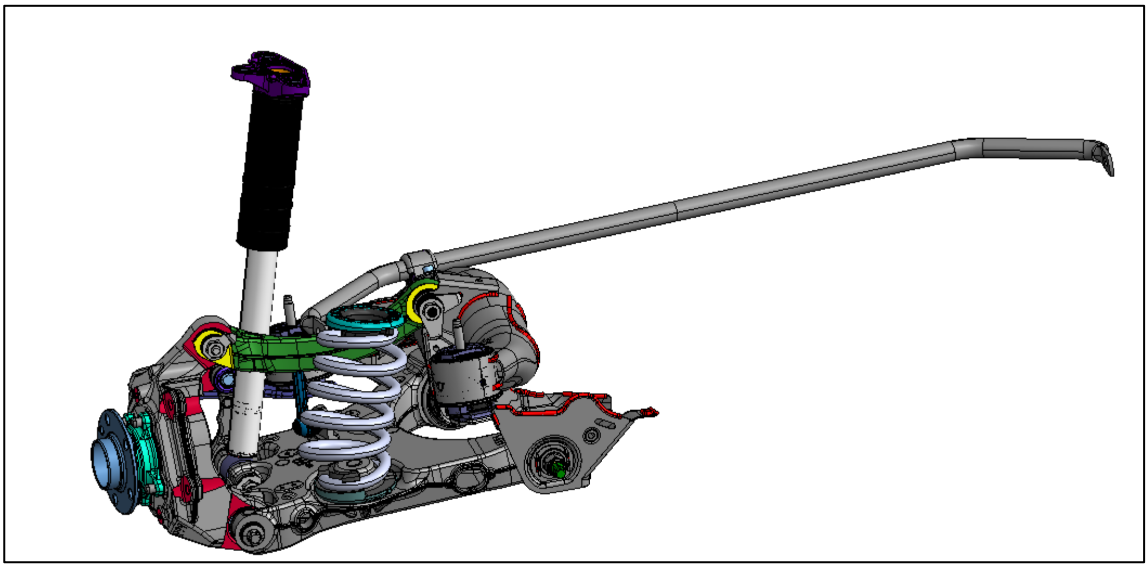

Figure 32: **Alfa Link rear suspension**

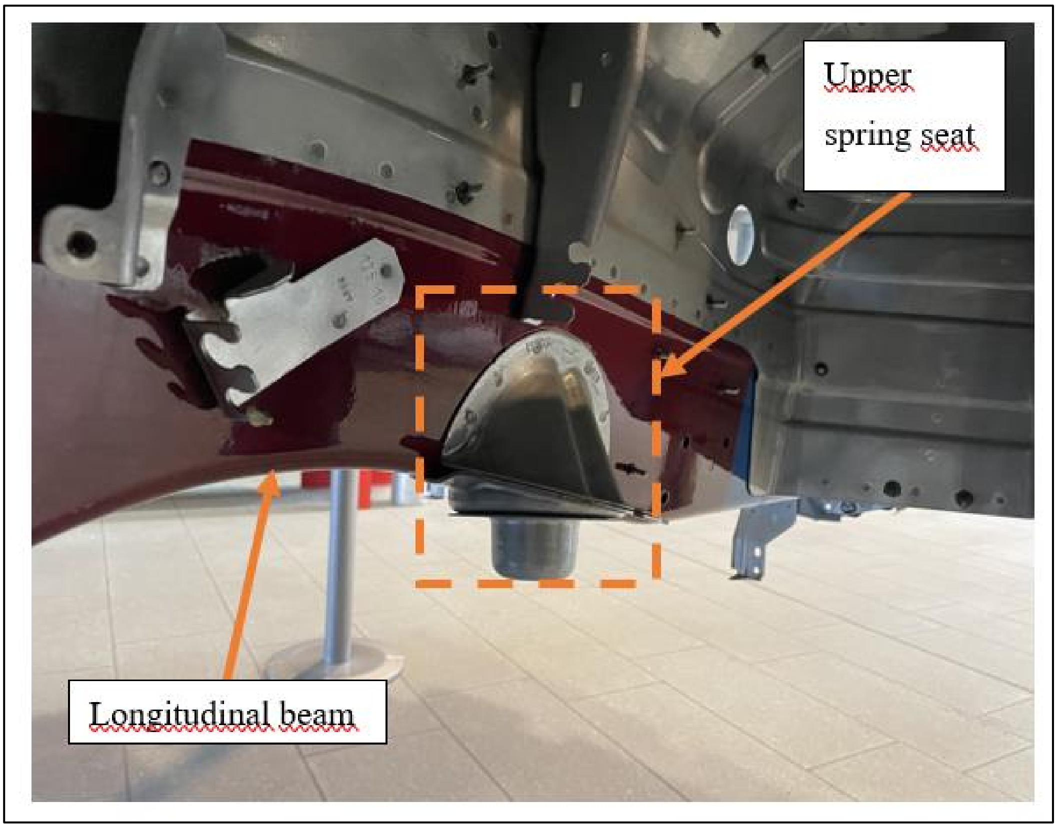

Figure 33: **Alfa Romeo Stelvio underbody**

For space problem, it becomes necessary to shift the position of the upper spring holder of 60 mm toward the y axes of the vehicle without changing the spring length, as depicted in Figure 34; it causes a modification in the spring inclination, without significant changing in the suspension kinematics.

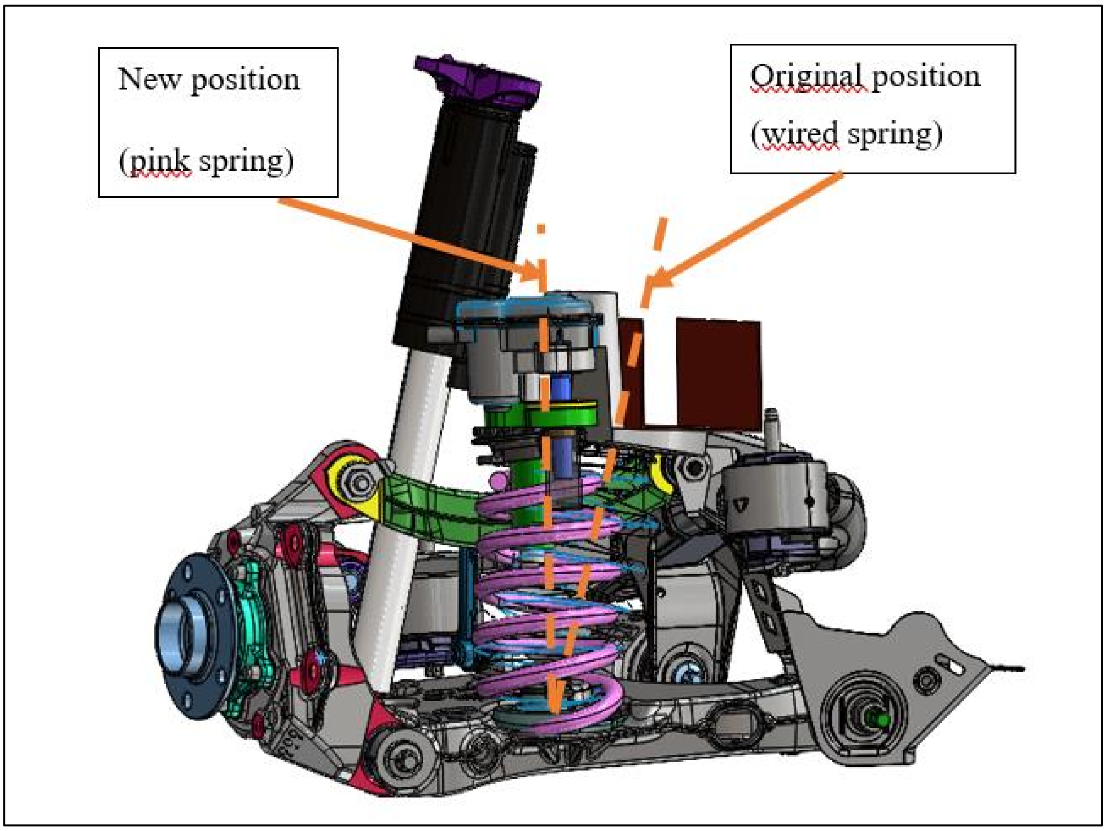

Figure 34: **Modification on upper spring seat position**

Up to now, irreversible motion conversion mechanisms are taken into account. It is characterized by high friction on the threads of the screw and the nut and this ensures that the system is not able to move solely due to vehicle weight.

However, some drawbacks are correlated, as:

- High electric energy consumption
- Long actuation time
- High wear of the screw's and nut's threads.

As described in Section 1.2.4 - *Electro-mechanical height adjustment system*, it possible to adopt also a reversible solution using a ball screw-nut motion conversion mechanism to improve global performances. These last are summarized in the following Table 3.

Performances	Reversible eccentric solution (d=16, p=4)	Relative percentage improvement respect irreversible eccentric solution
Current to lift – [A]	4.97	68%
Current to lower – [A]	2.35	51%
Lifting speed – [mm/s]	4.36	115%
Lowering speed – [mm/s]	5.53	69%
Power to lift – [W]	60	68%
Power to lower – [W]	28	52%

Table 3: **Reversible solution performances**

It should be noted that during the lowering phase, in the reversible solution, there may be regeneration.

Moreover, the usage of the latter increments the complexity of the actuator because a locking device must be added to avoid overhauling due to the vehicle weight; despite this, the reversible height adjustment system represents the best compromise.

As blocking component, an electro-magnet clutch has been chosen, according to the patent "Shock absorber and spring assembly for a vehicle suspension, comprising an electro-mechanical adjustment device for adjusting the vertical position of the spring and a relative blocking system, and a method for adjusting the vertical position of the spring". [20]

3. Design and Modeling of the height adjustment system

In this chapter, a detailed description of the designed reversible eccentric electro-mechanical height adjustment system is present along with the sizing of the main structural components.

In addition, the approach behind the development of the numerical model is explained.

3.1 Description and working principle

The reversible eccentric electro-mechanical height adjustment system, making reference to the exploded view in Figure 35, is composed by:

- 1: Upper spring seat;
- 2: Ball screw 16 x 4 mm;
- 3: Flanged nut with 4 internal circuits;
- 4: Anti-rotational device;
- 5: Screw adapter;
- 6: DC 12 V electric motor with reduction stage;
- 7: Electro-magnetic clutch;
- 8: Sliding element;
- 9: Support;
- 10: Rubber cushion;
- 11: O-ring

- 12: Sliding element bushings;
- 13: Magnet connector;
- 14: Retaining ring;
- 15: Screw bushing;
- 16: Screw dust boot.

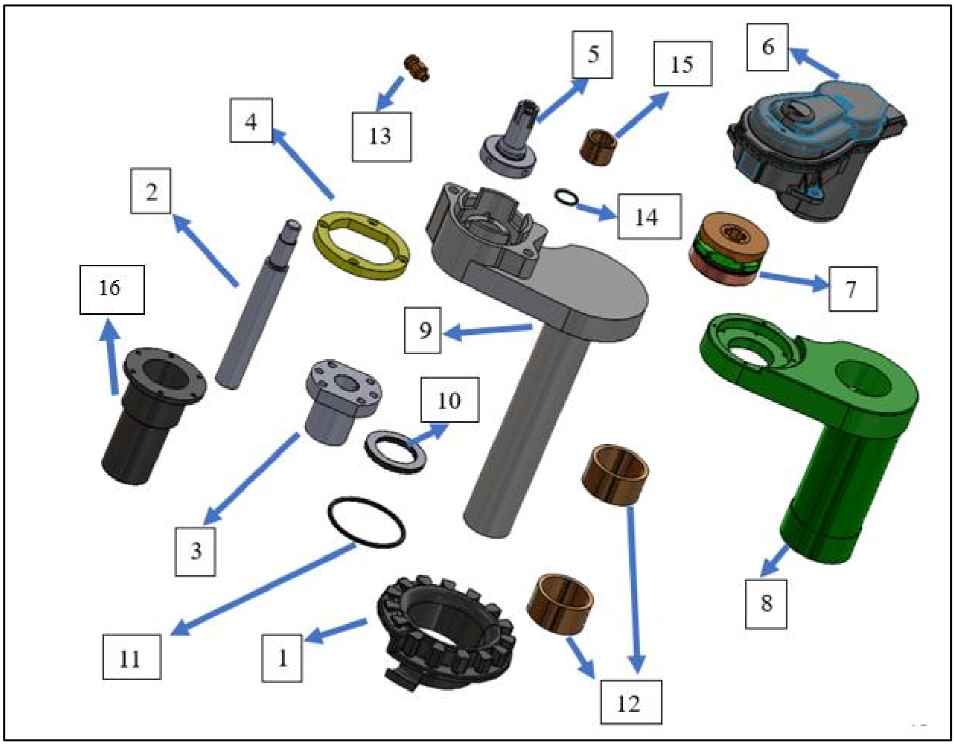

Figure 35: **Height adjustment system exploded view**

In the following Figure 36 and 37, respectively a 3D view of the actuator and its implementation on Alfa Link rear suspension are shown.

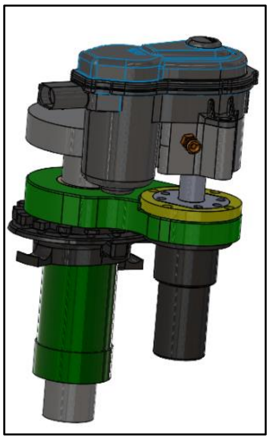

Figure 36: **Actuator 3D view**

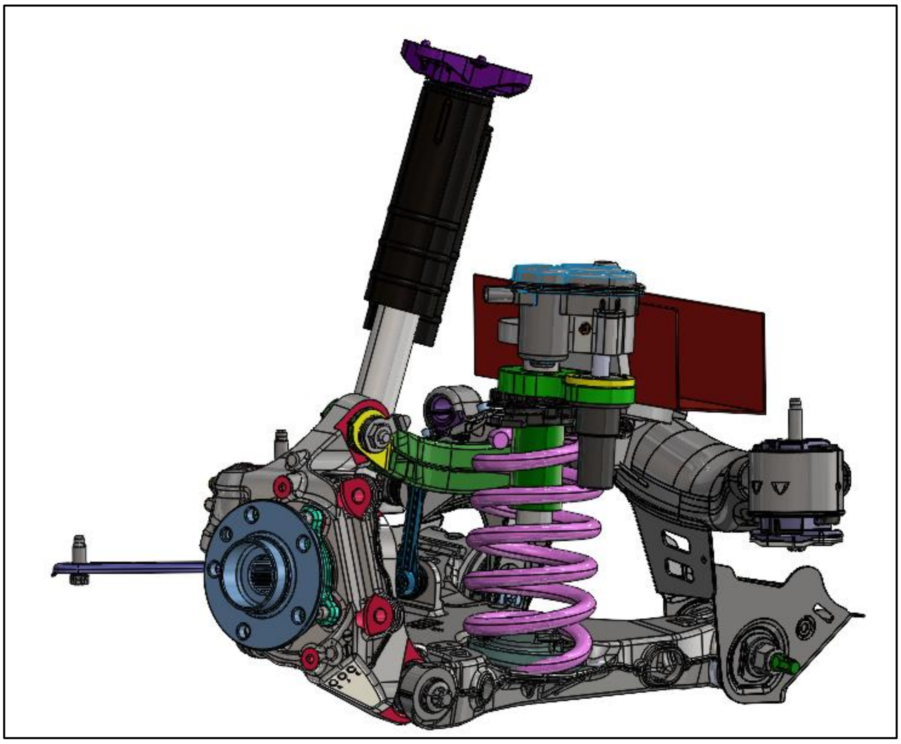

Figure 37: **Implementation 3D view**

Moreover, Figure 38 represent the 3D cross section view of the height adjustment system.

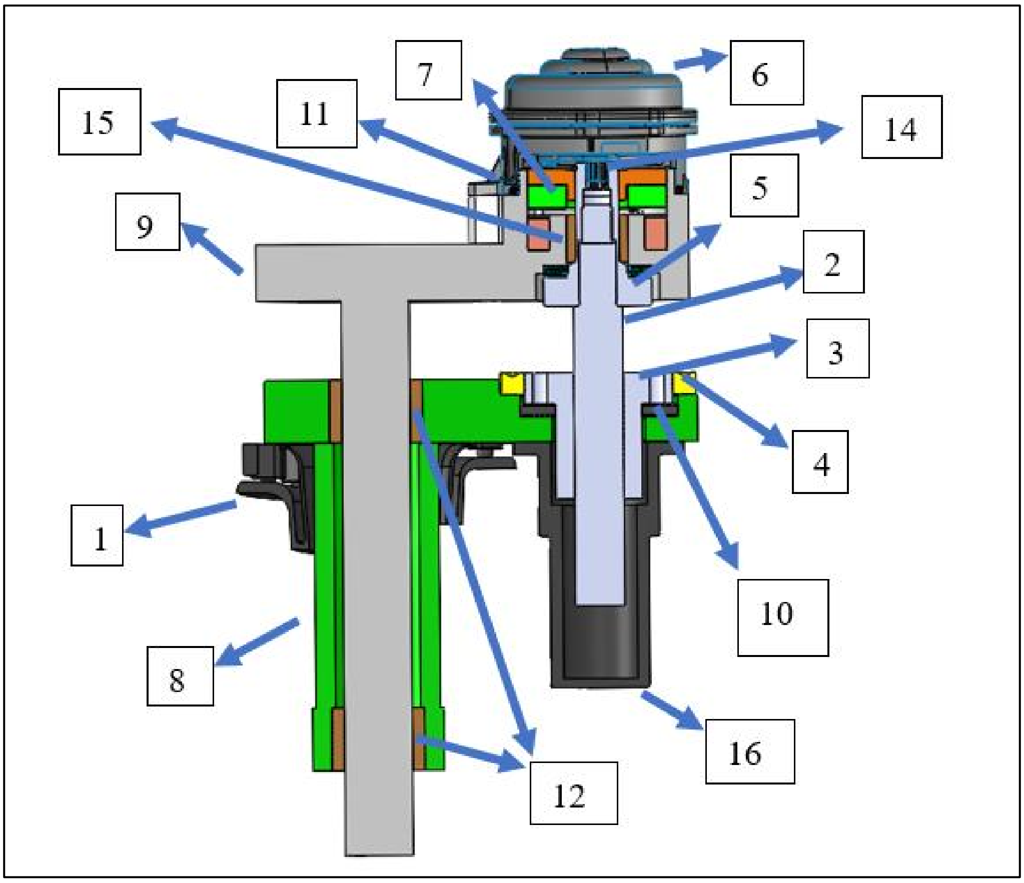

Figure 38: **Cross section 3D view**

The aim is to change the vehicle height controlling the position of the upper spring seat (1).

The device is actuated by an electric motor coupled with a mechanical speed reducer (6), that drives the ball screw (2) by means of a screw adapter (5) and a retaining ring (14); the power is supplied by an on-vehicle battery.

The chosen electric motor-reducer is the Johnson Electric HC685LG – 120; it is a DC 12 V TRW engine and permits to obtain performances, summarized in Figure 39, and reliability comply with the application with moderate cost and good compactness. [23]

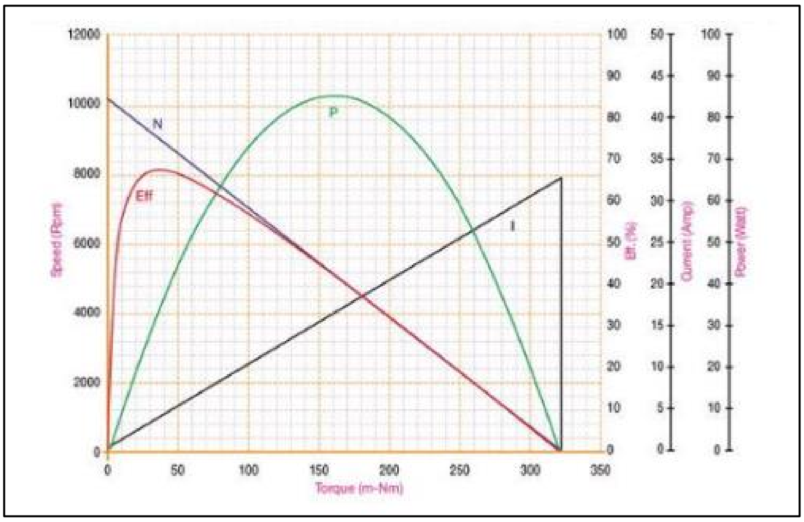

Figure 39: **Johnson Electric HC685LG – 120 performance curves** [23]

The screw rotational motion is transformed into linear by means of the nut (3), mounted on the arm of the sliding element (8). For the correct functioning of the actuator, it's necessary prevent the rotation of the latter (3), but at the same time ensure a slight flotation. Hence, to avoid the reworking of the nut (3), a flanged type, as Figure 40, is selected and an appropriate anti-rotational device is designed.

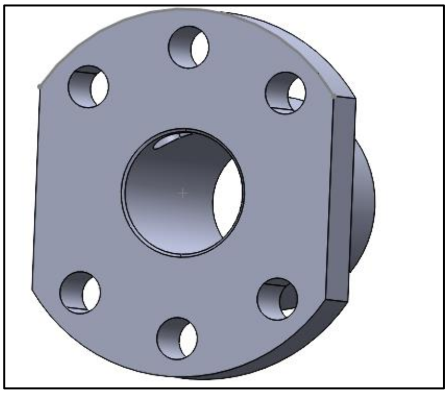

Figure 40: **Flanged nut**

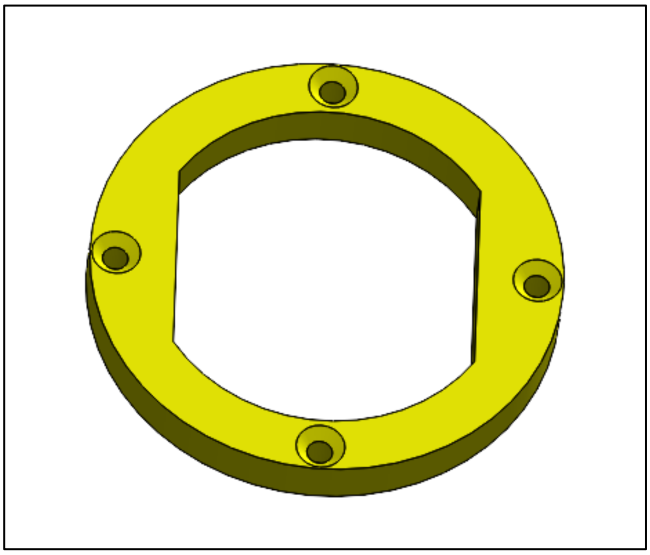

Figure 41: **Anti-rotational device**

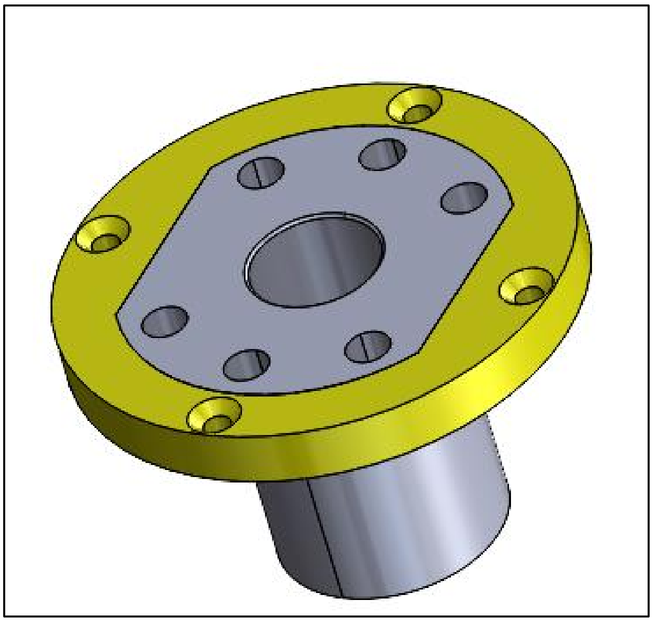

Figure 42: **Anti-rotational - nut device coupling**

To avoid functionality problems related to the misalignment, a rubber cushion (10) connects the nut (3) with the sliding element (8).

This last (8), in turn, can slide on the support (9) by means of two bushings (12) and is coupled with the upper spring plate (1). So, changing its position, the compression of the coil spring is modified and consequently the ground clearance.

As anticipated in the Sub-chapter 2.2 - *Choice of the height adjustment system*, choosing a reversible solution, a locking device is necessary; so, an electro-magnetic clutch (7) is introduced in the apparatus. An assembled 3D and an exploded view are shown is Figure 43 and 44.

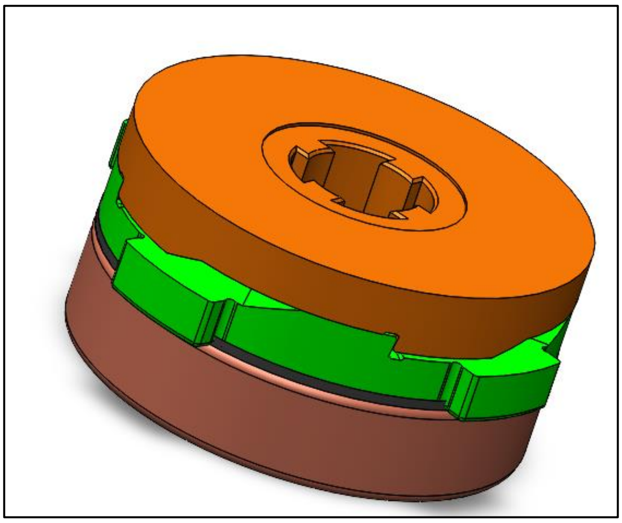

Figure 43: **Electro-magnetic clutch**

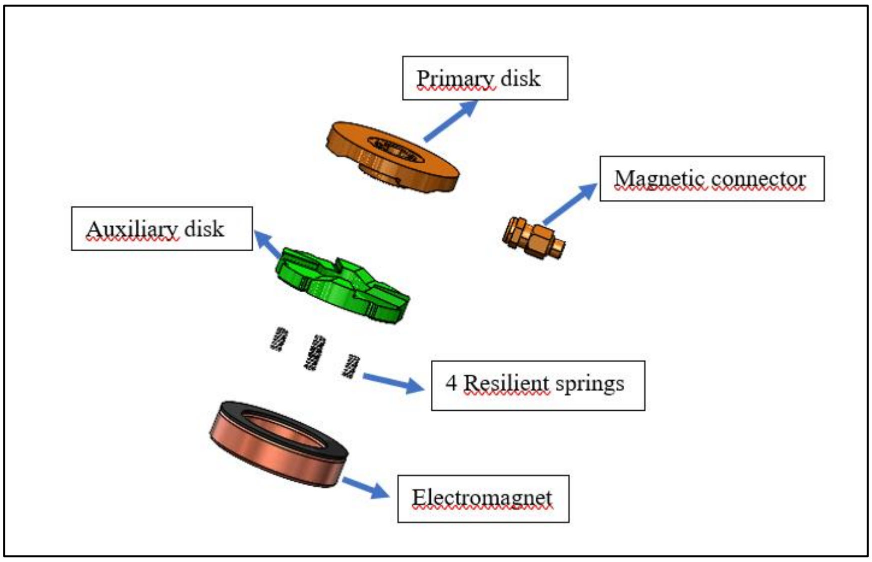

Figure 44: **Electro-mechanical clutch exploded view**

It is housed in a specific seat in the arm of the support (9) and consists of:

- a primary and auxiliary disk, characterized by a series of teeth with profiles that may engage one another.
- an electromagnet adapted to attract the auxiliary disk toward unblocking position by means of electrical excitation;
- four resilient springs adapted to push the auxiliary disk toward the blocking position;
- a magnetic connector.

The primary disk rotates together with the power screw (2), while the auxiliary one is not-rotatably fitted on the support (9) but is able to move axially along the screw (7) axis. Their profiles form different angles on the right-hand/first side and the left-hand/second side with respect to the base plane.

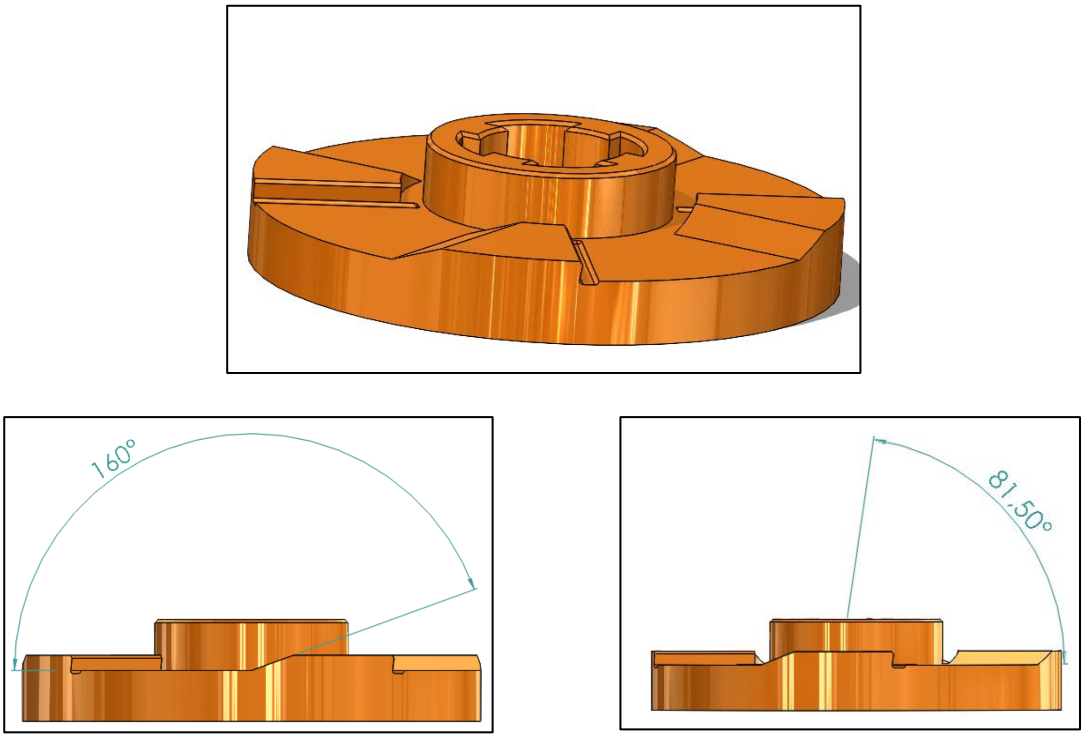

Figure 45: **Primary disk**

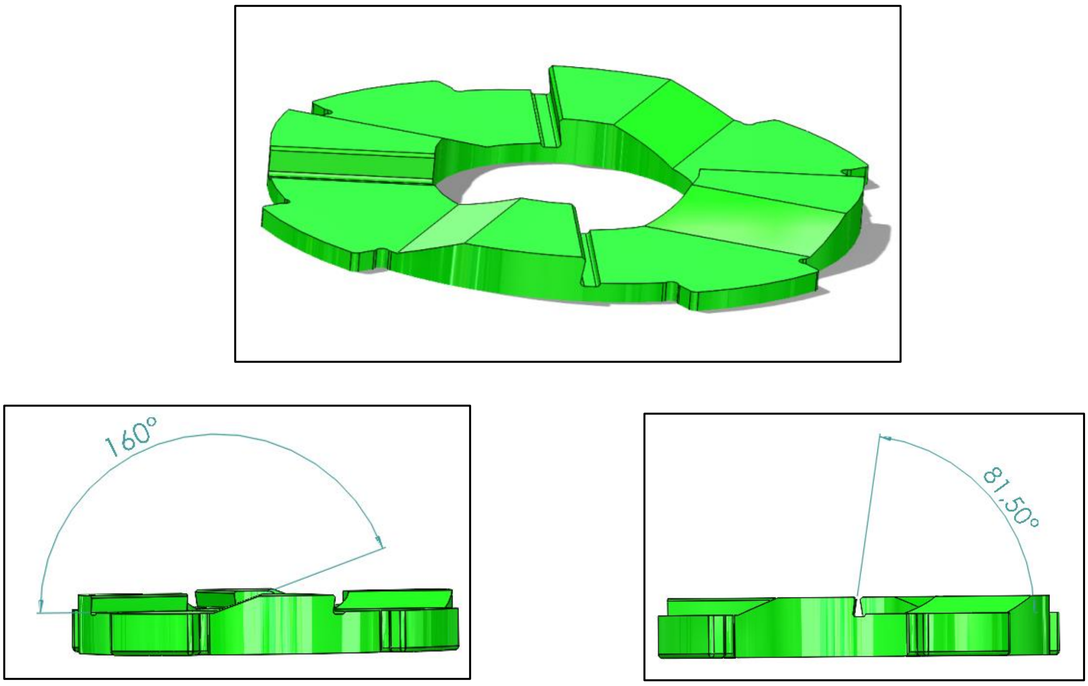

Figure 46: **Auxiliary disk**

In particular, when the screw (2) tends to rotate so as to cause a descending motion of the nut (3), the first side with smaller angle (81,50°) enters in engagement, not allowing a relative sliding between the profiles of the two disks, guarantee the blocking condition. Instead, the second side with greater angle (160°) is coupled in the ascending displacement and ensures the unblocking configuration, in which the mutual sliding of the disks is permitted. In addition, this last condition makes it possible to impart an initial course to the auxiliary disk along the axis of the screw-nut mechanism to be closer to the electromagnet, reducing the required energy to the disengagement of the clutch (7).

So, starting from the blocking setting, the steps for the upward and the downward movement are described in detail.

For the first one, the electric motor (6) is put in rotation in the direction that brings the teeth in conjunction on the left-hand side; consecutively the electromagnet is excited, attracting the auxiliary disk, which translates along the screw (7) axis, provoking the disengagement of the clutch. So, now the mechanism is unblocked. Hence, continuing to kept excited the electromagnet and the same rotational direction of the engine (6), the upper spring seat (1) can ascend. Once it has reached the desired position, the motor (6) is de-energized and the electromagnet is de-excited; thus, by means of the resilient springs, the two disk are brought back into contact and then, due to the vehicle weight, the right-hand side profiles come into engagement, blocking the system.

Instead, for the opposite motion, the only difference is that after the disengagement of the clutch, the electromagnet keeps to be excited while the engine (6) is stopped; so that under the only action of the vehicle weight, the upper spring holder (1) goes down to the wanted location.

The electro-magnetic clutch is controlled with a pick and hold strategy.

Moreover, given that the electromagnet is configurate to operate on the principle of the reluctance, it possible to estimate the position of the auxiliary disk respect to the primary one by means of measuring the inductance of the magnetic circuit.

3.2 Numerical modeling

The numerical model of the system under study has been developed; it permits to evaluate performances and understand the forces exchanged between the different components during actuation.

It has been obtained using a free body diagram approach, following the paper "Modeling and Experimental Validation of the Performance of Electromechanical Height Adjustment Vehicle Suspension with Eccentric Mounted Screw System" of Ruzimov et al. [22]

The following Figure 47 and 48 show respectively the free body diagram of the sliding element in lowering and lifting phase.

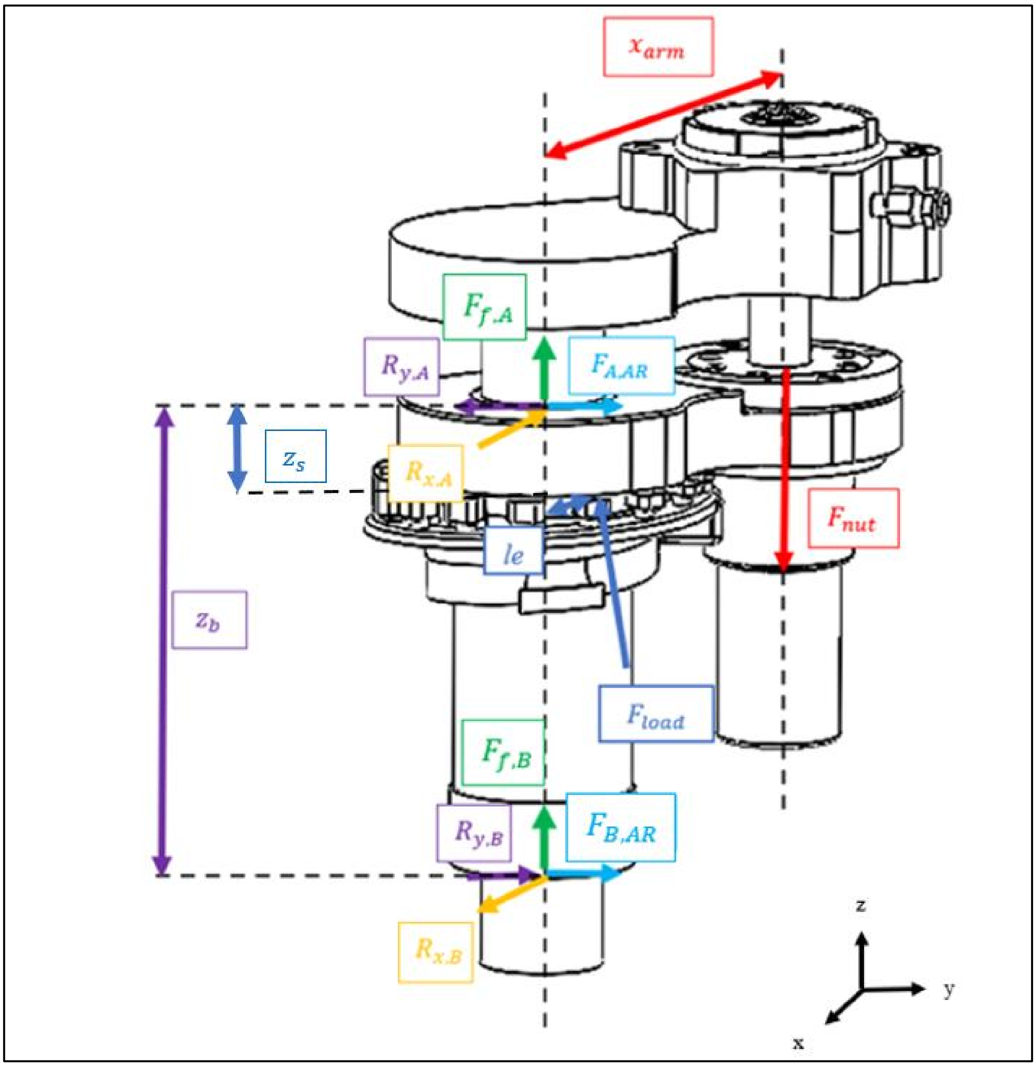

Figure 47: **Sliding element's free body diagram in lifting phase**

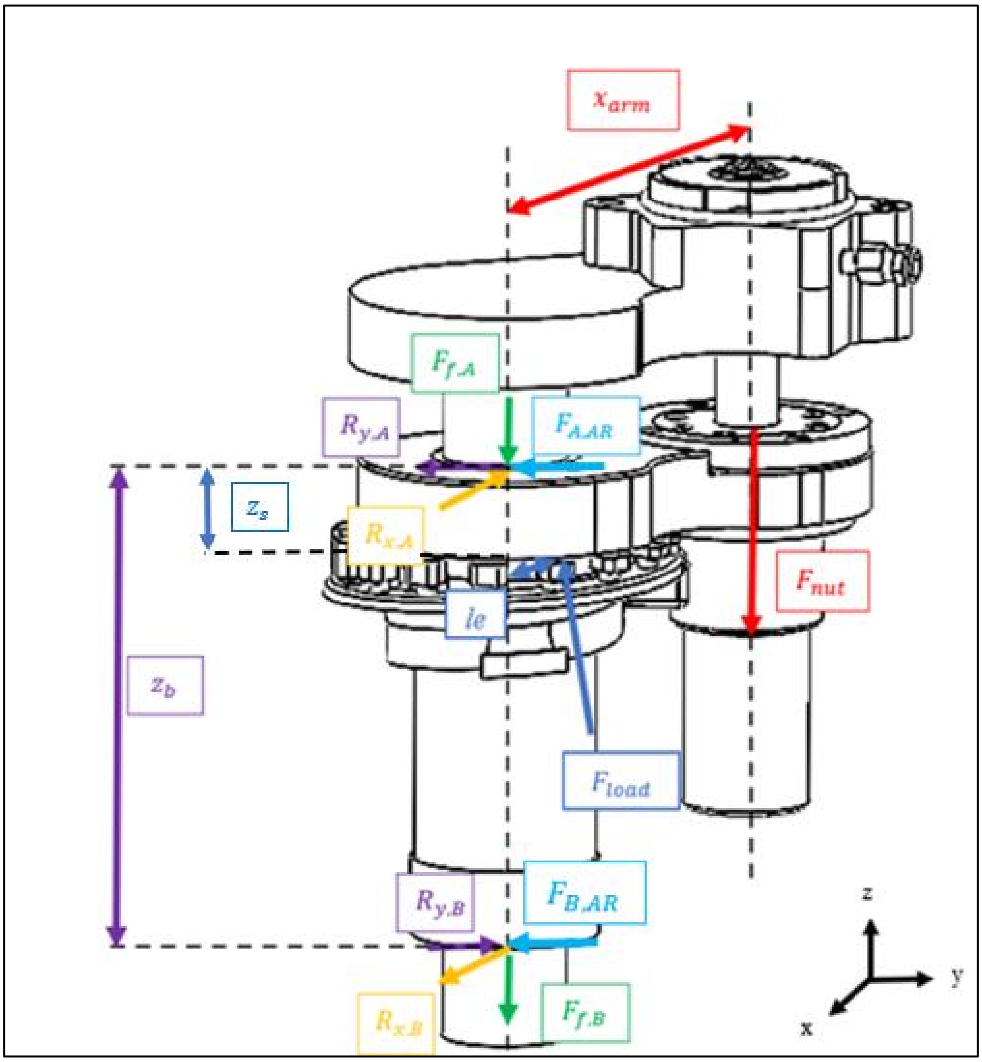

Figure 48: **Sliding element's free body diagram in lowering phase**

In the modeling process, the reference frame is chosen such that the Z axis coincides with the screw axis and the X axis coincides with the line connecting the screw and sliding element axes; the direction of the Y axis is obtained as a result and this lays on the actuator mid-plane, which represented the YZ plane.

Points A and B are located respectively at the upper and lower plane of the two bushings on the center axis of the sliding element.

The forces acting on the height adjustment device are:

- F_{load}, that is generated by the compression of the coil spring due to the vehicle sprung mass weight and it is applied to the upper spring holder; because of how the spring is implemented in the Alfa Link rear suspension, F_{load} have a certain inclination and offset le (is not in any reference planes) with respect to the sliding element axis, providing a bending moment in the XZ and YZ planes;

- F_{nut}, which is developed at the power screw to lift or lower the nut; in an eccentric solution, it has an arm x_{arm} respect to sliding element axis;
- $R_{x,A}$ and $R_{x,B}$, that are the reaction forces at the two bushings along the X axis, caused by the compressed spring load and the eccentricity of the screw-nut mechanism in XZ plane; in the model the assumption $R_{x,A} = R_{x,B}$ is made;
- $R_{y,A}$ and $R_{y,B}$, which represent the reaction forces at the bushings along the Y axis, caused only by the spring load in YZ plane; in the modelling, the hypothesis $R_{y,A} = R_{y,B}$ is done;
- $F_{f,A}$ and $F_{f,B}$, that consists in the friction forces at the two bushings due to resulting reactions R_A ($R_A = \sqrt{R_{x,A} + R_{y,A}}$) and R_B ($R_B = \sqrt{R_{x,B} + R_{y,B}}$), where:

$$F_{f,A} = f_b * R_A \tag{5}$$

$$F_{f,B} = f_b * R_B \tag{6}$$

  ($f_b = 0.25$ is the sliding friction coefficient between bronze bushings and the steel support element);
- $F_{AR,A}$ and $F_{AR,B}$, which are additional reaction forces, derivated from the balancing of the torque on the nut by way of the anti-rotational system. The arm is equal to the distance x_{arm}, which is relatively long; therefore, the friction forces along the Z axis caused by these ones have a small influence on the overall losses and so they can be neglected.

To evaluate performances, equilibrium equations of the sliding elements are written; the movement of the latter has been hypothesized as a uniform linear motion, so the inertia forces can be not considered.

The equilibrium of the moments in the YZ and XZ plane around point A permits to calculate the reaction on the bushings R_A and R_B:

$$R_y = \frac{(F_{load} * \sin\gamma_{YZ}) * (z_s \pm z) - (F_{load} * \cos\gamma_{YZ}) * (le * \sin\gamma_{XZ} * \cos\gamma_{YZ}) + F_{nut} * x_{arm}}{z_b} \tag{7}$$

$$R_x = \frac{(F_{load} * \cos\gamma_{XZ}) * (le * \sin\gamma_{YZ} * \cos\gamma_{XZ}) - (F_{load} * \sin\gamma_{XZ}) * (z_s \pm z)}{z_b} \tag{8}$$

where z_b is the vertical distance between bushings (i.e the distance between points A and B), z_s is the thickness of the sliding element, z is the upper spring seat vertical displacement during actuation, γ_{XZ} and γ_{YZ} are the angles between le and respectively the XZ and YZ plane; in the symbol $\pm$, the plus sign is used for the lowering, while the minus sign correspondes to the lifting.

Thus, assuming that $R = R_A = R_B$, it's possible to revel the force F_{nut} required on the power screw, through the equilibrium along the Z direction:

$$F_{nut} = (F_{load} * \cos\gamma_{YZ}) \pm (2 * f_b * R) \tag{9}$$

In this case, in the symbol $\pm$, the plus sign relates the lifting phase, while the minus sign the lowering phases.

At this point, the torque of the electric motor T_{em} required to develope F_{nut} is computed as:

$$T_{em} = \frac{F_{nut}}{i_s * i_g * i_{gb} * \eta_s * \eta_g * \eta_{gb}} \tag{10}$$

i_s represent the transmission ratio of the screw-nut mechanism ($i_s = \frac{2 * \pi}{p}$, where p is the pitch), i_g of the gear drive unit and i_{gb} of the gearbox.

η_s is the efficiency of the ball screw- nut motion conversion, that is different in the lifting mode or in lowering one, in which there may be regeneration:

$$\eta_{s,lift} = \frac{tan\lambda}{\tan(\lambda + \phi)} \tag{11}$$

$$\eta_{s,lower} = \frac{\tan(\lambda - \phi)}{\tan(\lambda)} \tag{12}$$

η_g and η_{gb} are the efficiencies of the gear drive unit and of the gearbox.

The aim is to assess the electric motor current absorption and the actuation speed in both phases. They depend on the dynamics of the electro-mechanical system; therefore, a dynamic model is developed.

It is based on three equations:

- The resistance torque T_{em} reported on the electric motor shaft, that is the Equation 10;
- The torque balance on the electric engine shaft:

$$k_t * I = J_{eq} * \dot{\omega}_{em} + T_{em} \tag{13}$$

where k_t is the motor torque constant, I is the current flowing in the windings, $\dot{\omega}_{em}$ is the angular acceleration of the electric engine shaft and J_{eq} is the equivalent moment of inertia of rotating and translating elements. This last is calculated as:

$$J_{eq} = J_{em} + J_{gb} + \frac{J_p}{i_{gb}^2} + \frac{J_g}{i_{gb}^2 * i_g^2} + \frac{m}{i_{gb}^2 * i_g^2 * i_s^2} \tag{14}$$

J_{em} is the electric motor shaft inertia, J_{gb} is the gearbox inertia, J_p and J_g are the pinion and the gear inertia of the gear drive unit respectively and m is the vehicle corner mass;

- The voltage balance for equivalent DC circuit of the electric motor:

$$V_{DC} - R * I - L * \frac{dI}{dt} = k_{em} * \omega_{em} \tag{15}$$

where V_{DC} is the supply voltage of the vehicle battery, R and L are the resistance and the inductance of windings respectively, k_{em} is the engine speed constant and ω_{em} is the electric motor angular velocity.

It's possible to write Equations 13 and 15 in a matrix form with a state space representation ($\dot{x} = A * x + B * u$), using as states of the system the motor shaft angular speed ω_{em}, the angular displacement θ_{em} and the current I, while as inputs the supply voltage V_{DC} and the resisting torque T_{em}:

$$\begin{Bmatrix} \dot{\omega}_{\mathrm{em}} \\ \omega_{\mathrm{em}} \\ \dot{I} \end{Bmatrix} = \begin{bmatrix} 0 & 0 & \frac{k_{\mathrm{t}}}{J_{\mathrm{eq}}} \\ 1 & 0 & 0 \\ -\frac{k_{\mathrm{e}}}{L} & 0 & -\frac{R}{L} \end{bmatrix} \begin{Bmatrix} \omega_{\mathrm{em}} \\ \theta_{\mathrm{em}} \\ I \end{Bmatrix} + \begin{bmatrix} 0 & -\frac{1}{J_{\mathrm{eq}}} \\ 0 & 0 \\ \frac{1}{L} & 0 \end{bmatrix} \begin{Bmatrix} V_{\mathrm{DC}} \\ T_{\mathrm{em}} \end{Bmatrix} \tag{16}$$

Moreover, the linear displacement of the upper spring plate and its translational velocity can be obtained as:

$$u_z = \frac{\theta_{em}}{i_s * i_g * i_{gb}} \tag{17}$$

$$\dot{u}_z = \frac{\omega_{em}}{i_s * i_g * i_{gb}} \tag{18}$$

Hence, the graph in the following Figure 49 representing the current consumption and stroke of the upper spring holder with respect to the time can be obtained as result. The next Table 4 contains the main parameters of the numerical simulation:

Parameters	Value
F_{load} − [N]	5579
le − [mm]	16.5
γ_{xz} − [°]	3.5
γ_{yz} − [°]	3.5
x_{arm} − [mm]	73
z_b − [mm]	124
z_s − [mm]	20
d − [mm]	16
p − [mm]	4

Table 4: **Numerical simulation parameters**

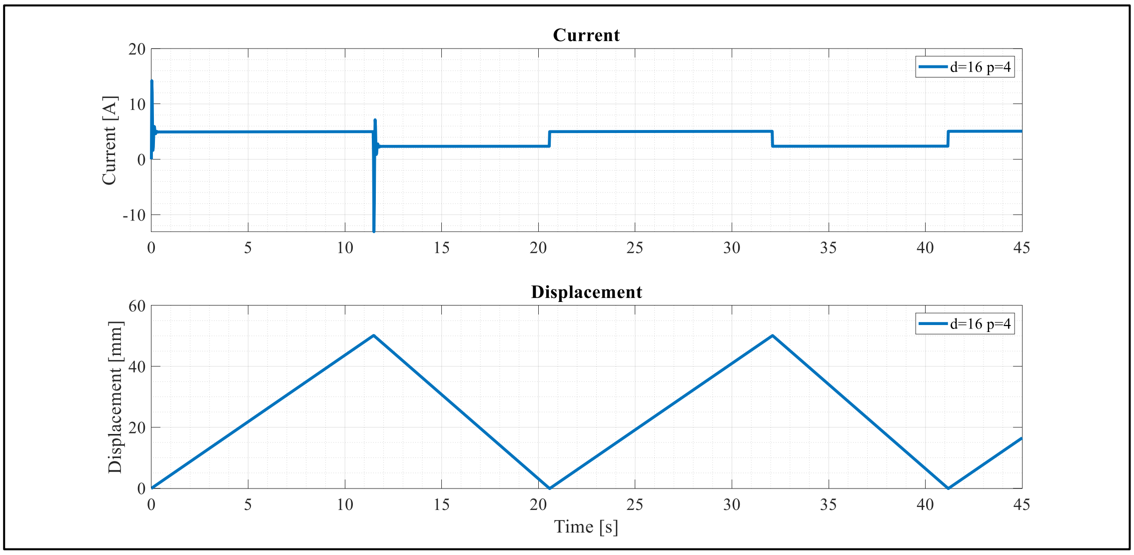

Figure 49: **Current consumption and linear displacement of the upper spring seat with respect to the time**

Exploiting the model, the main sources of power losses can be highlighted; considering the lifting phase, they are summarized in the following Figure 50.

Based on these outcomes, to improve performances, at the expanse of an increase of costs, it's possible to select a better performing motor-reducer and substitute the bronze bushings with rolling bearing.

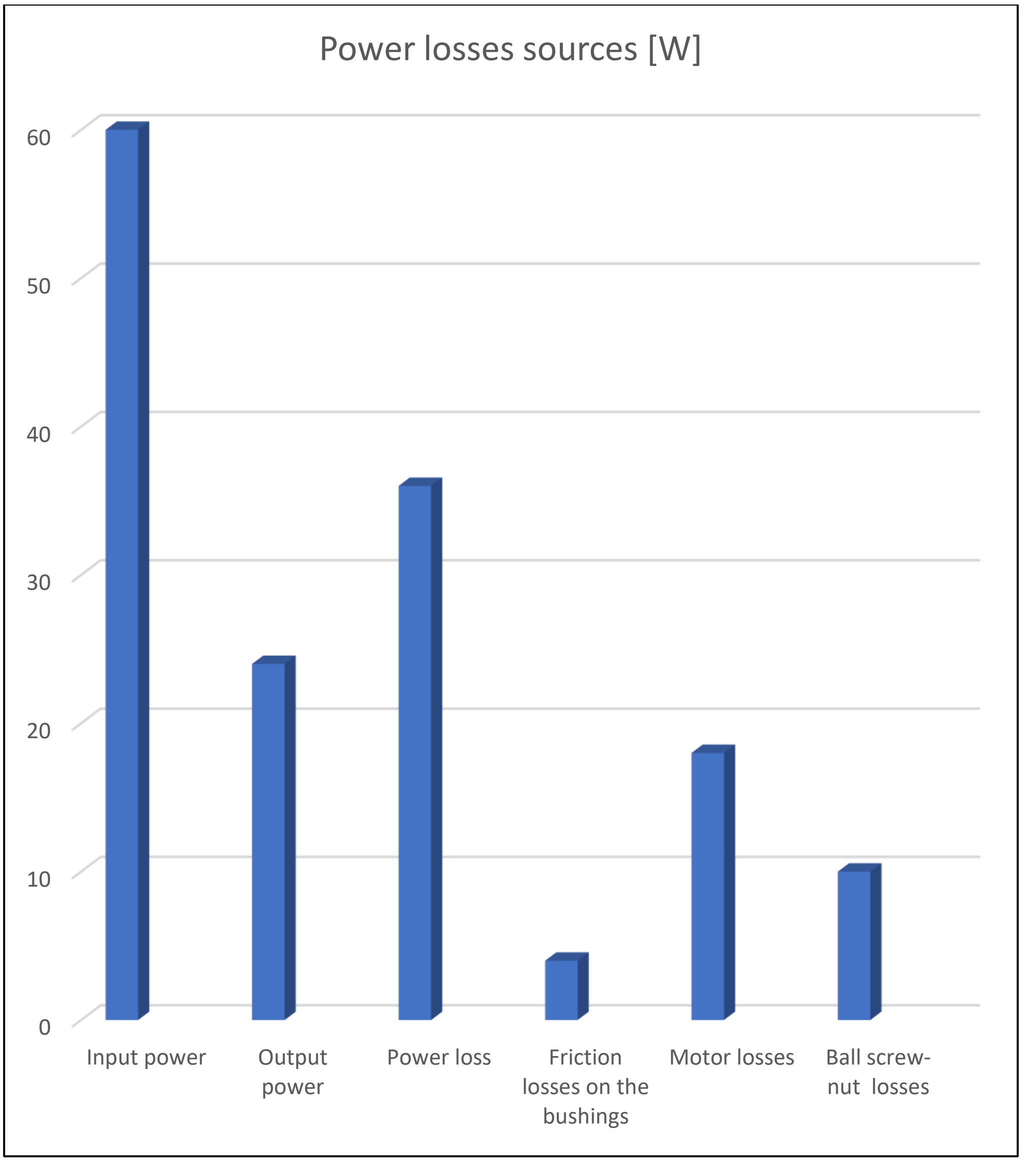

Figure 50: **Power losses sources**

3.3 Component sizing

This subchapter describes the design of the main components of the reversible eccentric electro-mechanical height adjustment system, comprising the selection of the electric motor, the definition of the power ball screw-nut mechanism with its anti-rotational device and the dimensioning of the internal guide and sliding element.

Moreover, detailed 3D and 2D drawings are obtained.

3.3.1 Selection of the electric motor and definition of the power ball screw-nut mechanism

Making reference to the work of Ruzimov et al. [22], the calculation procedure used for the selection of the electric motor and the definition of the power ball screw-nut mechanism is described through the flow chart depicted in the next Figure 50.

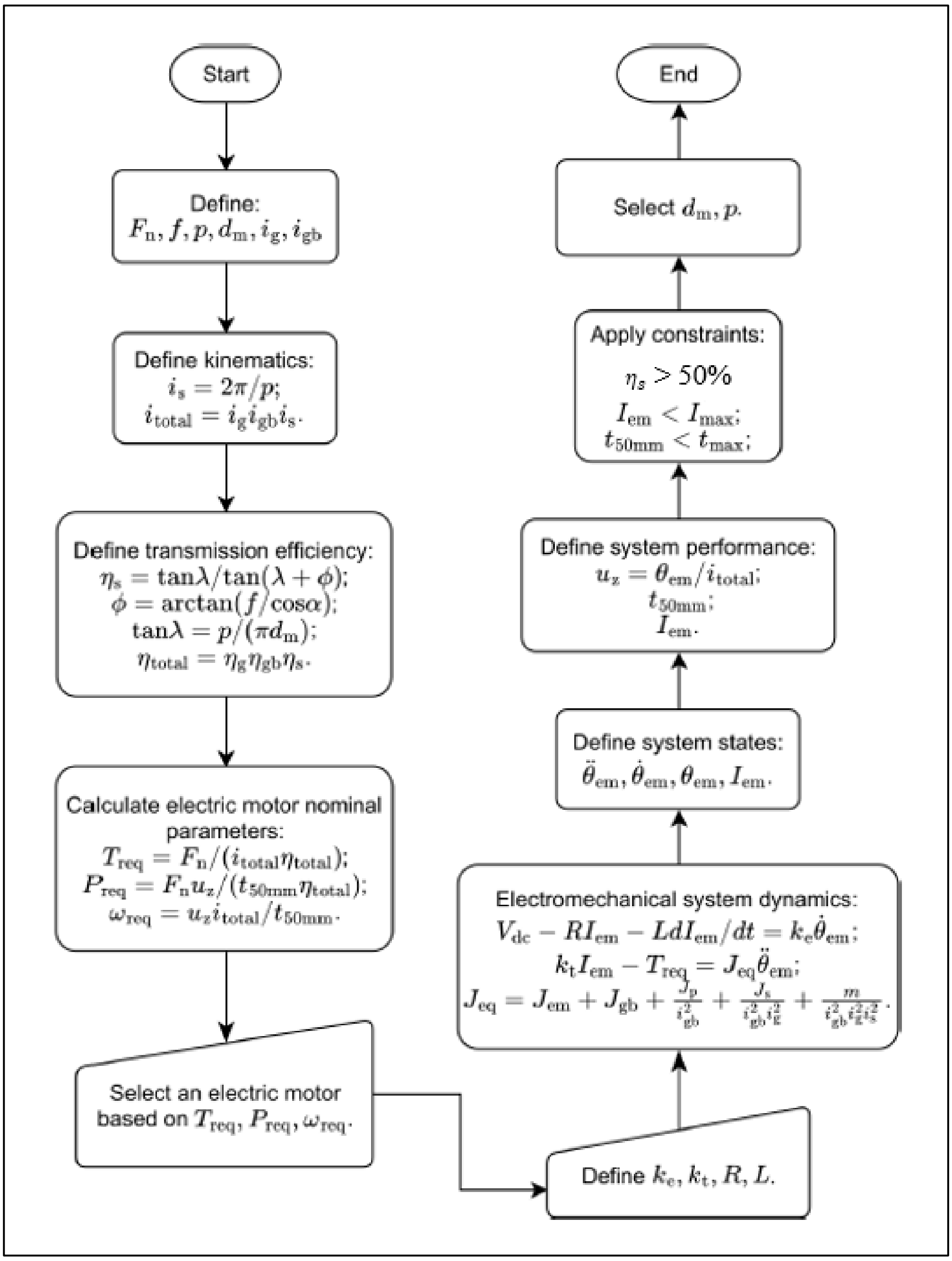

Figure 51: **Flowchart of the electromechanical design of the power screw actuation system**

The design methodology starts by defining the load F_n (that represents the corner weight), the friction coefficient between the screw and the nut f_s ($f_s = 0.003 - 0.01$ for ball screw), a first attempt values for the screw's and nut's mean diameter d_m and pitch p and a transmission ratios of the gearbox i_{gb} and the drive unit i_g.

So, the screw-nut transmission ratio i_s and consequently the total one i_{total} are obtained.

With Equations 11 and 12, the screw-nut motion conversion efficiency η_s is achieved in the lifting and lowering phase and also the total efficiency η_{tot} as the product of η_s, η_{gb} (gearbox's efficiency) and η_g(drive unit's efficiency).

At this point, the electric engine nominal parameters necessary to drive the load F_n can be calculated: required torque T_{req}, power P_{req} and angular speed ω_{req}.

Therefore, it's possible to define the commercial electric motor appropriate to the application. As mentioned above in Subchapter 3.1 – *Description and working principles*, the motor-reducer Johnson Electric HC685LG – 120 is selected; the specifications are described in the following Table 5.

Parameters	Value
Input voltage - [V]	12.0 V DC
No load speed – [rpm]	10125
No load current – [A]	0.57
Nominal speed – [rpm]	8944
Nominal torque – [mNm]	37.28
Nominal current – [A]	4.34
Stall torque – [mNm]	321.13
Stall current – [A]	32.88
Maximum output power – [W]	85
Maximum efficiency – [%]	67%
Weight – [g]	227
Operating temperature range – [°C]	-40 to 85
Storage temperature range – [°C]	-40 to 120
Reduction stage transmission ratio – [-]	150

Table 5: **Specification of the Johnson Electric HC685LG – 120** [23]

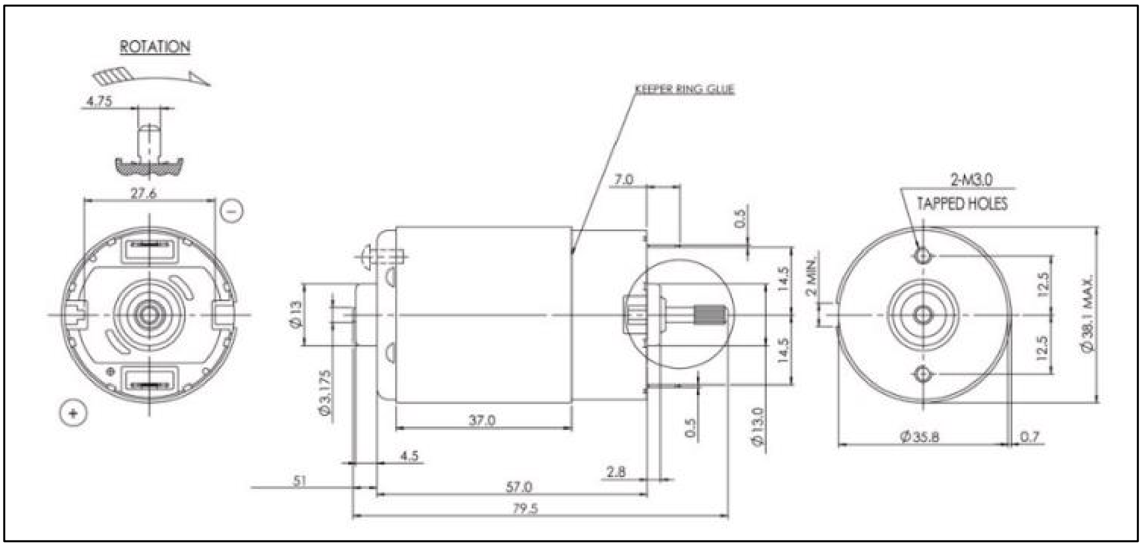

Figure 52: **Drawing of the Johnson Electric HC685LG – 120** [23]

Moreover, the torque constant k_t, the speed constant k_e, the phase-to-phase resistance R and the inductance L are specified from the available data.

Combining the electrical and mechanical dynamics of the actuation system in the numerical model described in Subchapter 3.2 – *Numerical modeling*, the states, such as motor angular acceleration $\dot{\omega}_{em}$, speed ω_{em} and displacement θ_{em} as well as actuation current I_{em}, are defined.

Thus, the performance of the system is determined in terms of time t_{50} to fulfill a stroke displacement $u_z = 50\ mm$ (or likewise lifting and lowering speed) and steady-state current I_{ss}.

Get there, to achieve the optimal screw's and nut's diameter d_m and pitch p, it's necessary to apply some constrains:

- Screw-nut motion conversion efficiency $\eta_s > 50\%$, that is related to the decision of using a reversible solution, as mentioned in Subchapter 2.2 - *Choice of the height adjustment system*;
- Steady state current flowing in the electric motor windings $I_{ss} < 5\ A$; this limit value can be set based on the maximum current capabilities of the electric engine and its control board or, like in this case, on performance consideration;
- Time to travel stroke of 50 mm $t_{50} < 20\ s$; the latter is an imposed performance target;
- A life of at least 5.000 lifting-lowering cycles must be ensured from the structural point of view.

So, exploiting the dynamic model developed in Subchapter 3.2 – *Numerical modeling*, it's possible to perform a parametrical sweep of numerical simulations to find suitable values for d_m and p; the design point is highlighted with a circle.

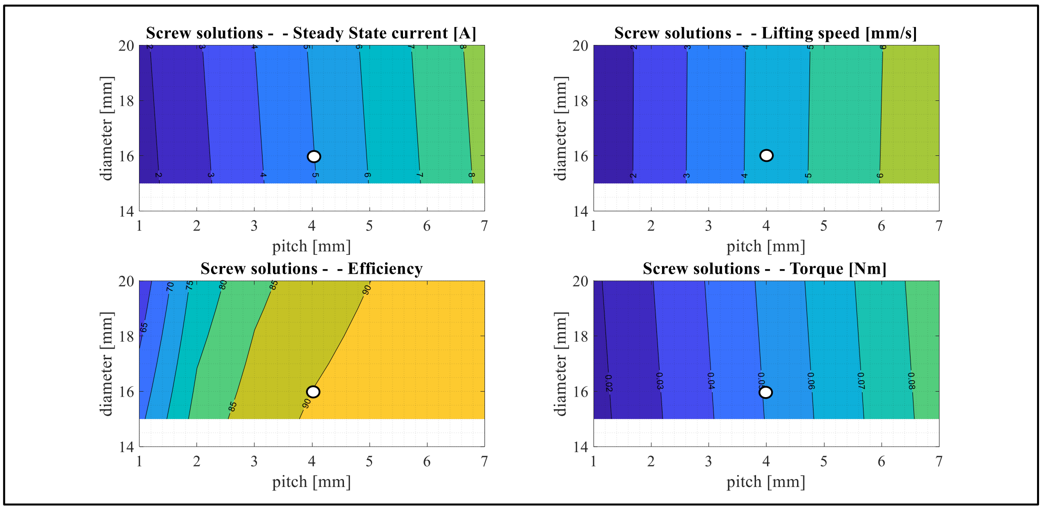

Figure 53: **Parametrical sweep on the diameter and the pitch of the screw-nut mechanism**

It's necessary to underline that, related to the load F_n and the chosen electric motor, the minimum diameter $d_{min} = 15\ mm$, otherwise the system is not able to lift the corner.

In Figure 53, the screw efficiency, current and time constraints are applied.

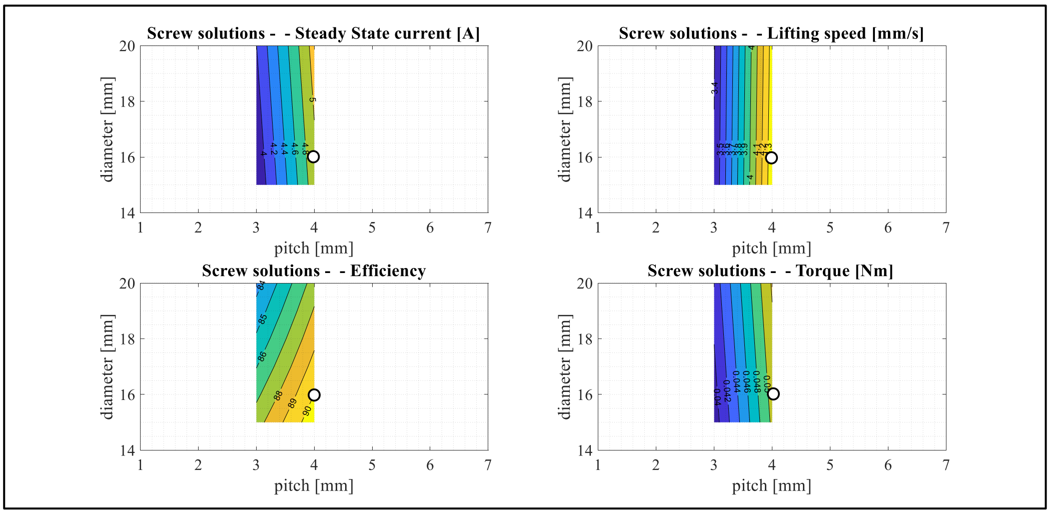

Figure 54: **Imposition of screw efficiency, current and time constraints to the parametrical sweep**

It's possible to notice that the current consumption depends mainly on the mean screw diameter d_m and is less influenced by change the pitch p; on the contrary, actuation time has a little dependence on the d_m, but the effect of p is more evident.

Then, the life constrain has to be considered. The life expectancy of the screw and the nut in terms of cycles can be given by this experimental formula:

$$L = \frac{(\frac{C}{F_a})^3 * 60 * 10^6}{n_{av} * t_{cycle}} \tag{19}$$

where C is the dynamic load rating (it can be obtained by the supplier's catalogues and represents the dynamic resistance of the specific ball screw-nut mechanism), n_{av} is the average angular velocity measured in rounds per minutes [rpm] ($n_{av} = n_{lift} * \frac{t_{lift}}{100} + n_{lower} * \frac{t_{lower}}{100}$, in which $\frac{t_i}{100}$ is the percentage of time passed at the velocity n_i), t_{cycle} is the time needed to compute a complete cycle with both lifting and lowering phase and F_a is the average axial load, calculated as:

$$F_a = \sqrt[3]{F_{a,lift}^3 * \frac{n_{lift}}{n_{av}} * \frac{t_{lift}}{100} * f_{p,lift}^3 + F_{a,lower}^3 * \frac{n_{lower}}{n_{av}} * \frac{t_{lower}}{100} * f_{p,lower}^3} \tag{20}$$

$F_{a,i}$ is the axial load and $f_{p,i}$ is the operative condition factor during the different two modes; this last can assume these values:

$\boldsymbol{f_p}$ [-]	Condition
1.1 – 1.2	Smooth functioning
1.3 – 1.9	Normal functioning
2.0 – 3.0	Functioning with strong vibrations and shocks

Table 6: **Operative condition factor $\boldsymbol{f_p}$**

For this application, $f_{p,lift} = f_{p,lower} = 2$ has been considered.

The goal in the choice of the ball screw and nut is to obtain the best tread-off between power consumption and actuation speed, guaranteeing the life target, but also looking to minimize the costs. Thus, the optimal solution is to use an off-the-shelf C7 precision rolled ball screw coupled to a single flanged nut with internal recirculation and four circuits with a mean diameter $d_m = 16\ mm$ and a pitch $p = 4\ mm$. C7 is a low class of precision (that are from rectified C3 to the rolled C10), which however is compatible with the height adjustment scope and permits to reduce the component cost. Using this one, 15.000 cycles are guaranteed with 90% of reliability. [24]

Then, also the static and the critical speed verification are performed and passed. The static verification makes reference to the following formula:

$$S_f * F_a(max) < C_0 \tag{21}$$

where $S_f = 2.5$ is the static saftey factor, C_0 is the static load rating and $F_a(max)$ is the maximum axial load during the actuation; while the critical speed verification refers to the graph:

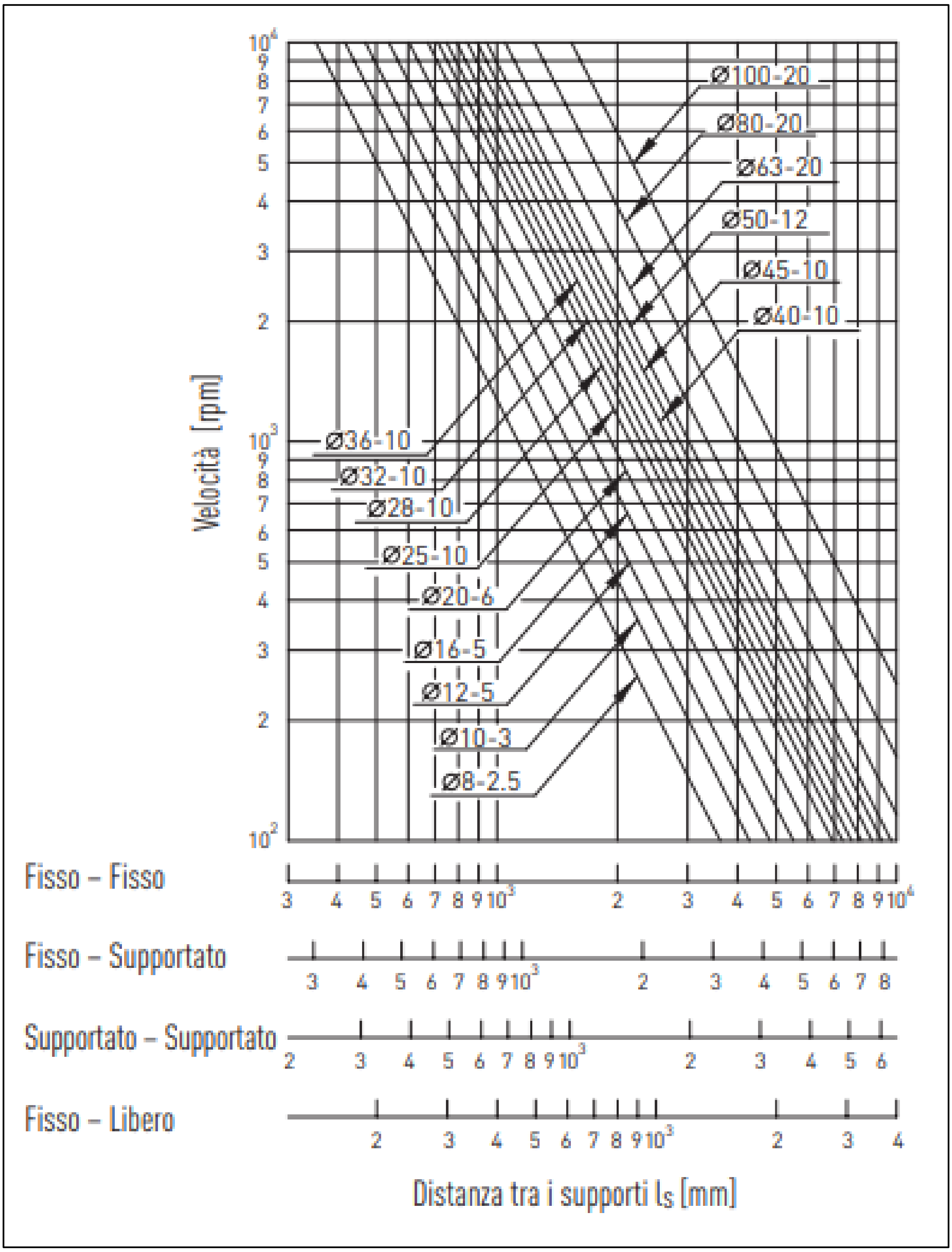

Figure 55: **Critical speed verification** [25]

As described in Subchapter 3.1 – *Description and working principle* and shown in Figure 40, for the anti-rotational feature and to avoid the reworking of the nut, a slotted flange is selected.

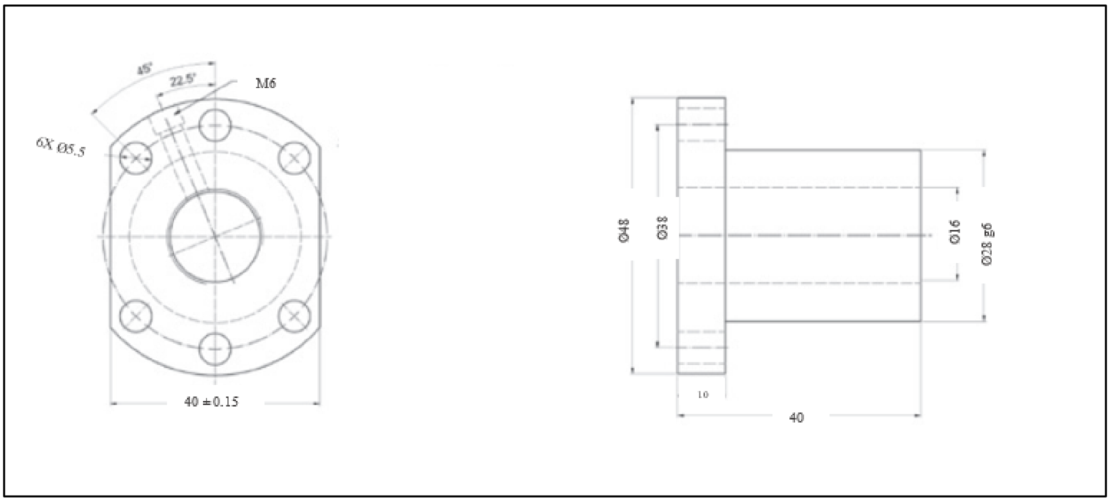

Figure 56: **Nut's 2D drawing** [24]

Figure 56 is the 3D drawing of the anti-rotational device; it has to prevent the rotative motion of the nut, but also allow a slight flotation. (The anti-rotational device's 2D drawing is in the Appendix A).

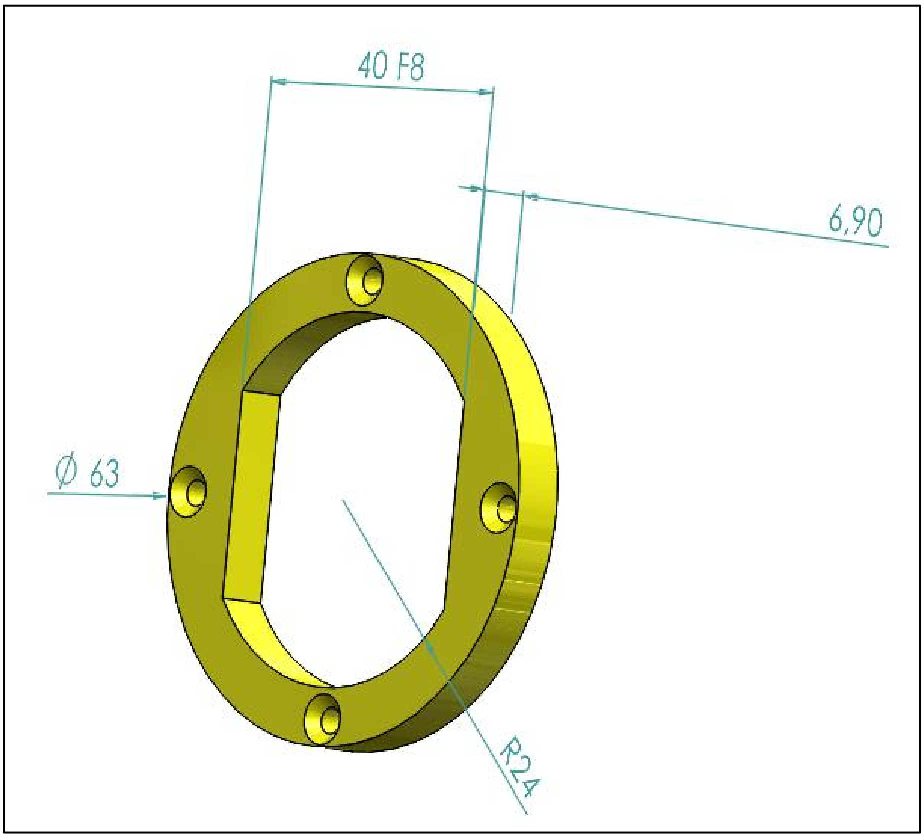

Figure 57: **Anti-rotational device's 3D drawing**

3.3.2 Design of the support and the sliding element

The next step is the design of the support; it is composed by a cylindrical member connected to an upper arm, which, in turn, has to be attached to the frame.

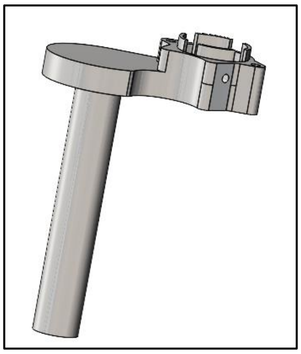

Figure 58: **Support**

The cylinder is modelized as a clamped hollow beam only subject to bending due to the eccentricity of the system (the axial friction forces has been neglected). The reaction forces R of the two bushings are obtained using the numerical model, described in detail in Subchapter 3.2 – *Numerical modeling*, with Equations 7 and 8. The distance between bushings $zb = 124\ mm$ is chosen because it permits to achieve acceptable power losses due to friction, without over-increasing the component length, that can create space problems inside the coil spring during the functioning.

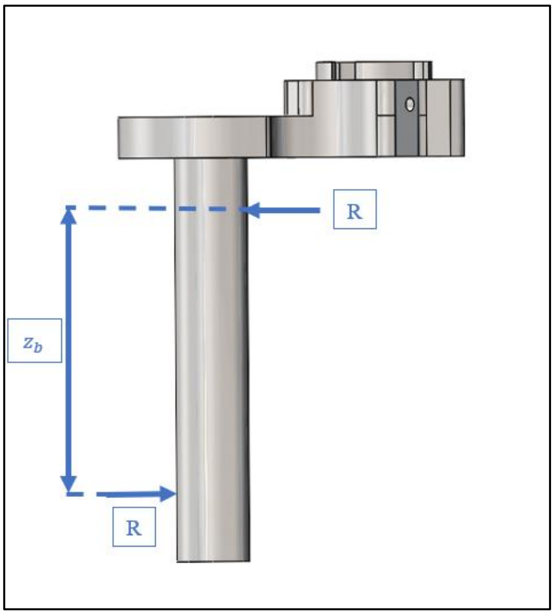

Figure 59: **Support's free body diagram**

Moreover, the cylindric element's length must ensure, as performance target, a sliding member stroke of 50 mm.

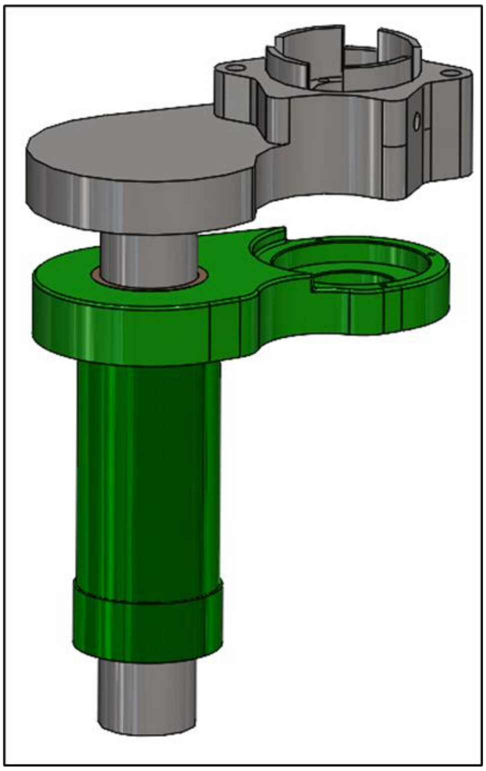

Figure 60: **Coupling between sliding element and support**

So, it has been dimensioning by means of a calculation routine, considering as material the C40 steel (in laminated and hardened state the yield strength $\sigma_y = 400\ MPa$ and tensile strength $\sigma_t = 700\ MPa$), typically used in automotive field for this type of application.

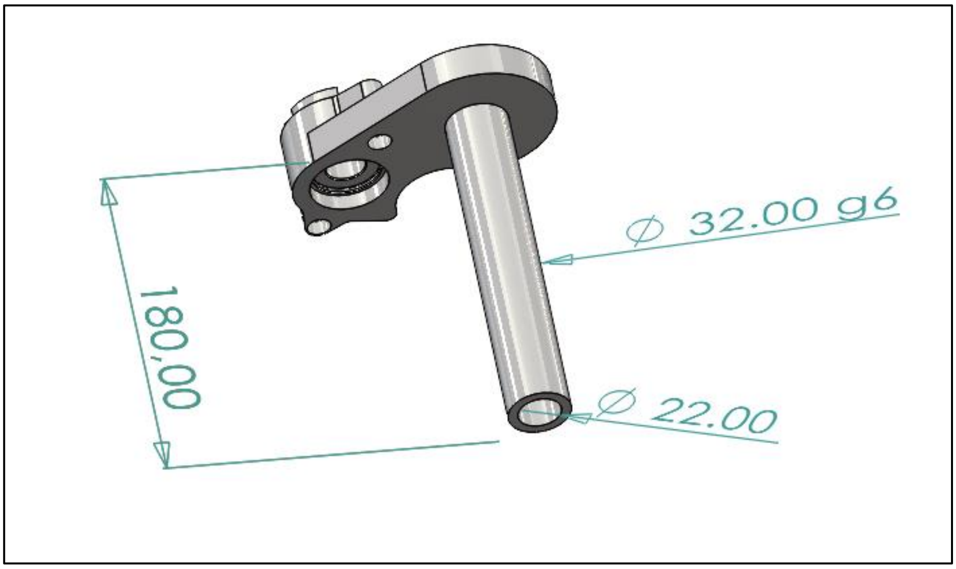

Figure 61: **Support's cylinder**

A static safety factor $SF_{static} = 1.54$ and a fatigue saftey factor $SF_{fatigue} = 2.11$ (calculated with constant mean stress σ_m in the Haigh diagram, depicted in Figure 61) are secured.

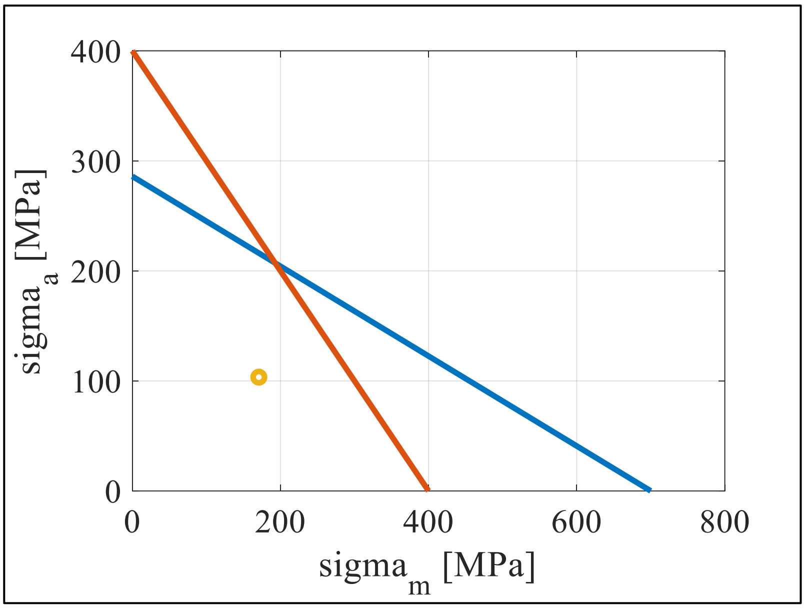

Figure 62: **Haigh diagram**

Instead, the upper arm is the housing for the electro-magnetic clutch, the screw adapter and the electric motor.

The support is coupled with the sliding element via bronze bushings. (The bronze bushing's 2D drawing is in the Appendix B).

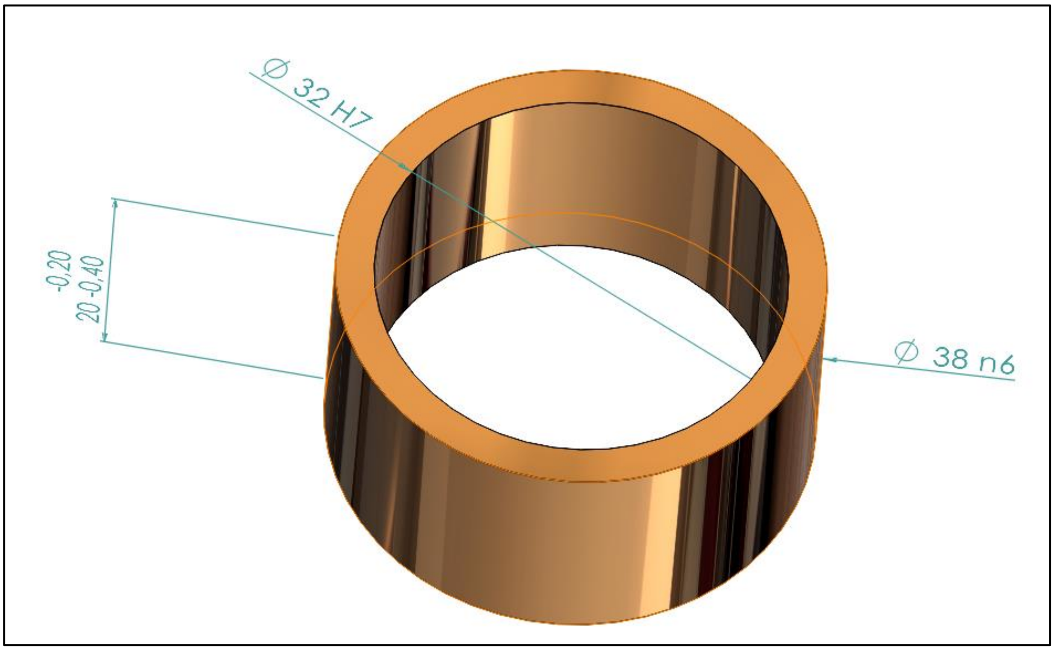

Figure 63: **Bronze bushing's 3D drawing**

The sliding component consists in an upper arm, on which is mounted the nut along with the anti-rotational device and the rubber cushion, and a cylindrical element; the latter is design to avoid impunities due to the deformation of the support during the actuation and to accommodate the bushings.

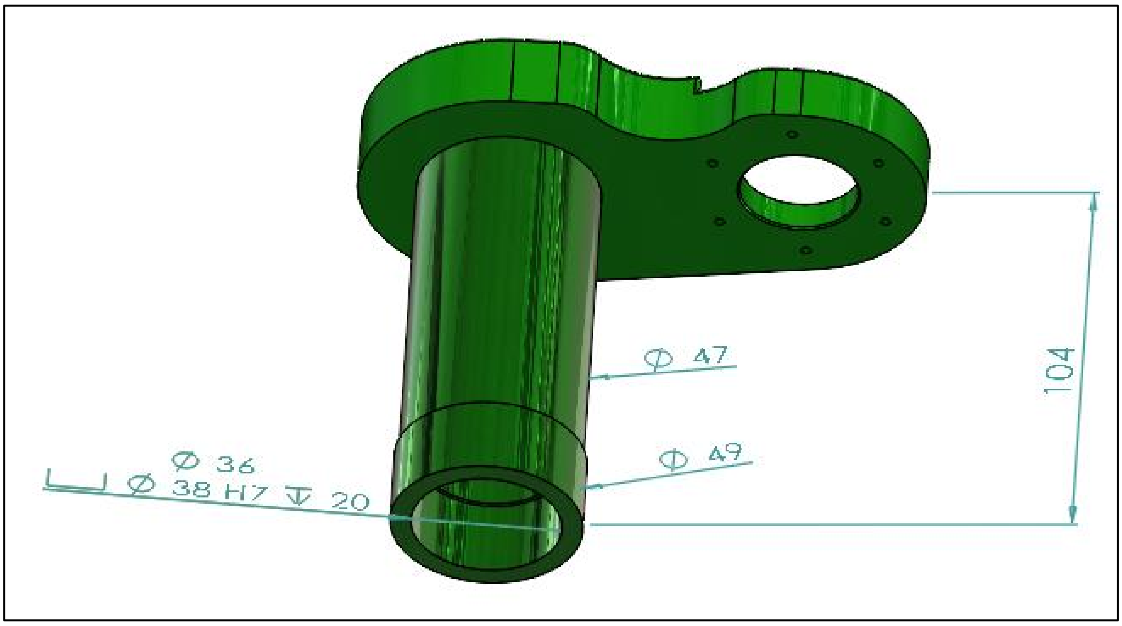

Figure 64: **Sliding element's 3D drawing**

4. Experimental validation

In this chapter, the experimental validation of the numerical model, developed in the Subchapter 3.2 – *Numerical modeling*, is provided.

It has been made by means of a reversible eccentric height adjustment system prototype for a MacPherson suspension, already realized for the vehicle Fiat 500X.

Figure 65: **Prototype**

The actuator has the same architecture, describe in detail in Subchapter 3.1 – *Description and working principle*, but with a lower spring seat actuation.

It is controlled by an electronic control unit (ECU), that receives the actuation signal from a user interface through a CAN communication.

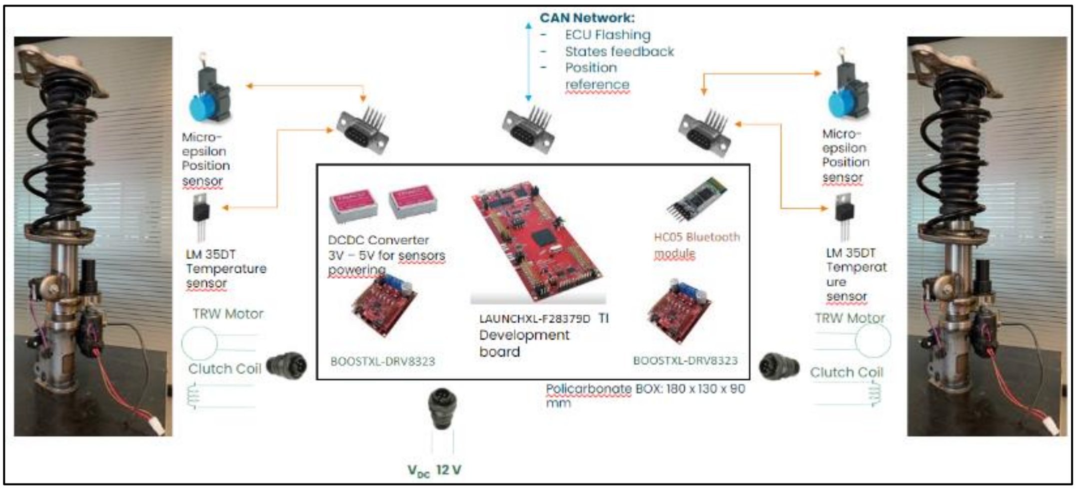

Figure 66: **ECU scheme**

With this ECU scheme, an entire axle can be controlled, that can be advantageous from the vehicle installation point of view

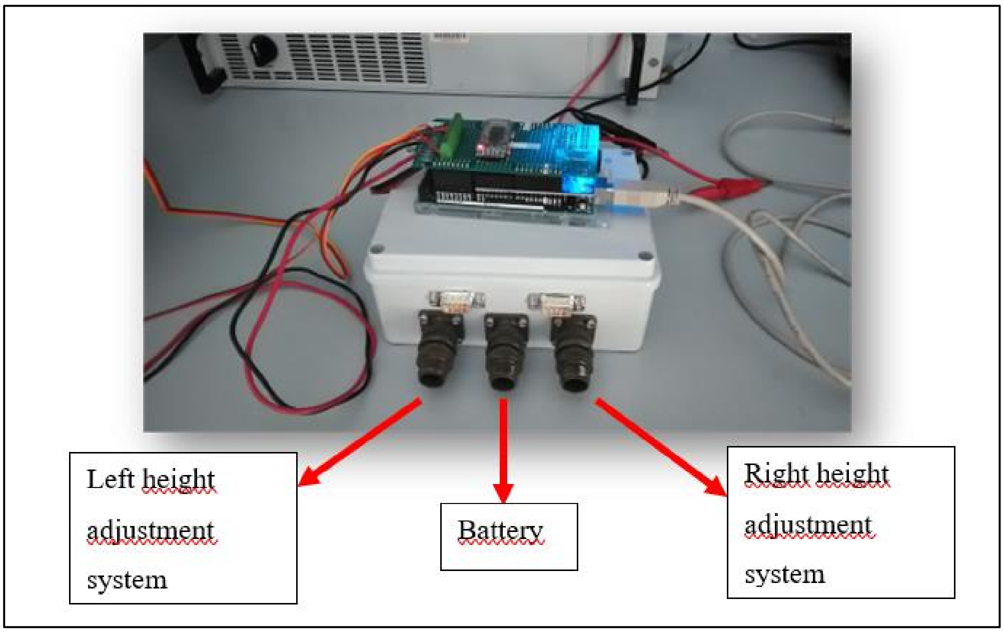

Figure 67: **ECU**

As shown in Figure 65, the device is sensorised via:

- Angular position sensor to measure the lower spring plate stroke;
- Current sensor (integrated in the board BOOSTXL-DRV8323) to measure the current flowing in the windings of the electric motor;
- Temperature sensor to measure the temperature of the electric engine during the functioning to avoid problems related to over-temperature.

So, the experimental data are obtained.

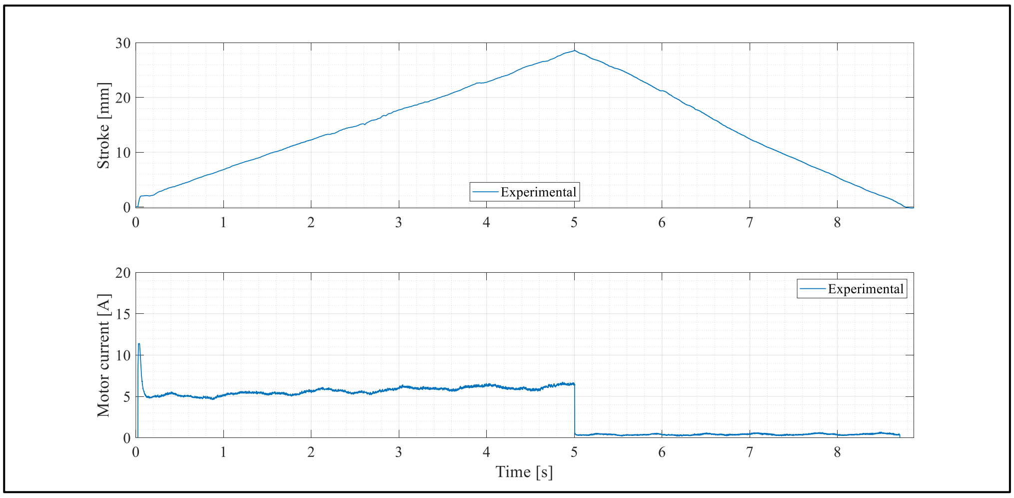

Figure 68: **Experimental data**

Because the height adjustment concept is essentially the same for the designed system and the prototype, the model, making reference to the free body diagram depicted in Figures 46 and 47, has been parametrized on:

- The load F_{load}, that is related to the corner weight;
- le, which represents the offset of F_{load};
- γ_{XZ} and γ_{YZ}, that are the angles between le and respectively the XZ and YZ plane;
- x_{arm}, which is the eccentricity of the force F_{nut};
- The vertical distance between the bronze bushings z_b;
- The thickness of the sliding element z_s;
- The diameter d and the pitch p of the screw and the nut.

So, modifying the value of these parameters, it's possible to adapt it to assess the performance of the prototype with numerical simulation.

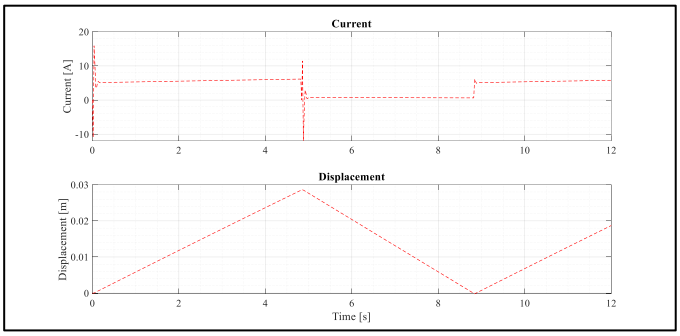

Figure 69: **Numerical simulation**

Comparing the experimental data and the simulated results, it possible to notice that the numerical simulation faithfully reproduces the experimental data; thus, the numerical model is validated.

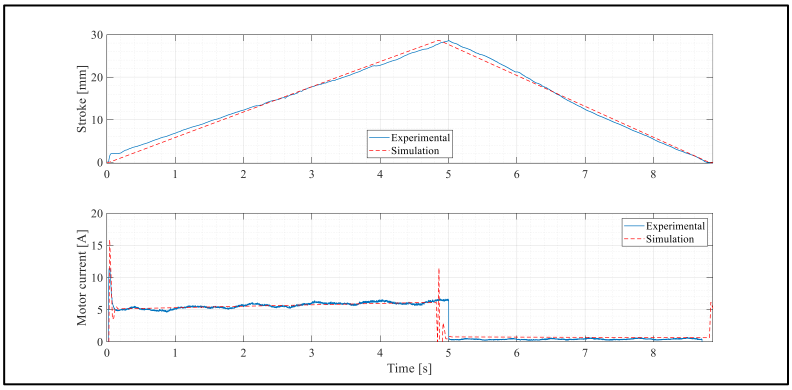

Figure 70: **Comparison between experimental data and numerical simulation**

Using a ball screw-nut mechanism for the motion conversion, an electro-magnetic clutch, as locking device, has been produced and the additional current consumption during the spring plate stroke has been measured.

Figure 71: **Electro-magnetic clutch’s auxiliary disk**

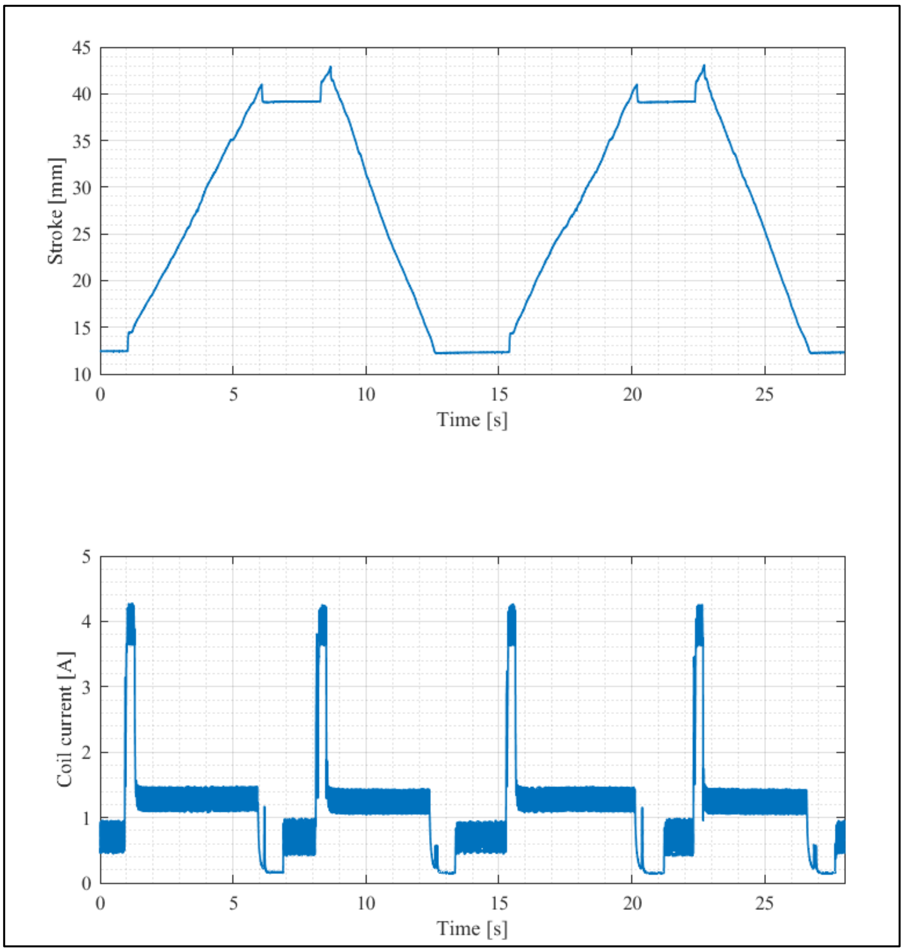

Figure 72: **Electro-magnetic clutch’s current consumption**

It’s possible to notice that, to facilitate the disengagement of the clutch in the actuator lowering phase, before a slight upward motion of the upper spring plate is necessary.

So, the total motor current consumption becomes:

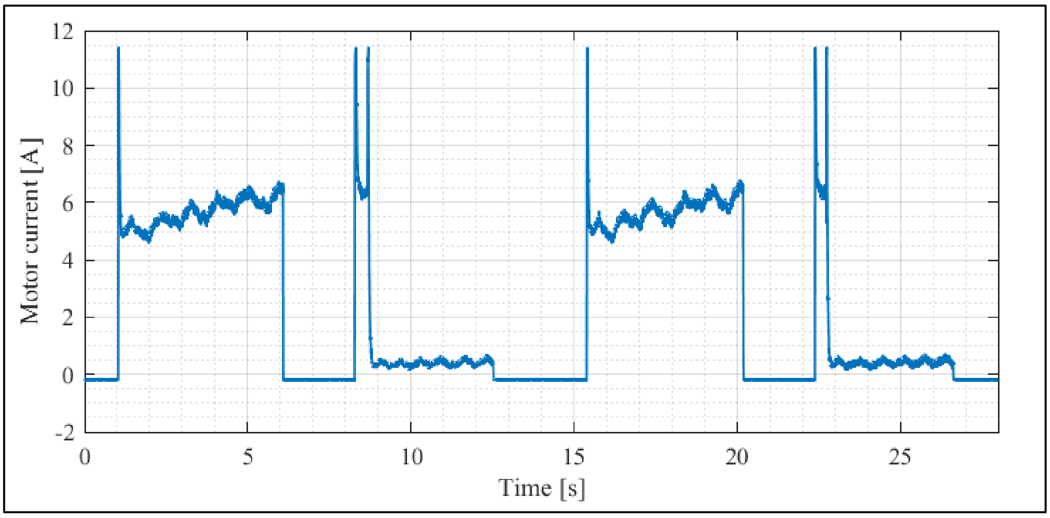

Figure 73: **Motor current consumption**

5. Conclusion

To conclude the dissertation presented, it is important to resume the main goals achieved during this master thesis project.

Starting from a literary review of the existing height adjustment systems, it has been possible to understand the benefits that they can offer in terms of reduction of fuel consumption and CO_2 emissions towards the sustainable mobility.

In the present work, the focus has been the computer aided design of reversible eccentric electro-mechanical height adjustment for the rear multi-link suspension of Alfa Romeo Stelvio. This solution has permitted to obtain the best compromise between functionality, performances and economic aspects.

The optimized design has been realized by running numerical simulations using parametrized models, which has been developed and experimentally validated.

In addition, 2D and 3D drawings of different components of the actuation device are obtained.

5.1 Future developments

The project, carried out within the context of this dissertation, is only a stage of the complete process for the development of the rear Alfa Link's height adjustment system. The scope and the time frame of master thesis project has been limited to the design and the numerical modeling.

The expected future developments are:

- A 3D printer mock-up to evaluate eventual space problems in the implementation on Alfa Romeo Stelvio;

- A detail design of the connection between the actuator and the longitudinal beam of the frame;
- The realization of the prototype;
- The acquisition of experimental data off and on vehicle.

Appendix A

Anti-rotational device's 2D drawing:

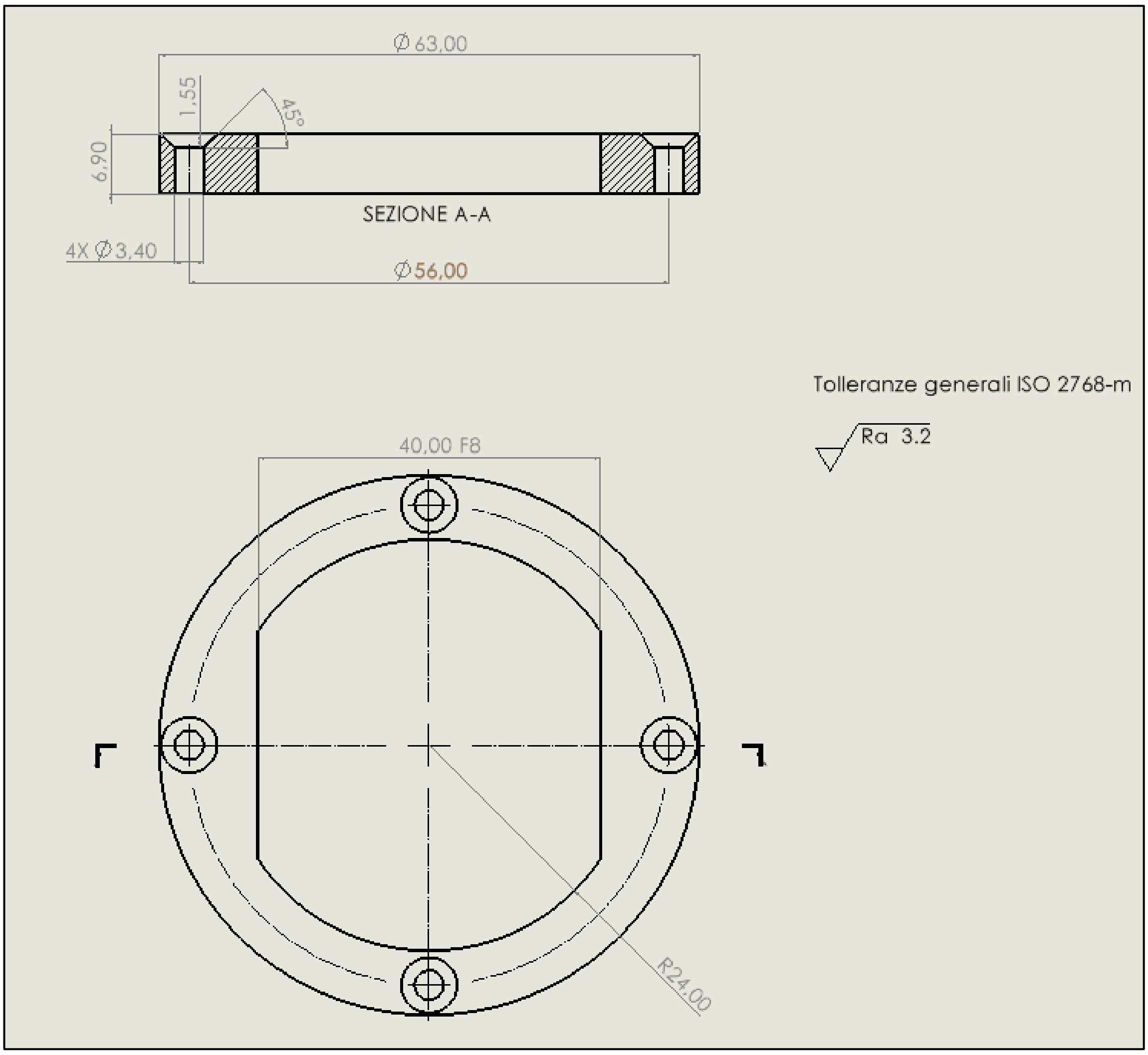

Figure 74: **Anti-rotational device's 2D drawing**

Appendix B

Bronze bushings’ 2D drawing:

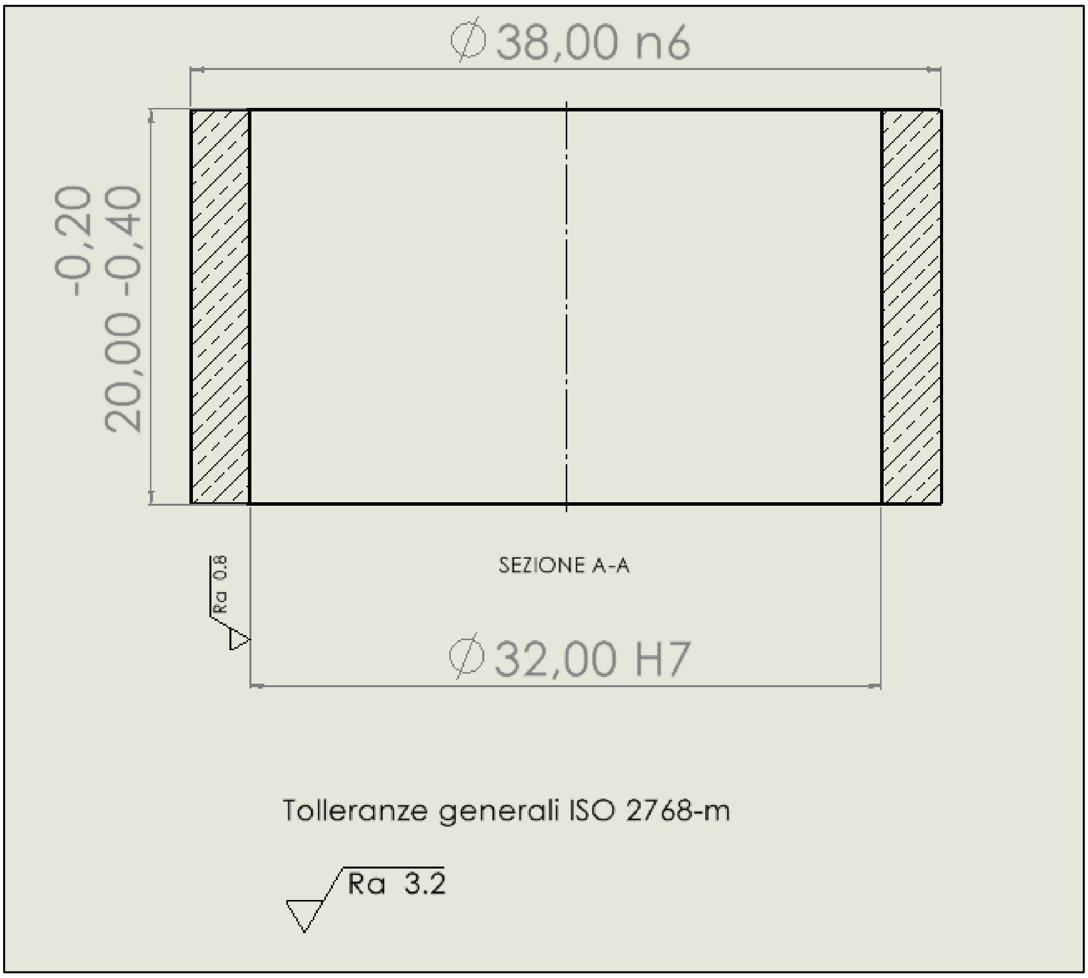

Figure 75: **Bronze bushings’ 2D drawing**

References

[1] International Energy Agency - *https://www.iea.org/energy-system/transport/cars and-vans* (accessed 12 October 2023).

[2] European Commission - *https://climate.ec.europa.eu/eu-action/transport/road-transport-reducing-co2-emissions-vehicles/co2-emission-performance-standards-cars-and-vans_en* (accessed 12 October 2023).

[3] M. Velardocchia, *Appunti di fondamenti di Meccanica del Veicolo*, 2021.

[4] N. Amati, A. Tonoli, L. Castellazzi and S. Ruzimov, *Design of electromechanical height adjustable suspension,* Proc. Inst. Mech. Eng. Part D J. Automob. Eng., 2017.

[5] L. Hanzhang, *Development of height adjustment system for vehicle slow attitude control*, 2019.

[6] J. Wirtz and H. Luncz, *Height adjusting shock absorber strut for a motor vehicle*, Patent 5,810,335, USA, 1998.

[7] D. Sahm, D. von Scarpatti, D.C. Karnopp and D. Beard, *Motor vehicle suspension system*, Patent 5,401,053, USA, 1995.

[8] B.M. Kim, S.B. Jang, J.M. Kim, J.S. Kim and D.S. Ko, *Shock absorber for vehicle*, Patent US 8,573,606 B1, USA, 2013.

[9] A.C.M van der Knaap, *Active chassis of a vehicle*, Patent US 7,611,152 B2, USA, 2009.

[10] M. Hirose, S. Matsushige, S. Buma and K. Kamiya, *Toyota Electronic Modulated Air Suspension System for the 1986 Soarer*, IEEE Trans Ind Electron 1988; 35: 193-200

[11] D.R. Tener, *Overcoming the Ride/Handling Compromise – A cockpit Adjustable Suspension System*, SAE paper 2001-01-1078, 2004.

[12] W. Bauer, *Hydropneumatic suspension system*, Berlin: Springer-Verlag, 2011.

[13] W. Michel, *Strut for a wheel suspension of motor vehicles*, Patent US 8,205,864 B2, USA, 2012.

[14] T. Hakui, H. Hirata, H. Kajiwara, K. Nakajima, Y. Akuta and K. Hatano, *Vehicle height adjustment system*, US 7,922,181 B2, USA, 2011.

[15] W. Michel, *Height adjustment on a wheel suspension for motor vehicles*, Patent US 7,784,800 B2, USA, 2010.

[16] W.Michel, *Wheel suspension for motor vehicles*, Patent US 8,317,003 B2, USA, 2012.

[17] J.M. Kim, S.B. Jang, B.M. Kim and D.S. Ko, *Electronic control suspension system for vehicles*, Patent US 8,833,775 B2, USA, 2014.

[18] T.Ohki, *Apparatus for adjusting vehicle height*, Patent US 7,237,780 B2, USA, 2007.

[19] W. Bruno, P.A. Conti, G. Greco, N. Amati, R. Galluzzi, S. Ruzimov and A. Tonoli, *Damper and spring unit for a vehicle suspension provided with an electro-mechanical adjustment device for adjusting the vertical position of the spring*, WIPO patent WO 2019/097461 A1, 2019.

[20] M. Di Vittorio, G. Greco, S. Marchetti and P. Monchiero, *Shock absorber and spring assembly for a vehicle suspension, comprising an electro-mechanical adjustment device for adjusting the vertical position of the spring and a relative blocking system, and a method for adjusting the vertical position of the spring*, WIPO patent 2022/112988 A1, 2022.

[21] Alfa Romeo Stelvio - *https://www.media.stellantis.com/uk-en/alfa romeo/gallery/1672* (accessed 12 October 2023).

[22] S. Ruzimov, L.M. Castellanos Molina, R. Galluzzi, R.Manca, N. Amati and A. Tonoli, *Modeling and Experimental Validation of the Performance of Electromechanical Height Adjustment Vehicle Suspension with Eccentric Mounted Screw System*, MDPI, Actuators 2023.

[23] Johnson Electric HC685LG datasheet - https://www.yumpu.com/en/document/read/13494462/hc685lg-120-low-voltage-dc-motors-johnson-electric (accessed 12 October 2023).

[24] Ball screw-nut mechanism NTN catalogue- *https://www.ntn-snr.com/it/ball-screws* (accessed 12 October 2023).

[25] Ball screw-nut mechanism Hiwin catalogue -
https://www.hiwin.it/images/download/documenti/viti-a-ricircolo-di-sfere-catalogo.pdf
(accessed 12 October 2023)