POLITECNICO DI TORINO

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Automotive Engineering

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Development of height adjustment system for vehicle slow attitude control

Hanzhang Li

Supervisor
Prof. Nicola Amati
Prof. Andrea Tonoli

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"Be brave, be curious, be determined, overcome the odds. it can be done."

– Stephen Hawking

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Acknowledgments

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Abstract

Under the general context of Energy conservation and emission reduction, vehicles' fuel consumption and emission reduction become more and more important since the road transport is responsible for about 20% of carbon dioxide emissions.

Except weight reduction, minimizing of the aerodynamic drag is also very important since it offers valuable benefits regarding to the fuel consumptions and carbon dioxide emissions. The height adjustment of the vehicle height is an effective way to reduce aerodynamic resistance and at the mean time introducing versatility to the vehicle, for example, off road uses.

The aim of this thesis is to present in a systematic way to current existing solutions of different types of height adjustment systems for passenger cars, summarizing the main features, working principles and the architectures. Particular attention is then focused on the electromechanical type, which represent the perfect combination of functionality and cost.

A design of Eccentric Height-adjustable System including the working principle and the description of designed prototype is presented.

Then the test rig has set up and the accelerated experiments have been proceeded. The electromechanical height adjustment system has been improved time over time based on the experimental results that having been studied. Failures and improvements are represented with the implementation of the Failure Mode and Effects Analysis methodology.

At the end, the possible improvements and future work are represented.

Contents

List of Tables

List of Figures

Chapter 1

Environmental Benefits of Vehicle Height-adjustment System

Transport emits CO_2, the most important greenhouse gas (GHG), and if global warming crosses the safety threshold of 2°C then the consequences could be anywhere between bad and catastrophic (Intergovernmental Panel on Climate Change, IPCC, 2014). In fact, there is evidence already that the safety threshold may actually be 1.5°C (Schleussner et al., 2016). To keep global warming below 2°C (or 1.5°C) atmospheric concentrations of GHGs must be stabilized and this will eventually require net zero annual emissions (IPCC, 2014). Worldwide, in 2014 transport as a whole was responsible for 23% of total CO2 emissions from fuel combustion and road transport was responsible for 20% (International Energy Agency, 2016). This justifies the trend towards more stringent legislative restrictions and standards on vehicle emissions such as, for example, those introduced by European Commission to set targets for new cars.[1]Such targets require that new cars have to emit less than 130 gCO2/km by 2015, and less than 95 gCO2/km by the end of 2020. Considering that between the period 2010- 2014 the average emission level decreased2 of 17 gCO2/km, an additional decrease of 35 gCO2/km for the same time frame seems to be challenging for vehicle manufacturers. The penalties for exceeding the limits on one side and the incentives towards lower emission vehicles2 on the other, encourages the application of innovative technologies to reduce emissions.

Technological measures to reduce fuel consumption in traditional combustion engines, can roughly be divided between improving specific engine technology, keeping engine speed within an energy efficient performance range, and optimized energy management. These engine technology measures are supplemented by reducing vehicle weight and rolling and air resistance, along with measures aimed at changing consumer behavior and driving habits. Fig.1.1 shows energy consumption in a passenger car as measured in the New European Driving Cycle (NEDC).[2]

Better fuel efficiency could be achieved by reducing vehicle weight along with air

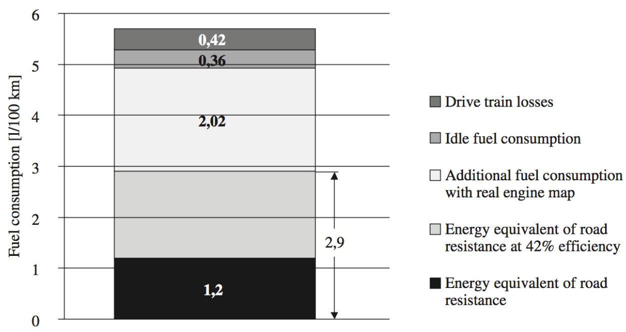

Figure 1.1. Energy Consumption in a Passenger Car in the New European Driving Cycle (NEDC)[6]

and rolling resistance. Fig.1.2 shows the impact of these vehicle parameters on fuel consumption in a hybrid vehicle. A reduction in vehicle weight offers the greatest potential for reduced fuel consumption.

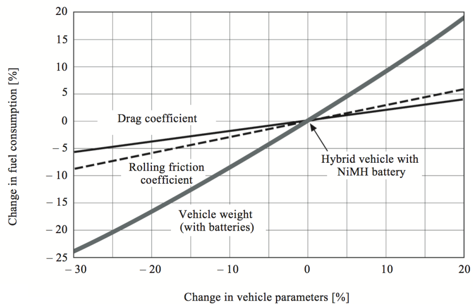

Figure 1.2. Hybrid Vehicle: Impact of Vehicle Weight and Rolling and Air Resistance on Fuel Consumption[6]

It can be achieved by optimizing the aerodynamic shape and reducing the frontal area. Since the lowering of the vehicle height could give benefits in both the aerodynamic drag coefficient and reference area, it is then considered a promising feature to be used on a modern vehicle to reduce fuel consumption and CO2 emission. Considering that one of the trends of the last decade shows a wide use of 4x4 crossover SUV cars (13% share in European market in 2015), especially in city conditions, the implication of height adjustment systems seems to give interesting opportunity to reach a compromise between versatility of the vehicle and fuel consumption.

Ride height is then a crucial parameter for the aerodynamic performance of a vehicle, An increase in ride height leads to higher drag, a low ride height scarifies the off road possibility. Height adjustment suspension is able to give a perfect compromise between fuel consumption and versatility.As shown in Fig.1.3,at high driving speed,lowering the vehicle height gives saving on fuel consumption, on rough road or off road lifting vehicle height can protect the under body. When vehicle load changes, height adjustment system can adjust the vehicle height accordingly in order to optimize the performance.

Figure 1.3. Versatility of height adjustment system

Based on aforementioned benefits of height adjustment suspension, the necessity of a study in height adjustment suspension system standalone is drawn. The first objective of this thesis is to study the environmental benefit of using a height adjustment suspension system and to compare detailed for different solutions of existing height adjustment suspensions in order figure out the most feasible solution. By doing so, the special attention is focused on the electromechanical height adjustment suspension system, the design methodology of an electromechanical height adjustment suspension system is presented in the following chapter and the prototypes of such a system comprising electric motor, speed reducer and screw-nut mechanism have been built based on the previous work done by Prof.Amati and Prof.Tonolli[6]. The second objective of this thesis is to perform the reliability test of such electromechanical height adjustment suspension prototypes, to analysis the results, improving the system from failures and giving suggestions for the future improvements.

From the result of Prof.Amati's paper of 'Design of Electromechanical Height Adjustable Suspension'[6], the benefits of adjusting vehicle height are analyzed by using a vehicle model including the longitudinal vehicle dynamics.

An average A-segment vehicle was chosen for the analysis. The vehicle data are given in Table1.1. The engine torque and the specific fuel consumption maps implemented in the model are experimentally measured by the manufacturer.

The aerodynamic drag coefficient of the vehicle at nominal road clearance is 0.35. It has been measured numerically and experimentally, that decreasing the road clearance by 20 mm and 40 mm, this coefficient will reduce to the values indicated in Table 1. Running the vehicle simulator at different road clearances, fuel

Table 1.1. The vehicle data used in the simulations

Parameters	Value
Vehicle mass	1090 kg
Engine volume	900 cm^3
Nominal power	60 kW
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

consumption and CO2 emission values for New European Driving Cycle (NEDC) and Worldwide Harmonized Light vehicles Test Procedures (WLTP) are obtained. During the simulations, the value of the aerodynamic drag coefficient is decreased from nominal to one of the reduced values (0.33 or 0.31) only on extra urban parts of the homologation cycles. Fuel consumptions obtained by the simulations are then converted in CO2 emission by using the conversion factor suggested in EC Regulation No 443/2009 for gasoline engine (i.e., 2330 gCO2 per one l of petrol). The results of simulations are depicted in Fig.1.4.

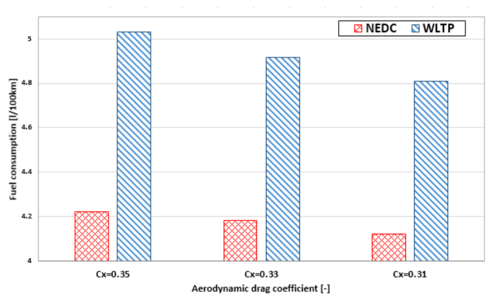

Figure 1.4. Vehicle fuel consumption at different values of aerodynamic drag coefficient while driving on two homologation cycles[6]

The fuel consumption reduction at different road clearances compared to the nominal value is in the range of 0.037-0.22 l/100km (0.88 - 4.41 , which corresponds to reduction of CO2 emissions by 0.87 - 5.17 g/km. These results justify the use of height adjustment systems on vehicles to achieve lower CO2 emissions. Besides that, added benefits include possible reductions of body roll and improved vehicle accessibility. By rising the vehicle height, higher versatility,i.e. adaptability of the vehicle to different road conditions can be achieved.

Chapter 2

State of the Art

By vehicle suspension we mean a mechanism that links the wheel directly to the body or to a frame that attached to it. This second solution is adopted by most vehicles, while the first, widely found in horse carriages in the past, is now used on particularly slow vehicles. In many cases, to suspension deformation must be added structural deformation, which plays an important role in the handling and comfort characteristics of a vehicle. So the roles of the suspensions are:[3]

- Allow a distribution of forces, exchanged by the wheels with the ground, complying with design specification in every load condition
- Determine the vehicle trim under the action of static and quasi-static forces
- To absorb and smooth out shocks that are received by the wheel from road irregularities and transmitted to the body.

The main components of a generic suspension systems are bearing components, primary elastic members, secondary elastic members and damping members. Bearing components are part of the mechanism linking the wheel to the body and guarantee the degrees of freedom of the wheels and their correct positions with reference to the vehicle body; they also transfer to the body part of the load delivered from the tire contact patch.Primary elastic members include springs(coil, bar and leaf springs), anti-roll bars and stop springs, these members connect to the wheel to the body elastically and store the energy produced by an uneven road profile or, in another word, to isolate the vehicle body from impacts coming from road irregularities.Secondary elastic members are elastic bushings on linkage joints and it is related to the elasto-kinematic behaviour of the suspension and its comfort properties.Damping members are basically shock absorbers, they are provided to waste the elastic energy stored by the elastic members and to allow the oscillation damping of the vehicle body, avoiding stationary vibration or resonance. Considering elastic and damping systems, suspension can be classified as passive, semi-active, and active. In passive suspensions the elastic reaction of the suspension system is determined solely by its deformation and the damming system can only waste part of the received energy.In semi-active suspensions the amount of damping can be turned and the damping ratio can be adjusted based on the measured data from vehicle sensors[4][5] .In active suspensions the system can receive energy from other sources(the engine, or some intermediate storage of engine energy) to affect body motion, with the objective of limiting this motion close to its static equilibrium condition.In general the semi-active suspensions required only a small amount energy used to drive the electronically controlled element.Vehicle height adjustable suspensions are classified into active suspensions with small actuation bandwidth.[6]

Essentially, height adjustment is a feature that allow to vary relative distance between the vehicle body and the wheel hub.This can be mainly realized by acting on different parts of the suspension strut such as on upper spring seat, on lower spring seat or on shock absorber tube(telescopically). Height adjustment system can be classified to: pneumatic, hydraulic, hydro-pneumatic and electromechanical by their actuation mode. The state of art of different typologies of height adjustment systems are covered in detail in the following sections.

2.1 Hydraulic

Hydraulic height adjustment system usually comprises of linear (hydraulic cylinder) or rotary (hydraulic motor) actuators, hydraulic pump (standalone or combined with already existing one), pipes and hydraulic fluid, control valves and sensors. Relative displacement of the piston and cylinder of the actuator unit allows to modify the height of the vehicle.

In order to stably control vehicle postures varying during a travel of the vehicle and to improve riding quality and stability, Hyundai Motor Company[7] has developed a vehicle height controlled shock absorber with a monotube-type cylinder configuration, which is advantageous in weight, size, manufacturing costs, and securing installing space.

As shown in Fig.2.1, the shock absorber apparatus include an oil pump **21** for generating a hydraulic pressure, an accumulator 23 storing an oil and installed to be connected to the oil pump via a connecting pipe line, a solenoid valve 24 including a spool 24a selectively slid in accordance with a controller, a supply pipe line 25 connecting the oil pump and the solenoid valve, a first return pipe line 26 and a second return pipe line 27 connecting the accumulator and the solenoid valve, and a first valve pipe line 28 connecting the upper chamber of the cylinder and the solenoid valve and a second valve pipe line 29 connecting the lower chamber of the cylinder and the solenoid valve, wherein the first valve pipeline or the second valve pipe line is selectively connected to the oil pump or the accumulator by the movement of the spool. Fig.2.2 illustrates a state where a height of the vehicle is raised the oil pump

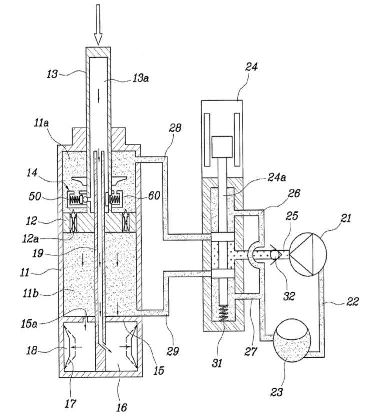

Figure 2.1. Hyundai height adjustment system in normial position

21 is operated and the spool 24a of the solenoid valve 24 is lowered while overcoming a force of the solenoid valve spring 31, and accordingly, the supply pipe line 25 and the second valve pipe line 29 are connected to each other and the first return pipe line 26 and the first valve pipe line 28 are connected to each other.

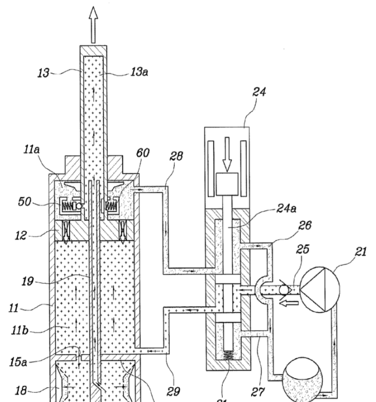

Figure 2.2. Hyundai height adjustment system in raised position

Fig.2.3 illustrates a state where a height of the vehicle is lowered, at this time, the oil pump 21 is operated and the spool 24a of the solenoid valve 24 is raised. Accordingly, the supply pipe line 25 and the first valve pipe line 28 are connected to each other, and the second return pipe line 27 and the second valve pipe line 29 are connected to each other.

DaimlerChrysler AG [8] has patented an active suspension system for vehicles,

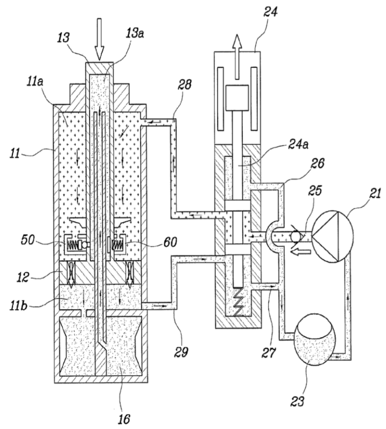

Figure 2.3. Hyundai height adjustment system in lowered position

has active supporting assemblies which are arranged between the vehicle body and the vehicle wheels.The supporting assemblies are controlled as a function of body accelerations of vehicle, and low-frequency and higher-frequency fractions of body acceleration, which are taken into account differently. As a result, the vehicle follows the road profile comparatively directly while the comfort is good. As shown Fig.2.4, a supporting assembly 3 is arranged between a wheel 1 and body 2 of a vehicle, which supporting assembly 3 in each case has a passive spring 4 as well as a hydraulic assembly 5 arranged in series thereto.By means of the hydraulic assembly, the body-side abutment of the spring can be vertically adjusted relative to the vehicle body 2.A shock absorber 6 is arranged in parallel to the spring 4. By way of control valve arrangement 7, the hydraulic assembly 5 is controllably connected with a hydraulic pressure source 8. This way, for example, be a pressure reservoir which is regularly recharged by a pump which is not shown. The control valve arrangement 7 is operated by an electronic control circuit 10. In Fig.2.5, 11 is a path generator assigned to each wheel 1 or each supporting assembly 3, the signal of the path generator 11 being correlated with the distance between the respective wheel 1 and the vehicle body structure 2; 12 is a acceleration sensor that vehicle-body-side vertical acceleration can be detected; 13 and 14 are the sensors which can directly record longitudinal and lateral accelerations of the vehicle body 2. Together with these sensors and several electronic device, the circuit 10 is able to obtain signal fractions which are defined by the motion velocities of the vehicle body 2 and signal fractions which are determined by there outer moments and forces acting upon the vehicle. So

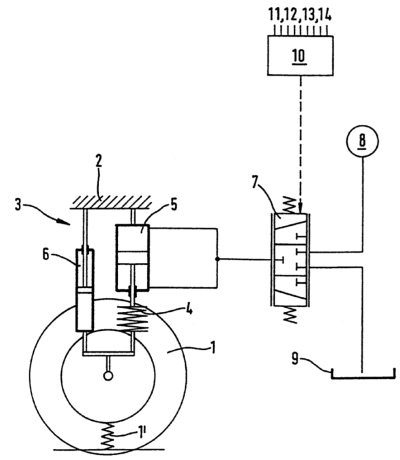

Figure 2.4. DaimlerChrysler AG active suspension system detailed view

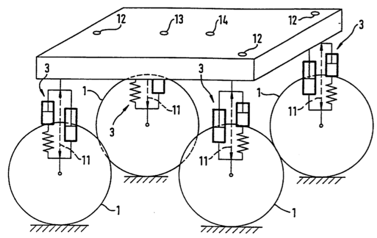

Figure 2.5. DaimlerChrysler AG active suspension system overview

finally the signal fractions which are defined by the deviation of the actual position of the vehicle body relative to the ground with respect to a corresponding desire position. Then the signals are transformed into electronic control current for the control magnets of the control valve arrangement 7 of the wheels 1 of the vehicle. At the end, the system is capable to, for example, when a vehicle body acceleration take place in the upward direction, the output signals represent commands for shortening one or several of hydraulic assemblies 5 and when a vehicle body acceleration in the downward direction, the output signals represent commands for lengthening of one or several of the hydraulic assemblies 5. As a result, the vehicle follows the road profile comparatively directly while the comfort is good.
The Mercedes-Benz AG [9] has developed a height adjusting shock absorbers strut for a motor vehicle wherein a shock absorber damping cylinder is operatively attached to the vehicle wheels and a piston rod interacting with the damping cylinder

extends from the damping cylinder and is provided are its free end with a mounting structure for attachment to a vehicle body and a spring is disposed between the damping cylinder and a spring support disposed on the piston rod, the spring support structure comprises a support member which is axially movable mounted on the piston rod for controlling the support location of the spring and, consequently, the height of the strut under vehicle load. As shown in Fig.2.6 .The shock absorber strut

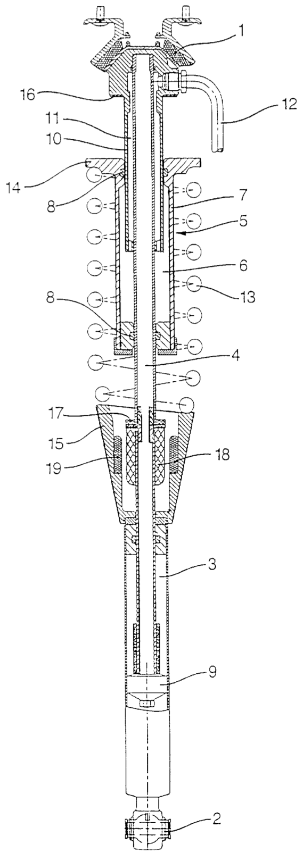

Figure 2.6. Mercedes-Benz AG height-adjustment shock absorber strut detailed view

has a body-side rubber support joint 1 supporting a vehicle body (not illustrated) and a wheel-side joint 2 on a wheel-support component (likewise not illustrated), such as an axle. The strut includes a damping cylinder 3 of a shock absorber adjacent the wheel-side joint 2, while the piston rod 4 of the shock absorber is connected to the body-side support joint 1 .The piston rod 4 projects from the damping cylinder 3 over a relatively large rod length. within this rod length, a hydraulic height adjustment device 5 is disposed on the piston rod 4. The purpose of this hydraulic adjusting device 5 is to change the body height of the vehicle and to adapt to the respective current operating conditions. By means of such an adjusting device, it is possible to adapt to yaw and rolling movements and to compensate for inclined positions of the vehicle body. The adjusting device 5 has a cylindrical housing enclosing an operating space 6 and supported on the piston rod 4 so as to be axially movable whereby the volume of a hydraulic operating space 6 is changed. The housing 7 is sealed at its axial ends relative to the piston rod 4 by means of annular seals 8 to provide a sealed termination of the hydraulic operating space 6. The lower one of these two seals 8 is disposed in an area of the piston rod 4 in which the latter has a first diameter rod portion which remains the same all the way to a displacement piston 9 at the end of the piston rod 4 inside the damping cylinder 3. The upper seal 8, in contrast, is disposed around the piston rod 4 in an area with a larger second diameter of the piston rod. The larger second diameter is formed by an axially extending sleeve structure 10. An annular gap 11 between the sleeve structure 10 and the piston rod 4 forms a hydraulic fluid supply passage to and from the hydraulic operating space 6. The annular gap 11 is in communication, via a line12, with a controlled or regulated hydraulic fluid storage space(not illustrated).

By changing the volume of the hydraulic operating space 6, the housing 7 moves along the piston rod 4 and, at a given load, thereby causes the distance between the two joints 1 and 2 to change. With the volume of the hydraulic operating space 6 kept constant, the cylindrical housing 7 of the hydraulic adjusting device 5 is functionally fixed to the piston rod 4.

During up and down movement of a wheel and a corresponding deflection or rebound of the spring 13, only the damping cylinder 3 moves with the vehicle wheel while the piston rod 4 remains virtually at rest. With the adjusting device 5 inactive, also the cylinder 7 and the collar 14 remain stationary with respect to the rod 4.

The cylindrical housing 7 of the hydraulic adjusting device 5 can be axially displaced by changing the volume of the hydraulic operating space 6 between the stops 16 and 17 of the piston rod 4. In this area, the height of the vehicle body, that is to say the distance between the two joints 1 and 2, can be changed. To limit the oscillation amplitude of the shock absorber, a further padded stop 18 is provided on the piston rod 4 opposite the stop 17 which can strike against the damping cylinder 3. A padded stop 19 serves to engage the bottom of the cylindrical housing 7 to limit upward movement of the damping cylinder 3 in order to prevent any contact between the spring coils, which might otherwise be possible at the maximum stroke of the adjusting device 7, that is, with a full load, and maximum upward movement of the damping cylinder3.

Daimler-Benz AG [10], patented a suspension system for motor vehicles has each wheel assigned a spring and a hydraulic unit and arranged in series with the latter. The ground clearance of the vehicle body can thus be altered wheel-by-wheel. A hydraulic feed system enables hydraulic fluid to be displaced between hydraulic units and allows pitching and rolling movements of the vehicle body to be counteracted. As seen in Fig.2.7, a first cylinder 4 is arranged on the body-side joint 2 and accommodates in telescope-fashion a second cylinder 5 arranged on the wheel-side joint 3, i.e. the second cylinder 5 can be displaced axially within the first cylinder 4. The second cylinder 5 forms the cylinder component of a controllable shock damper which has a piston rod 6 held rigidly on the body-side joint 2. During axial stroke movements of cylinder 5 relative to cylinder 4, the piston rod 6 thus remains virtually at least, stationary relative to the cylinder 4. Arranged on the outside of

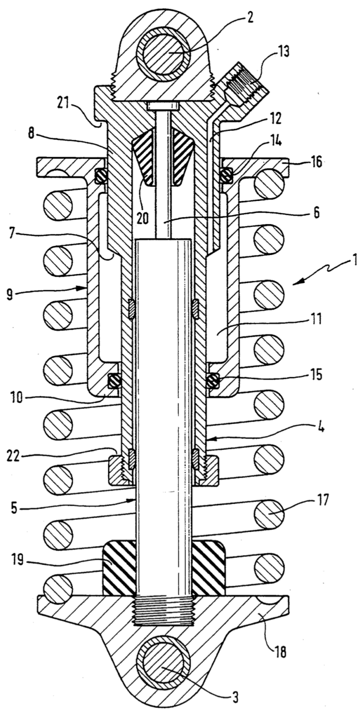

Figure 2.7. Daimer-Benz AG height-adjustment system partial cross section view of spring sturt

cylinder 4 is a piston-like thickened portion 8 which forms an annular step 7 and on which is guided, in axially displaceable fashion, a third cylinder 9. At the end at the bottom in Fig.2.7, the cylinder 9 has a base 10 in the form of a circular disc with a central opening for the cylinder 4. Formed axially between the base 10 in

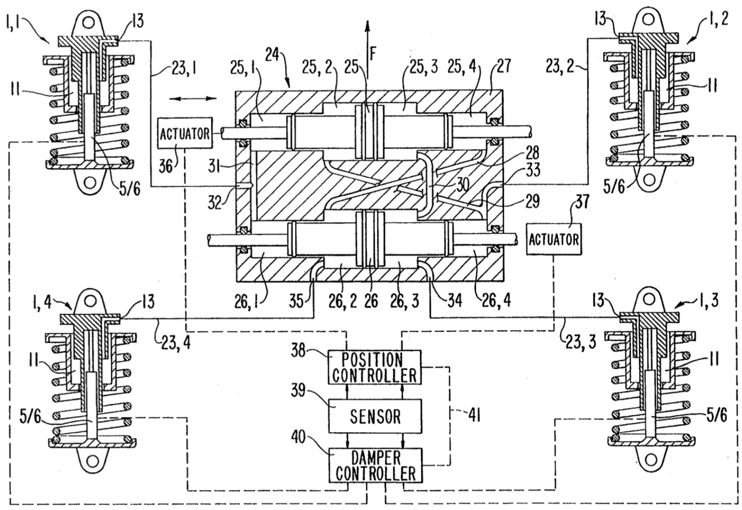

Figure 2.8. Daimer-Benz AG schematic representation of overall height-adjustment system

the form of a circular disc and the annular step 7 is an annular space 11 which can be supplied with hydraulic fluid via an axial hole 12 in the thickened portion 8 and via a hydraulic connection 13 adjoining the said thickened portion. Sealing rings 14 and 15 for sealing off the annular gaps between the thickened portion 8 and cylinder 9 and between cylinder 4 and the central hole in the base 10 shut off the annular space 11 from the outside in a hydraulically sealed manner. By feeding hydraulic fluid into the annular space 11 or by discharging hydraulic fluid from this annular space 11, the cylinder 9 and the body-side abutment formed by the flange 16 for the helical compression spring 17 can be displaced-relative to the body-side joint 2 and the ground clearance of the vehicle body at the respective spring strut 1 changes accordingly. The displacement travel of cylinder 9 in the upward or downward direction relative to cylinder 4 is limited by stops 21 and 22 in the form of annular steps. The maximum displacement of the cylinder 5 relative to the cylinder 4 is limited by the maximum displacement travel of the piston (not shown) fixed on the piston rod 6 of the shock damper, the cylinder compartment of which is formed by the cylinder 5.

Assume in Fig.2.8that a vehicle (not shown specially) traveling in the direction F of forward travel has a total of four spring struts 1,1 to 1,4 of the type depicted in Fig.2.7to support the vehicle structure on the wheels. Spring struts 1,1 and 1,2 are assigned to the front wheels, and spring struts 1,3 and 1,4 are assigned to the rear wheels. The hydraulic connections 13 of these spring struts 1,1 to 1,4 are connected via lines 23,1 to 23,4 to a hydrostatic feed unit 24 which, in the manner explained below, makes it possible, on the one hand, to displace hydraulic fluid between the annular spaces 11 of the front spring struts 1,1 and 1,3 on the left-hand side of the vehicle 1.As a result, the vehicle body tries to perform a pitching movement relative to the vehicle transverse axis or a rolling moment relative to the vehicle longitudinal axis, respectively. The piston working spaces 25, 1 to 26,4 are connected to one another via lines 28 to 31 and via connection 32 to 35 to lines 23, 1 to 23,4 so that, when stepped piston 25 is displaced, hydraulic fluid is displaced between the annular spaces 11 of spring struts 1,1 and 1,3, on one hand, and those of spring struts 1,2 and 1,4, on the other hand. When stepped piston 26 is displaced, however, hydraulic fluid is displaced between the annular spaces 11 of spring struts 1,1 and 1,2 on one hand, and those of spring struts 1,3 and 1,4, on the other hand. As a result, by displacement of stepped piston 25, of counteracting a pitching movement of the vehicle body dan be counter acted during braking or during starting, while a dis placement of stepped piston 26 allows rolling movements of the vehicle body to be counteracted. The stepped pistons 25, 26 or, more precisely, their piston rods, are actuated by respective actuating units 36, 37 which, for their part, are controlled by a position controller 38. The input of the position controller 38 is connected to body and/or wheel sensors 39, from the signals of which the position controller 38 can deter mine how the vehicle body is moving or is attempting to move. The sensors 39 can, for example, be stroke transmitters whose signal represents the retraction travel of piston rod 6 in cylinder 5. Body movements and wheel movements can then be determined from the change in these signals. The hydraulic height adjustment systems are advantageous as they deliver control force at first rate (hence, allow fast leveling and provide active feature), main components of the system can be fixed on vehicle sprung mass (thus, unsprung mass modification is little) and have compact installation sizes. Since the hydraulic fluid present in the system, moisturization of the fluid must be avoided to prevent metallic components of the system from corrosion. This requires use of special fluid and, furthermore, tight tolerances manufacturing the cylinder-piston group.These constitute the main drawbacks of hydraulic suspension height-adjustment systems, that can be minimized leading to increased costs[6].

2.2 Pneumatic

Pneumatic suspensions are mainly used in public transportation buses, heavy-duty vehicles and service cars where increased passenger comfort is required. They feature height adjustment systems to allow easy enter/exit of passengers. Fig.2.9(left) lists main components used in pneumatic suspensions. They consist of air springs (usually, reinforced rubber sleeve/bag inflated with pressurized air), high pressure reservoir (tank), air compressors, air dryer (to remove moisture) and control system (including control valves, height and pressure sensors). Adjustment of the ride height is straight- forward and can be accomplished by pumping the air into or out of air spring bag. Hyunsup Kim and Hyeongcheol Lee have proposed a new nonlin-

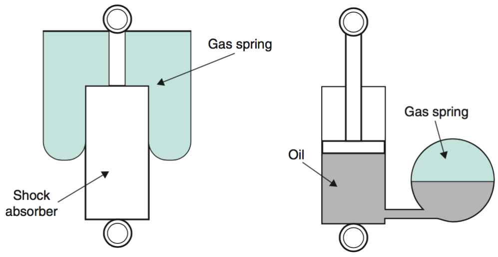

Figure 2.9. Schematic representation of a gas spring implemented with pneumatic spring (left) and with hydro-pneumatic spring (right).

ear controller to adjust the height of the vehicle sprung mass (height control) and to regulate the roll and pitch angles of the vehicle body (levelling control) by the air suspension system[11]. A sliding mode control algorithm is designed to improve the tracking accuracy of the control and to overcome nonlinearities and uncertainties in the air suspension system. The system layout is represented in Fig.2.10 As shown in figure, the air suspension system includes four air springs installed at each corner of the vehicle, four air spring displacement sensors, four air spring valves regulating air flows into the air springs, a compressor, two reversing valves deciding the air path, a reservoir storing the compressed air, a pressure sensor measuring the inlet pressure of the air spring valves, and air filter removing dirt in the circuit, and an air dryer absorbing moisture from the intake air to prevent moisture damage. The

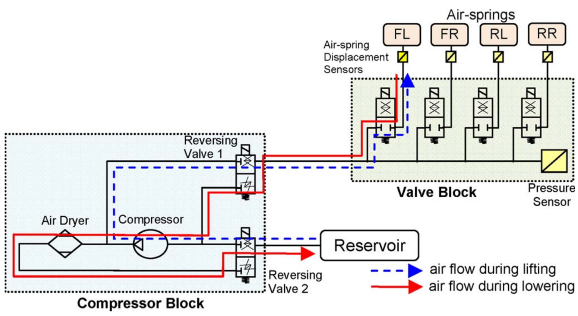

Figure 2.10. Diagram of the air suspension system for height control.

lifting or lowering procedure of the target air suspension is as follows:

- Lifting procedure: The dashed line shows how to build-up pressure in the air springs to lift the sprung mass. The compressed air flows from the reservoir to the valve block through the reversing valve 2, the compressor, and the reversing valve 1. Each air spring valve regulates air flow into the corresponding air spring in an on-off manner.
- Lowering procedure: the dotted line shows how to reduce pressure in the air spring to lower the sprung mass. The compressed air flows from the corresponding air spring to the reservoir through the reversing valve 1, the compressor, the air dryer and the reversing valve 2.

The main force of the air suspension is the height control and the leveling control to facilitate passenger entry and exit, reduce the air drag and driving safety, for these purpose, the air suspension has several operation modes: Easy entry mode: offering convenience during entry into and exit from the vehicle by lifting the front and rear sides;Normal mode: normal driving state and offering a reference vehicle height for other modes;Highway mode: reducing air drag and enhancing safety during high-speed driving by lowering the front and rear sides;Off-road mode: enhancing ride comfort and protecting the vehicle bottom during driving off-road by lifting the front and rear sides; Easy loading mode: offering convenience when loading or unloading the trunk by lowering the rear side;Parking mode: looking elegant during parking by lowering the front and rear sides. The desired heights at each mode is shown in table 2.1.

Table 2.1. Desire heights at each mode

Mode	Front Side Height	Rear Ride height
Easy entry mode	+20mm	+20mm
Normal mode	0mm	0mm
Highway mode	-15mm	-15mm
Off-road mode	+15mm	-15mm
Easy loading mode	0mm	-30mm
Parking mode	-10mm	-10mm

Audi A6 '98 4-level air suspension system in the allroad quattro[12][13], the 4-level air suspension system is a fully- supporting level control system with conventional shock absorbers at the front axle (Fig.2.11) and load dependent shock absorbers (PDC dampers) on the rear axle(Fig.2.12). The vehicle level is determined separately at each axle side by means of 4 level sensors. Each air suspension strut is allocated a so- called air spring valve (transverse check valve) so that each axle can

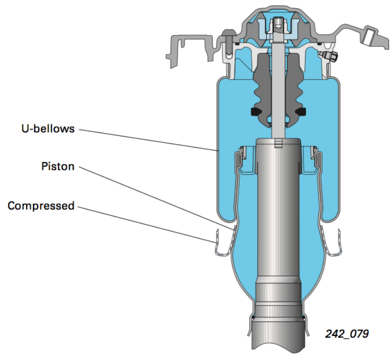

Figure 2.11. Front suspension strut in the Audi allroad quattro

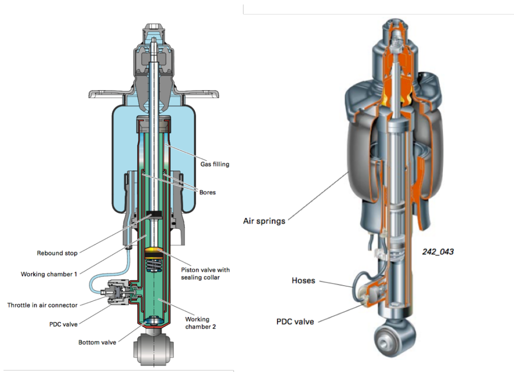

Figure 2.12. Rear suspension strut in the Audi allroad quattro

be controlled individually. The height-adjustment feature is realized by pneumatic system consisted by several component as shown in Fig.2.13 .By switching the valves position, air could charge or discharge the air spring, hence adjust the height of vehicle. One of the special features of the system is the ability to alter the ground clearance by 66 mm in 4 stages as shown in Fig.2.14. The 4 stages can be controlled manually or automatically. The levels are designated as follows:

- Level 1 = low level (LL)
- Level 2 = normal level (NL)
- Level 3 = high level 1 (HL1)
- Level 4 = high level 2 (HL2)

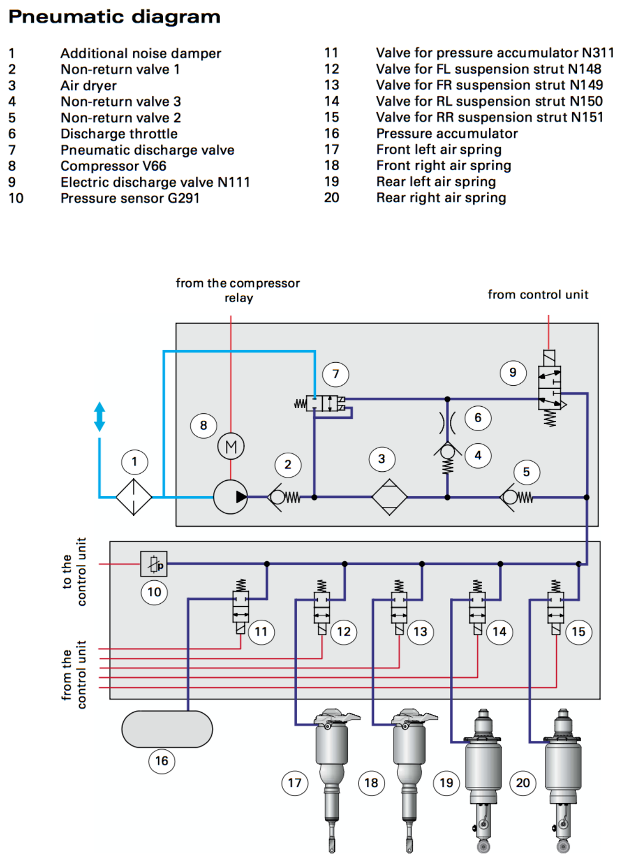

Figure 2.13. Audi allroad quattro 4-level pneumatic diagram

- Parking level PL = high level 1

The stages selection is decided by the control strategies(Fig.2.15).

- Automatic lowering: The following preconditions relating to driving speed apply for high level 1 and high level 2. If the vehicle is at high level 2, it will lower automatically to high level 1 at a speed of >35 km/h. The system will respond to a request to shift to high level 2 only up to a speed of <30 km/h. At a speed of >80 km/h in high level 1, the system will automatically lower the vehicle to normal level. The system will respond to a (manual) request to shift to high level 1 only up to a speed of <75 km/h. During driving operation, no automatic raising to high level 1 or 2 is performed. It must be selected manually by the driver. The parking level is an exception. In this mode the vehicle automatically rises to high level 1 once it has been parked and locked.

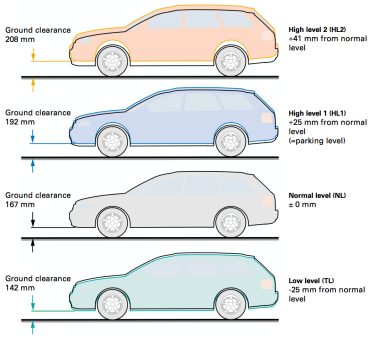

Figure 2.14. Audi allroad quattro 4-level air suspension system stages

- Motorway mode: If the vehicle travels for longer than 30 seconds at over 120 km/h (vehicle is already at normal level), it will automatically be lowered to low level. This reduces air resistance (saves fuel) and lowers the vehicle's centre of gravity (improved driving dynamics). The vehicle rises automatically to normal level at the following speeds and time thresholds (table2.2):

Table 2.2. Speed and time thresholds in Motorway mode

Vehicle speed	Time
<70 km/h	>120 seconds
<35 km/h	>30 seconds
<5 km/h	Immediately

- Parking level control: The parking level ensures that the vehicle is maintained at a suitable level after parking for a long period of time (normal volumetric reduction due to cooling or diffusion). It also facilitates entering and loading the vehicle and optimises the appearance of the stationary vehicle. The parking level corresponds to high level 1 (HL1). The vehicle is set to parking level:

  — When the system is in run-on mode and the vehicle is locked from the outside.

  — When sufficient pressure is present in pres- sure accumulator.

  — When the system is not switched to manual mode.

  PL(=HL1) is only canceled when a driving speed of 80 km/h is exceeded or when switching to a lower level manually. If the vehicle is already in HL2 it is not lowered to parking level.

- Manual mode: The motorway mode and parking level control functions are deactivated in manual mode.

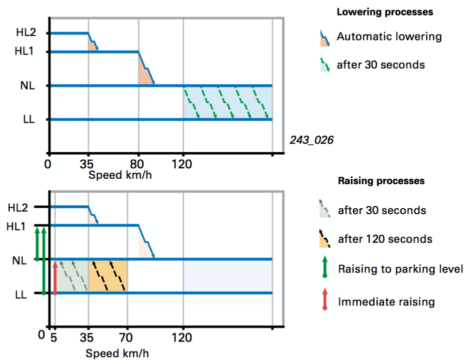

Figure 2.15. Audi allroad quattro 4-level air suspension raising and lowering processes

This 4-level air suspension system offers advantages as:

1. The 4-level air suspension is a sophisticated electronically controlled air suspension system at both axles. The system enables variation of the floor level by 66 mm and offers four defined height levels with between 142 and 208 mm ground clearance.

2. Depending on the driving conditions and requirements you can select greater ground clearance or a low vehicle centre of gravity and a good cw value.

3. The 4-level air suspension keeps the pre-set vehicle level constant, regardless of load and weight distribution.

4. The 4 level stages can be controlled manually or automatically within defined limits.

5. Individual automatic functions or the entire system can be switched off via the control system.

6. LEDs in the operating unit indicate the operating status and the control procedures to the driver.

7. Pressure accumulator system for maximum comfort.

Toyota has developed an electronically controlled air suspension system capable of meeting mutually exclusive requirements such as soft riding and stable maneuverability on higher levels[14]. This modulated air suspension system applied to the Toyota Soarer marketed in Jan.1986, uses electronic technology to control vehicle height, spring rate, and damping force, thus achieving automatic control of suspension system characteristics. The system consists of three control steps. First is the detection of vehicle traveling conditions, second is the classification of the detected traveling conditions into one of several preset patterns, and third is to adjust vehicle height, spring rate, and damping force so that they will correspond to the characteristics designed for the pattern detected. The controls performed for each pattern are outlined below:

1 Braking at high speed, sharp steering and rapid starts are detected by the vehicle speed sensor, stop lamp switch, steering sensor, and throttle position sensor, and spring rate and damping are adjusted upward when these phenomena are detected. This minimize changes in vehicle attitude.

2 During high speed cruising, height is lowered and both spring rate damping force are increased to improve driving stability.

3 The presence of rough road conditions is detected by speed and height sensors, and height is increased and the spring rate and damping force are increased higher to prevent bottoming.

4 Bumps on roads are detected by the speed and height sensors and spring rate and damping force are lowered to reduce bumps.

Fig.2.16 shows the system configuration, in addition to the sensors and actuators discussed above, other major components include a system controlling ECU, a suspension unit, and a mode select switch used to select control mode. Both of compressor and height control valve which used to open and close the air passages are to adjust height. Actuators driven by dc motors are used to adjust spring rate and damping force.The configuration of the height control valve is shown in Fig.2.17. For air spring and shock absorber unit, the basic construction for both the front and rear is identical. They consist of the shock absorber and the air chamber, consisting of the main and sub chambers, as shown in Fig.2.18. spring rate is controlled by changing the air passage area in the rotary valve which is provided between the main and sub air chambers. The rotary valve with orifice is used to control the damping force of the absorbers. An actuator drives the rotary valves of the shock absorber and air chamber and is constructed to drive the two rotary valves using a

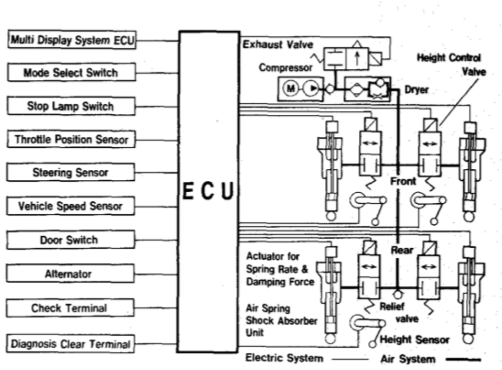

Figure 2.16. Toyota electronic modulated air suspension system configuration.

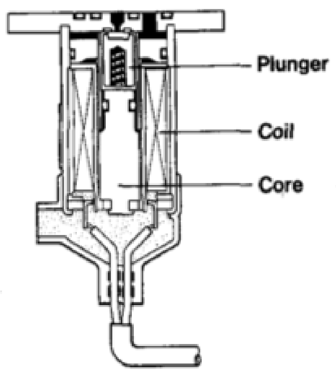

Figure 2.17. Toyota electronic modulated air suspension system height control valve.

single motor, and the figure 11 shows the configuration of the actuator. The system provides two control modes, NORMAL AUTO and SPORT AUTO, for spring rate and damping force, and another two control modes, NORMAL AUTO and HIGH AUTO, for vehicle height control mode. For vehicle height control, it functions to match the actual vehicle height with the target height, which is set according to the vehicle speed and road conditions. The target vehicle height is selected from the three height levels:LOW, NORMAL, and HIGH. The target height during low-speed traveling on a flat road is taken as the basic height and height level is set according to the selected control mode: NORMAL: height level is set when the control mode is in the NORMAL AUTO position and HIGH: height level is selected when the control mode is in the HIGH AUTO position. The target height is changed by the

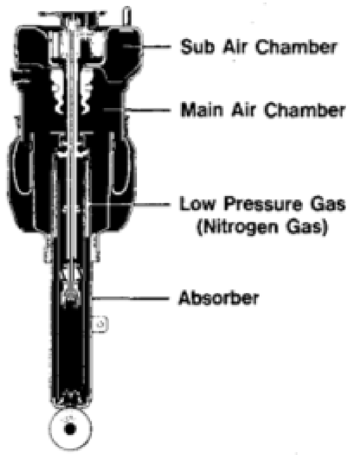

Figure 2.18. Toyota electronic modulated air suspension system air spring and shock absorber unit.

traveling conditions in the following two cases:

- During high-speed traveling, target height is lowered by one level to improve stability and aerodynamic characteristics. This change is executed at a vehicle speed higher than 90 km/h taking general traveling patterns and road conditions into consideration. After the target height has been changed, return to the basic height level occurs when vehicle speed is lowered to 60 km/h. Hysteresis of 30 km/h is provided in the control characteristics of prevent frequency occurrence of height level changes during traveling.

- During traveling on rough roads, target height is changed one level higher when the vehicle speed exceeds a preset valve so that bottoming is reduced and drivability on rough roads is improved. The vehicle height signals from the left front wheel are used to detect rough roads. Vehicle height is changed higher when traveling on rough roads with remarkable angulation and suspension movement under such conditions is close to the natural oscillation of the sprung.

The air chambers in the suspension are, under normal conditions, separated from the air lines with the height control valves provided for individual chambers closed. When there arises a need to change vehicle height, these valves open to connect the chambers with the air lines. To increase the vehicle height, the pressure in the air lines is increased by operating the compressor with the exhaust height control valve closed. When vehicle height is to be lowered, the compressor is stopped and the exhaust valve is opened to lower the pressure in the air lines. The air pressure inside the chambers of the individual suspensions is adjusted to bring the vehicle height to the target height. Spring rate and damping force characteristics are changed at the same time. This change occurs for the four wheels at the same time also. This control of characteristics is applied in two ways before vehicle behavior and following such behavior.

1) For the control of the spring rate and damping force, change of the characteristics should be completed before the beginning of suspension stroke. This is especially necessary for one-shot behavior. The predictive control adopted to the system is shown in Table 2.3.

2) If the vehicle traveling conditions are detected from the vehicle behavior, the problem that the change of characteristics called for cannot be completed until after actual occurrence of such conditions is inevitable. In such cases, continuity of the conditions to be controlled is required to make the control effective. The tracking control implemented by this system is summarized in Table 2.4.

Dean.R.Tener[15] has developed a cocktail adjustable air suspension system.Conventional air suspension systems have been employed extensively in high duty truck and trailer applications, luxury passenger cars and some sport utility vehicles. While this cocktail adjustable air suspension system is suitable for sport compact vehicles, sport sedans, light trucks sport utility vehicles, off road vehicles, and many other vehicles traditionally equipped with metallic springs. Ride height is readily adjusted to improve handling, or to increase ground clearance. Spring rate may be lowered to improve isolation, or increased to increase handling performance. Because of the wide range of user adjustability, the vehicle can exhibit exemplary handling or a comfortable ride. The suspension system offers the following features:

1. Load levelling –ability to maintain ride height regardless of changes in loading condition.

2. Adjustable ride height-system can increase or decrease distance between ground and sprung mass when commanded by a cockpit mounted push bottom. This feature may provide improved handling through reduction in CG height, additional ground clearance, improved ingress/egress, and enhancement of vehicle appearance.

3. Adjustable ride and handling- the system offers considerable change in ride and handling performance through cockpit adjustment of overall spring rate.

Table 2.3. Spring rate and damping force control(predictive control)

Item	Sensor	Purpose
Anti-dive	Speed sensor Stop lamp switch	Before the vehicle attitude change begins suspension is changed harder to restrict attitude change
Anti-roll	Speed sensor Steering sensor	
Anti-squat	Speed sensor Throttle position sensor	
Anti-bump	Speed sensor Height sensor	Irregularity of roads is detected by the vertical movement of the front wheels and suspension is changed softer before the rear wheels pass through the detected irregularity to reduce shock

Table 2.4. Spring rate and damping force control(tractive control)

Item	Sensor	Purpose
Response to speed	Speed sensor	Suspension is set harder to improve travelling stability at high speed cruising. Since vehicle speed change is gradual tracking control has satisfactory effect
Response to rough road	Speed sensor Height sensor	Suspension is set harder to restrict pitching and bouncing on rough road

4. Adjustable understeer/oversteer balance – the suspension system can be designed to vary understeer level by adjusting front and rear axle spring rates

independently.

5. Automatic equalization of total weights on opposite diagonals.

The system consists of air springs, suspension position sensors, electric switch, computerized control unit, valve block, air compressor, desiccant and pneumatic accumulator. Rolling lobe air springs are used here, as can be seen from the Fig.2.19, this type of spring is characterized by a cylindrical rubber tube or sleeve, sealed at one end by a “cap”, and equipped with a piston at the other end. Spring rate of the rolling lobe spring is determined by air volume, piston diameter, and to a lesser extent, mechanical stiffness of the rubber sleeve. And rolling lobe air springs provide height control by admitting or exhausting air from the spring cavity. Spring rate can be controlled either simultaneously with the operating height or independently. There are two ways to control the spring rate. First one is with the constant piston

Figure 2.19. Rolling Lobe Air Spring

area, and by adding or removing air. Application of the general process equation for isentropic compression of air: $P_1/P_2 = (V_1/V_2)^{1.3}$.This equation explains how rapidly compression of a fixed air mass causes an increase in pressure. So the spring rate is altered by the variation of air volume. The second method is use variable piston area. Piston geometry has a large effect upon spring rate. Taper can be used to affect both the initial rate and the progressive nature of the spring. The piston is designed with several different zones of contact:

1. Digressive taper for soft ride.
2. Progressive taper for increasing rate.
3. Large diameter cylindrical shape for high final rate.

Although three rates are defined but the spring is a continuously variable device. The system consists: four air springs, air compressor, air reservoir, ECU, height sensors, valve block, and accelerometer. The height sensors at each can provide feedback of vehicle ride height and ECU determine the estimated vehicle height, roll and pitch angle. The measured body position is compared to the most recent command, if there is any difference, air is admitted or exhausted from the suspension springs.

The main advantage of pneumatic suspensions is the possibility of controlling the spring rate of the air springs to find a good compromise between improved handling performance and ride comfort. Other advantages are those of intrinsically owning the height adjustment feature and a relatively fast actuation speed in presence of stored pressurized air. However, high cost, reduced reliability and robustness (mainly due to failure of rubber air springs) and high maintenance requirements are the main drawbacks of this kind of suspension systems. Moreover, pneumatic suspensions are less efficient when significant changes of the loading conditions take place or fast and frequent height adjustment are required (especially, when pressurized air store is depleted). Due to high cost of the components used in pneumatic suspensions, they are mainly used on top E and F segment models[6].

2.3 Hydro-pneumatic

Hydro-pneumatic suspensions use elements of both hydraulic and pneumatic suspensions. The hydraulic part of the system is responsible for providing damping and can be used for height adjustment, while pneumatic part with high pressure accumulator provides elastic properties to the suspension. Simple schematics of the system is depicted in Fig.2.9(right). Industrial pioneer in using hydro-pneumatic suspension has been Citroen, applying it on the vehicle back in 1954 [16]. Nowadays, the same system with some upgrades on control logic and new components is used. In the system proposed by Citroen, a non-preloaded, hydropneumatic suspension system with single-acting cylinders and accumulators directly mounted to them. In the fluid path between cylinder and accumulator a damping unit is located, which acts similarly to a damping unit of a monotube-damper. Both suspension cylinders are hydraulically connected so the entire roll stability is provided by the mechanical stabilizer bars at the front and the rear axle. The system is equipped with two level control units which are assigned to the front and rear axle. Fig.2.20 shows a simplified representation of the setup using a level controlled axle suspension. It is the task of a level control valve to react to a deviation from the suspension design position with a supply/release of hydraulic fluid to/from the suspension circuit, to bring the suspension back to the desired position. A 3/3-way spool valve is used with the spool actuated by a connection rod. The connection rod, for its part, is connected to the axle and therefore is moved with every relative movement between input side and isolated side. Hence if the axle is moved towards the chassis (compression) the connection rod is pushed upward and the spool of the leveling valve

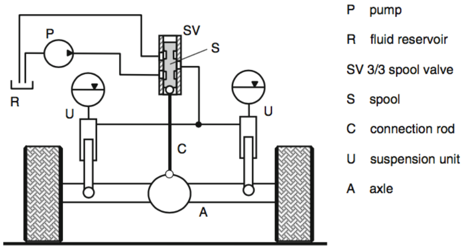

Figure 2.20. Basic setup of mechanically controlled leveling

with it. This opens a path in the valve from the pump to the suspension units, hydraulic fluid flows into the units and the suspension level is increased. On the other hand if the axle is moved away from the chassis, the spool is moved downwards and therefore connects the suspension units with the fluid reservoir – the suspension level is lowered. In the center position of the spool all three ports of the valve are separated from each other and the level is kept constant. The level control units are connected to a central vehicle hydraulic supply system which, for example, also feeds the brakes, the steering and gearbox and clutch. Fig.2.21 illustrates the schematic setup of the system.

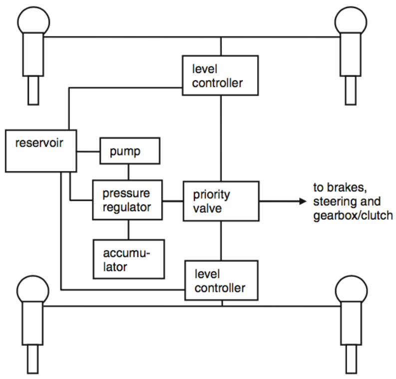

Figure 2.21. Schematic setup of the basic Citroen suspension system

Sarel F. van der Westhuizen, Pieter S. Els[17] studied an hydro-pneumatic suspension developed by Els[18] called "4S4" (4 state semi-active suspension system).

The 4S4 uses semi-active hydro-pneumatic springs and hydraulic dampers to give optimal ride comfort (soft springs and low damping) and to give optimal handling (hard springs and high damping) on its specific settings.

The 4S4 can switch between hard and soft spring settings and between high and low damping in less than 100 ms. Switching between the different settings happens automatically based on acceleration measured on the vehicle body. The switching is done by opening or closing solenoid valves. A schematic diagram of a 4S4 unit can be seen in Fig.2.22. The wheel displacement relative to the vehicle body results in a displacement of the piston in the main strut which is filled with oil, the oil then in turn is pushed through the damper packs or valves until the oil pressure and the pressure of the accumulator equalizes. If valve 3 is closed the suspension is on the stiff spring setting where only the 0.1 L nitrogen accumulator is used as a pneumatic spring. If valve 3 is open it is on the soft spring setting where both the 0.4 L and the 0.1 L accumulators serve as a pneumatic spring. If valves 1 and 2 are closed the system has high damping because the oil is forced to flow through the dam- per pack, and if they are open the system has low damping because the oil can flow past the damper pack through the open valve.

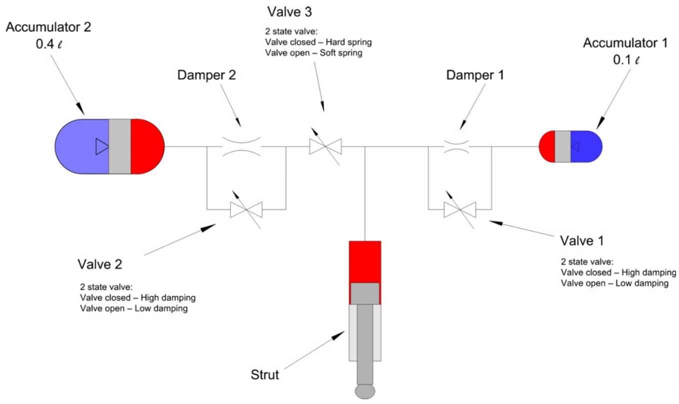

Figure 2.22. Schematic diagram of the 4S4.

Based on such solutions, the main advantages of using hydro-pneumatic suspensions can be highlighted. They have more compact size, faster adjustment speed, higher reliability and improved performance, however high production costs limit its use to high end segments.

2.4 Electromechanical

Starting from the mid 2000s, new solutions using electromechanical actuators and electronic control units started to appear in literature, specifically in form of patents. This could be the result of car manufacturers' attempts to find low cost height adjustment suspension systems applicable also on low cost car segments. So far, no solutions using electromechanical system are present on the market.

In principle, all the electromechanical height adjustment systems make use of linear translational part, usually by means of threaded screw-nut mechanism connected to the suspension components and actuated by electric motors directly or through speed reducer.(Fig.2.23)[6]

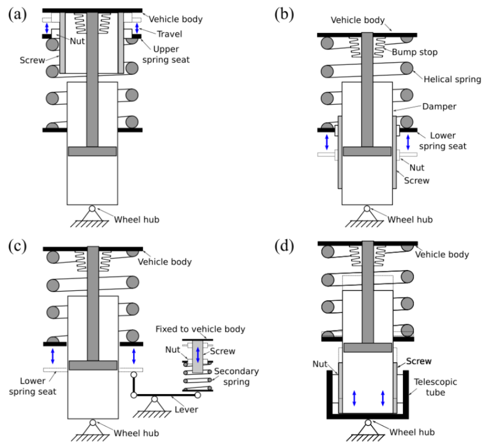

Figure 2.23. Electromechanical height adjustment system schematics applied to independent suspension: (a) Upper spring seat actuation; (b) Lower spring seat actuation; (c) Actuation by means of lever mechanism; (d) Shock absorber tube actuation.

Audi AG [19] has patented an electromechanical height adjustment on a wheel suspension for motor vehicles,having a shock absorber which has a piston rod,which shock absorber is dynamically connected to the vehicle body, and to which shock absorber a helical spring is coaxially assigned which is clamped between two spring plates that can be changed in their active distance and that determine the spring deflection and rebound path relative to a normal level, the body-side end of the shock absorber being assigned a shock absorber cap which influences the free spring path “s” (Fig.2.24) of the helical spring and which is made adjustable at the same time with the adjustable spring plate. A ball-screw is connected to an electric motor through reduction stage of gears and a nut, connected to the lower spring seat(Fig.2.25). The clearance between a rubber bump stop and the shock absorber tube is varied by connecting the surface to the translating nut to avoid over compression of the spring during bumps.

Honda Motor co.,Ltd. presented a solution of a height adjustment system using electromechanical actuation [20]. the vehicle height adjusting system comprising: a housing connected to said one of the vehicle end member and wheel end member;

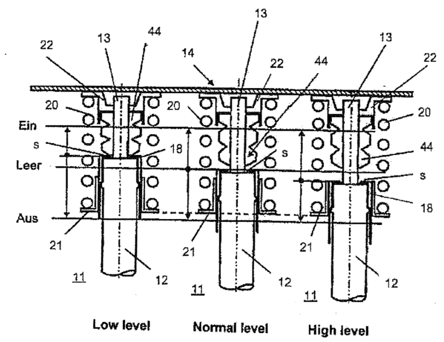

Figure 2.24. Spring paths for the deflection and rebound of the strut

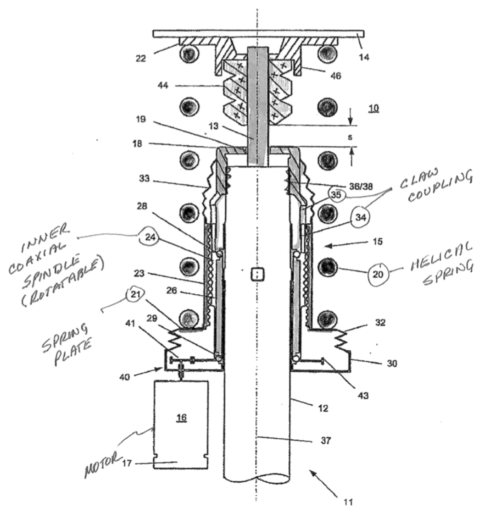

Figure 2.25. Detail of the strut

a spring seat retainer connected to said opposing end of the suspension spring; a first rotor rotatably supported by the housing and provided with a first driven gear along an outer periphery thereof;a second rotor rotatably supported by the spring seat retainer and provided with a second driven gear along an outer periphery thereof,the second rotor being coaxially disposed in relation with the first rotor around a common axial line and joined with the first rotor via a threading coupling that converts a relative rotation around the common axial line into a relative linear movement of the first and second rotors towards and away from each other along the common axial line;a drive shaft rotatably supported by the housing and provided with a first drive gear meshing with the first driven gear and a second drive gear meshing with the second driven gear; and a drive mechanism mounted on the housing for turning the drive shaft; a gear ratio between the first drive gear and first driven gear being different from a gear ratio between the second drive gear and second driven gear. Thereby, as the drive shaft turns, the first and second rotors are made to turn at a much lower rate so that the rotation of the drive mechanism is transmitted to the threading coupling at an extremely high gear ratio with a minimum torque loss and the torque required for actuating the spring seat retainer relative to the housing can be provided by using a highly compact motor that consumes very little electric power. In particular, because the first and second rotors are disposed in a mutually coaxial relationship, a highly compact and simple arrangement is possible.(Fig.2.26)

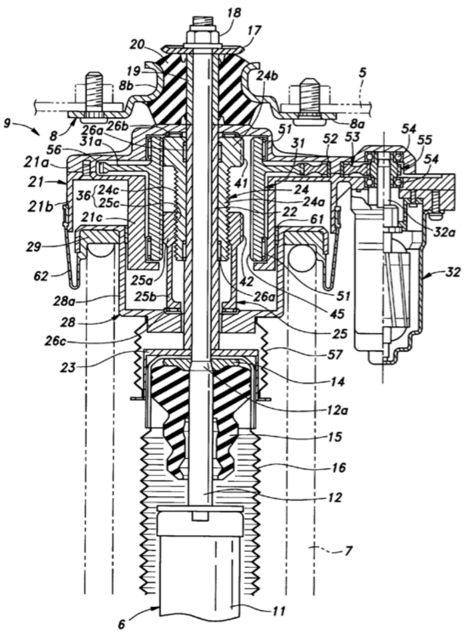

Figure 2.26. Front wheel suspension system to which the 4th embodiment is applied

The main disadvantages of the electromechanical height adjustment systems are lack of ability to vary elastic behavior and lower leveling speed. Since the former disadvantage is inherent to all the modern passive suspensions, it can be omitted. 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. In these cases, the leveling speed is not critical, as actuation can be performed in a larger time. The following particularities of these systems can be attributed to their advantages:

- Higher reliability and robustness (when failure of the system occurs, the system behaves much like traditional suspension system)
- Lower initial cost of the system
- No special experience needed for the service
- Overall compact size of the components
- Possibility of having modular design (the system can be retrofit to the traditional suspension)

Therefore, it is reasonable to use electromechanical height adjustment systems when a trade-off between high reliability, low cost, compact size and fast actuation is required.[6]

Chapter 3

Design of Eccentric Height-adjustable System

As the advantages of electromechanical are realized according to the previous chapters, an electromechanical eccentric height adjustable system has been designed and built using the mechanical components such as threaded screws, gear trains, which offer high reliability and reasonable responsiveness and commercial electric motors for automotive applications which could give an acceptable balance between cost and benefits.

3.1 Working principle

An electric motor 4b coupled with mechanical speed reducer 3a, 3b and 3c drives the nut 5b (Fig.3.1). Power is supplied by on board electric battery 4a. The speed reducer can be of different kind like planetary or parallel axis gear trains, belt and pulleys or chain and sprockets. Even if in all the cases the functionality does not change, reliability, weight, cost and transmission efficiency can be driving factors in choosing one instead of the other. In the arrangement shown in Fig.3.1, the speed reducer is composed of a planetary gear head 3b and a parallel axis gear stage with pinion 3a and gear 3c. Rotational motion is then transformed into linear by means of screw 5a and nut 5b that is part of gear 3c. The screw 5a is internally hollow and mounted on the shock absorber tube 2b. The compound bearing 6 (axial thrust and radial bearings) decouples the axial motion from the rotations of the nut (3c and 5b). In case of extreme deflection of the spring 2a, the upper part of the shock absorber tube 2b comes into contact with rubber bump stop 8 to avoid failure of the former.[6]

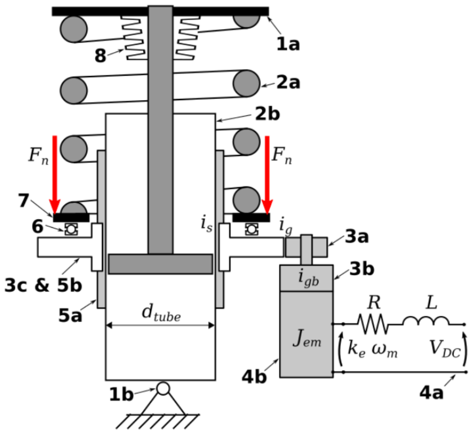

Figure 3.1. Schematics of the height adjustment system with lower spring seat actuation[6]

3.2 Description of the system

Based on the working principle and the requirements, an eccentric height-adjustment system has been designed and modeled with CAD.

In the drawing Fig.3.2 of the cross section, 1 is a spring holder used to hold the lower end fo the coil spring,2 is a screw dust boot which is a plastic cover to prevent dust goes on the surface of screw threads, 3 is the upper arm which is the housing for the nut and used to transmit force from the nut to sliding elements, 4 is a conical washer which allowing the alignment of the nut and guiding element,5 is used to allow mounting of the conical washer, 6 is nut, 7 is power screw Tr 12 x 3, 8 is an O ring, 9 is bronze bushing, 10 is a screw adapter used to fix the screw on the housing, 11 is axial needle bearing, 12 is lower arm which is welded on the shock absorber tube, 13 are retaining rings used to lock axially movement of bushings 14, 15 is guiding element used to guide the spring holder on the shock absorber tube, 16 is the shock absorber.

An assembled 3D CAD view of such an eccentric height adjustment suspension protocol together with the 3D section view is shown in Fig.3.3.

A detail disassembled 3D CAD view of the height adjustment part is shown in the Fig.3.4, where the components are indicated together with the components, the upper part of the suspension (spring, shock absorber, etc) is not shown in the figure.

According to the design and the CAD drawings, the prototype has been built and assembled as shown in Fig.3.5 a), while the normal position and the lifted position of the suspension are reported in Fig.3.5 b) and Fig.3.5 c) respectively.

Disassembled view of key components of the prototype is shown in Fig.3.6,which

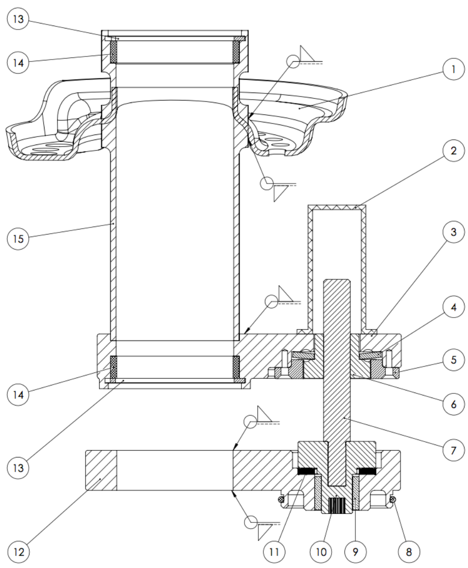

Figure 3.2. Section view of the actuator assembly

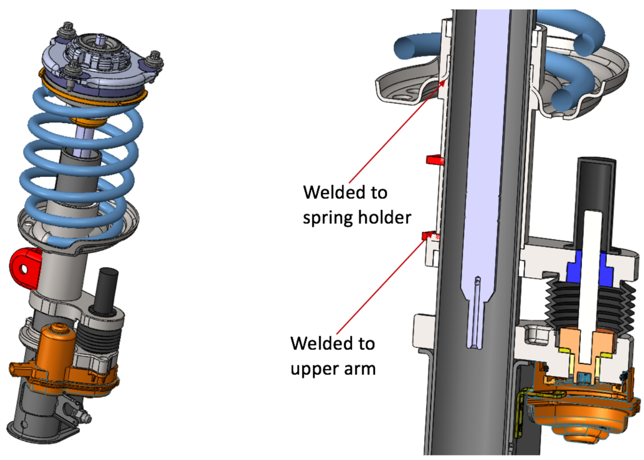

Figure 3.3. System assembled on suspension

in detail, a) is the TRW actuator. b) is the sliding element integrated with spring holder and upper arm which corresponding to 1, 3 and 15 in the drawing Fig.3.2, c)is the bronze bushing which corresponding the 9 in the drawing Fig.3.2, d) shows the practical selected power screw, bronze nut and axial bearing, e) shows the lower

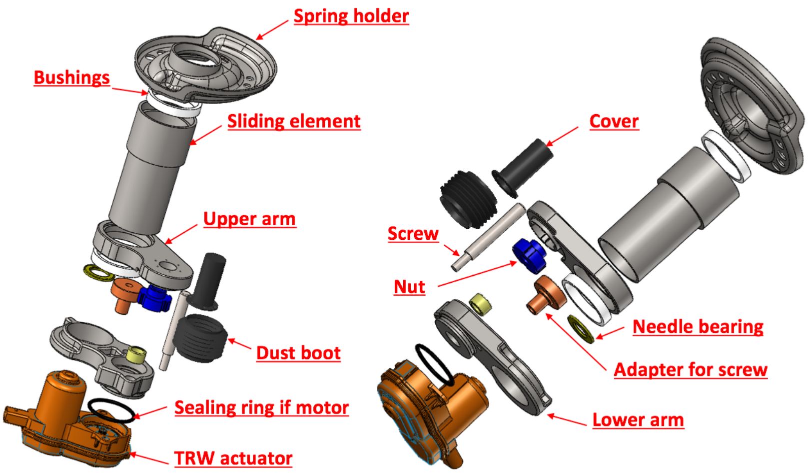

Figure 3.4. Explored view of the suspension

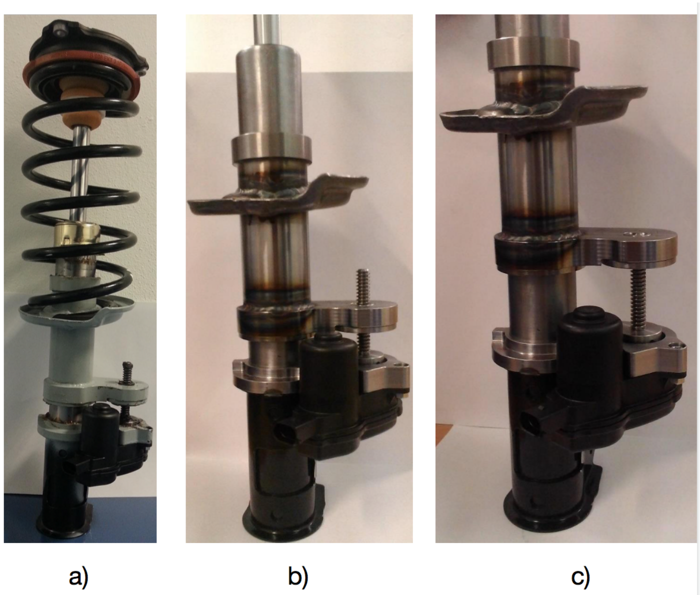

Figure 3.5. Assembled view of real prototype: a) Overview of the assembled prototype. b) Normal position of the suspension. c) Lifted position of the suspension.

arm welded to the suspension tube as a whole which is consistent to the 12 of Fig.3.2 represented.

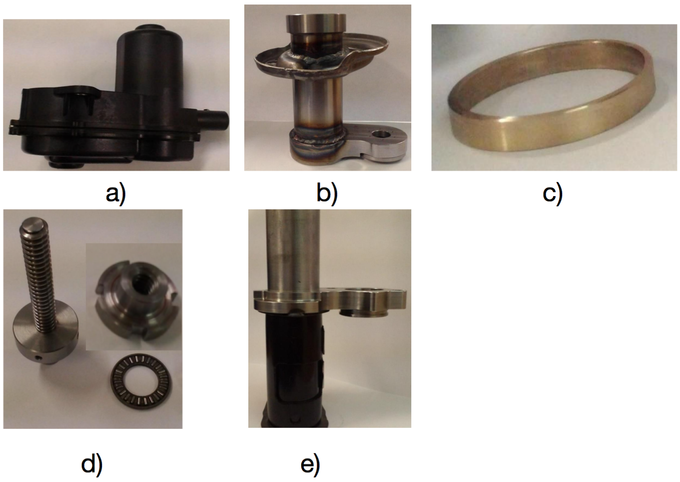

Figure 3.6. Components view: a) TRW actuator. b) Integrated spring holder, upper arm and sliding element. c) Bronze bushing. d) Power screw, nut and axial bearing. e)Lower arm.

The coil spring that mounted on the system has a stiffness equal to 23.8 [N/mm] and a free length of 415 [mm], as the spring manufacturer's specification indicated. The spring rebound length after assembled on the shock absorber is 288 [mm].From Hooke's law

$$F = Kx$$

, it can be calculated that the payload

$$F = 23.8 * (415 - 288) = 3020[N]$$

The spring elastic characteristic curve is shown in Fig.3.7. The TRW motor as indicated on the specification has a dimension: Ø38.1×57.0 mm, shaft diameter: Ø3.174 mm as shown in Fig.3.8 and an input voltage of 12 V Direct Current, the other characteristic specification as well as the performance curve of the TRW motor are reported in Table3.1 and Fig.3.9, correspondingly. Experiments are performed based on the precondition that the screw-nut transmission coupling has a friction coefficient $f = 0.3$, the friction forces on the sliding elements equal to 300N, in order to understand the motor current absorption to lift up such a spring load and when lowering the system. The results have shown that in order to lift up the spring load the motor requires about 8A of current absorption and for lowering it requires around 4A of current absorption as shown in Fig.3.10 and Fig.3.11.

The above mentioned prototype is the very initial version of the height adjustment system, as the experimental test proceeds, some improvements and modifications have been done based on this version of prototype, it will be discussed more

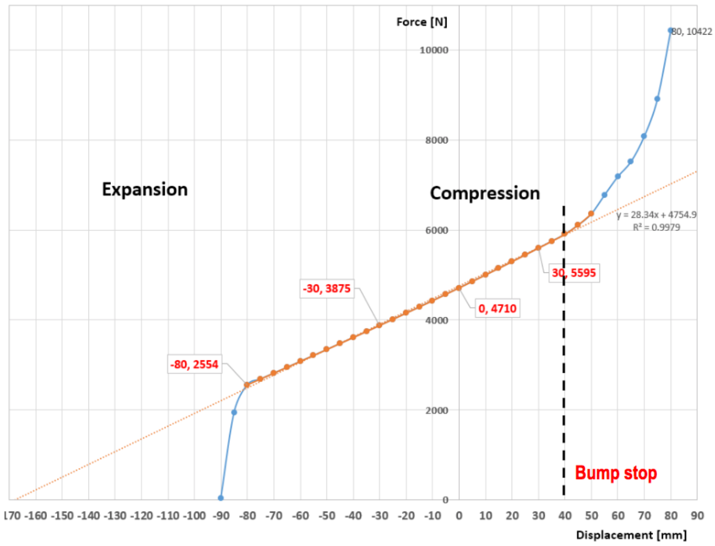

Figure 3.7. Spring elastic characteristic curve

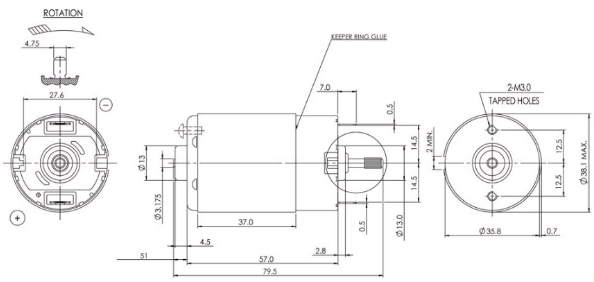

Figure 3.8. Drawing of TRW motor

in detail in following chapters.

Table 3.1. Specification of TRW motor

No Load Speed	10125	rmp
No Load Current	0.75	A
Nominal Speed	8944	rpm
Nominal Torque	37.48	mNm
Nominal Current	4.34	A
Stall Torque	321.36	mNm
Stall Current	32.88	A
Maximun Output Power	85	W
Maximun Efficiency	67	%
Weight	232	g
Operation Temperature	-40 to 85	ºC
Storage Temperature	-40 to 120	ºC

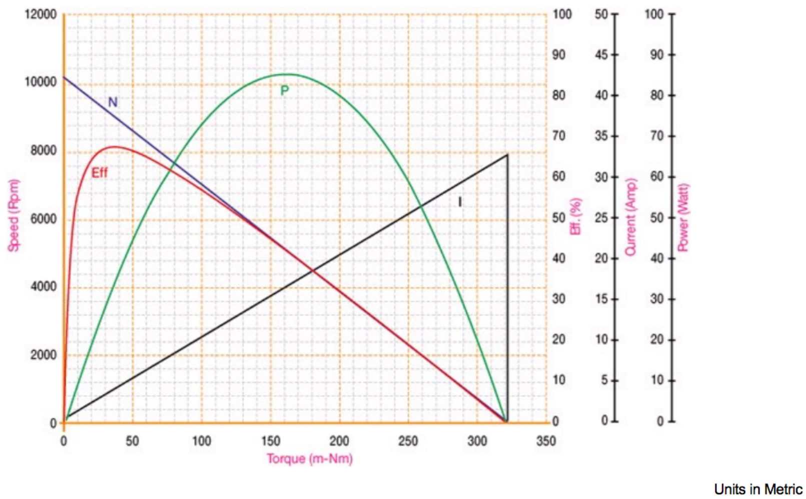

Figure 3.9. Performance curve of TRW motor

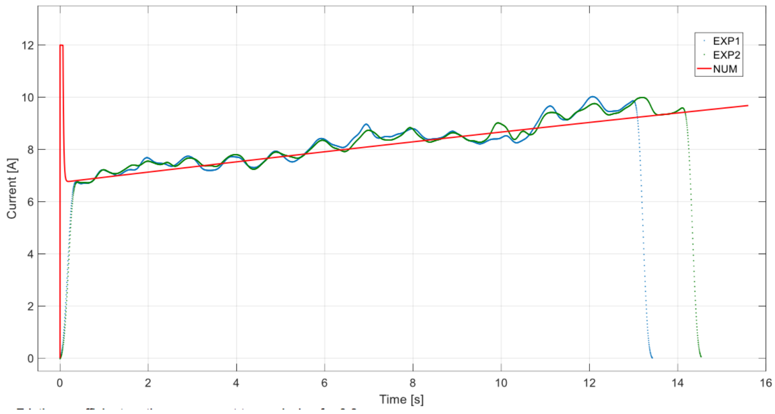

Figure 3.10. Current absorption to raise(Compressing the spring)

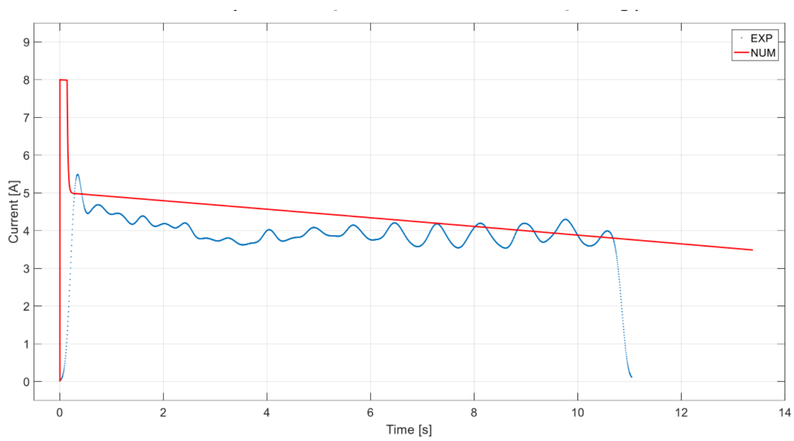

Figure 3.11. Current absorption to lowering(Decompressing the spring)

Chapter 4

Accelerated Experiments for System Components' Life Estimation

In this chapter the accelerated test method will be discussed so as the test rig setup,the failures and improvements and performance comparison.

As aforementioned, the prototype of electro-mechanical height adjustment suspension has been built and therefore, in order to understand the components' life and to determine the critical components, the reliability test has to be performed. Here we used an accelerated experimental method to accelerate the time required to reach the target 10,000 cycles or to reach the first component failure of the system or better, the fatigue life estimation. With such a method, instead of testing the entire lifting and lowering stroke, a reduced travel distance of +5mm and -5mm respected to nominal position is considered. 5KN of force is added to the system in order to simulate the real sprung mass under the assumption of 1/4 of 2000 kg vehicle mass is assigned to each suspension, with such a load, the spring is compressed an additional distance of 80mm, see also Fig.3.7. The entire test is controlled automatically via the control unit which is functioning with pre-loaded program. The screw subjects variable dynamic load form lifting and lowering so the wear of the screw is not easy to be evaluated, while the life expectancy of the screw and nut can be given by the formula:

$$Life = \left(\frac{C_\partial}{F_m}\right)^3 10^6$$

where the Cα is dynamic load rating [N], Fm is the average axle load [N], the Life is in the unit of [revolution].Therefore the number of moves will be given by :

$$N_{moves} = \frac{Life}{L_{stroke}/p}$$

where L_{stroke} is the stroke distance and p is the pitch. The screw wear calculation method is by using 2 pins each putting in a side of the same screw pitch, measure the distance of the 2 pins before the test and after a certain cycles of experiments. This calculation method is based on the assumption of uniform wear and one side wear due to major stresses are subjected when lifting the spring as shown in Fig.4.1. As shown in Fig.4.2, using a micrometer to clamp the 2 pins together with the screw and to measure the distance M1 and M2 which is distance before test and after test receptively (at the temperature), $\theta = 30°$ as the screw specification indicated.In Fig.4.3, O_1 and O_2 are the centers of pins, CE is the wear.

$$O_2A = \frac{M_1 - M_2}{2}; O_1O_2 = \frac{O_2A}{cos(\theta/2)} = \frac{M_1 - M_2}{2cos(\theta/2)}$$

;

$$CD = O_1O_2; CE = CDsin(30°) = \frac{O_1O_2}{2} = \frac{M_1 - M_2}{4cos(\theta/2)} = \frac{M_1 - M_2}{3.864}$$

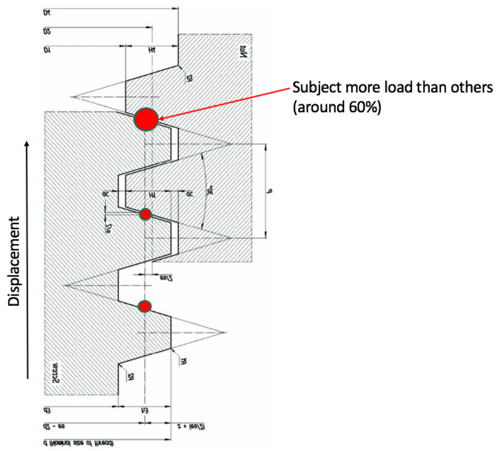

Figure 4.1. Schematic view of major stress distribution

4.1 Test rig set up

The test is performed at laboratory of LIM in Verres, with the test rig supported by Magneti Marelli (Fig.4.4),such a test rig is composed of an air spring cylinder with diameter of 200mm, a pneumatic supply inlet port, a force transducer and a mechanical safety valve. In order to simulate 5KN of sprung mass load, a pressure of 1.6 bar has to be supplied into the air spring as

$$P = \frac{F}{A} = \frac{5000}{\frac{\pi}{4} * (200 * 0.001)^2} \approx 1.6[bar]$$

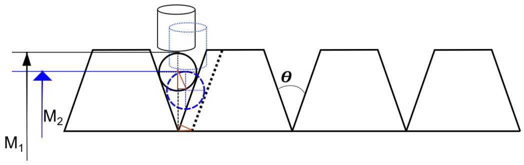

Figure 4.2. 2 pins method of screw wear estimation measuring

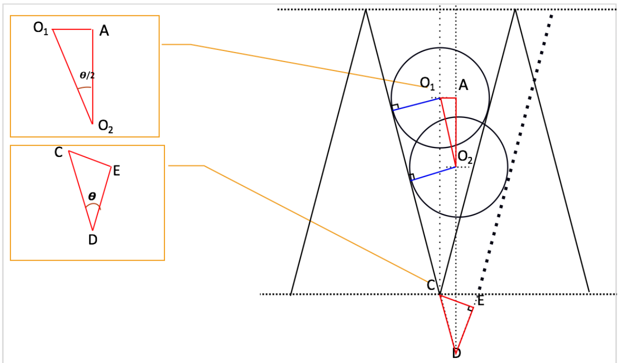

Figure 4.3. 2 pins method of screw wear estimation calculation

The 1.6 bar pressure is suppled by a external compressor, via pneumatic pipes,valves and pressure sensor into the air spring cylinder. The pressure supply system equipped 2 solenoid valves with ON/OFF control by the control unit based on the information coming from the pressure sensor as shown on the right of Fig.4.6 in order to maintain 1.6 bar of pressure constantly. The travel distance is controlled by the control unit based in the angular position coming from an angular position sensor (Left of Fig.4.6)which is fixed on the test rig frame (unmove) and through a rigid linkage to the upper arm strut (move) as shown in Fig.4.7. Arduino Mega has been selected to be the control unit due to its simplicity and cost effectiveness, together with the power board (Fig.4.8) manage the control algorithm and actuation of the motor. During the test the temperature of the system increases fast and therefore the system is set that 1 minute of work is followed by 2 minutes of rest and a cooling fan cooperated.

The complete layout of the testing circuit is reported in Fig.4.5 where the temperature sensor is introduced after the first failure of the TRW motor, it will be discussed in next section "failure and improvement.

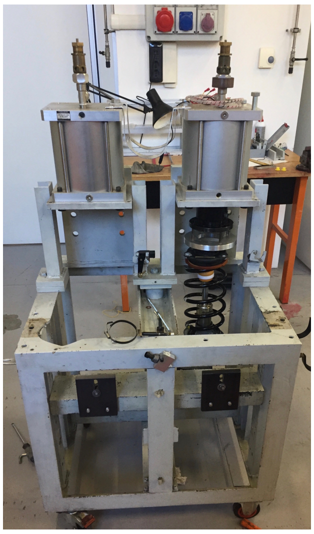

Figure 4.4. Test rig for the accelerated experiments

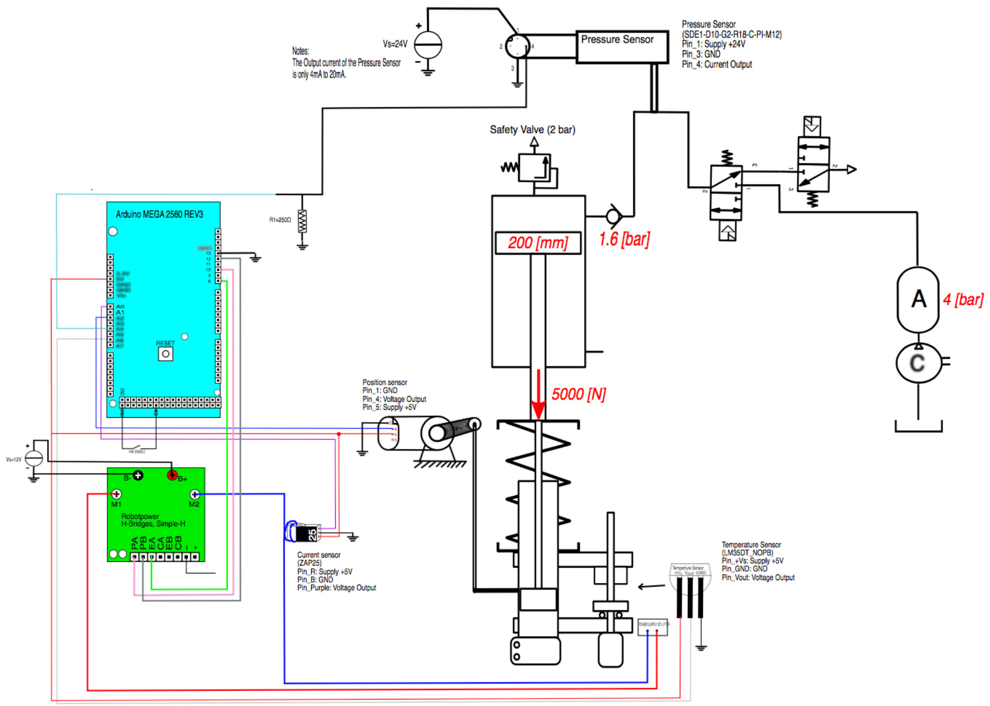

Figure 4.5. Layout of the testing circuit

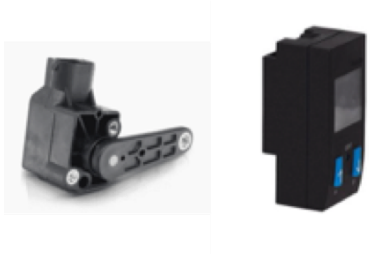

Figure 4.6. Angular position sensor (left) and Pressure sensor (right)

Figure 4.7. Angular position sensor installation

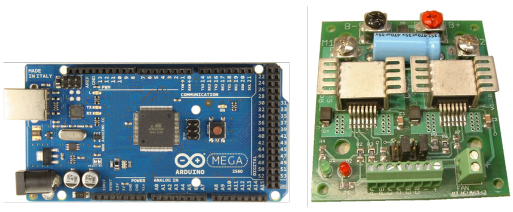

Figure 4.8. Arduino control unit (left) and power board (right)

4.2 Failure and improvement

During the experiments, some failures and improvements has been done. At the first some hundreds cycles the motor was burned (Fig.4.9) ,this is due to high temperature during its working and not effective definition of cooling-working cycle. As the countermeasure ,a temperature sensor was added (Fig.4.10), the installation of the temperature sensor is on the motor inner surface by cutting a portion of the motor cover so that as a window for the temperature sensor. The control algorithm is then no longer as 1 minute work followed by 2 minutes rest but based on the signal coming from the temperature sensor, whenever the temperature of the motor is higher than 55 degree, the system is set to rest and restart to work when the temperature of motor drops back to 30 degree. With this improvement, the control is more reliable and complete since there are more control signals and the motor's life can be significantly elongated.

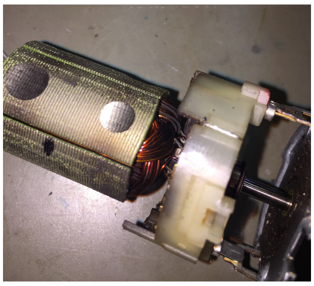

Figure 4.9. The burned motor due to over-temperature

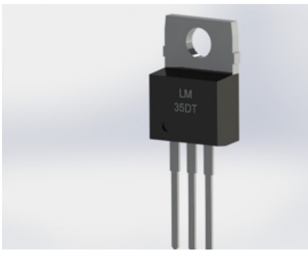

Figure 4.10. The temperature sensor

After a certain number of cycles, the screw stripping was found on the system as shown in Fig.4.11, this is reflected on the computer monitor screen as increased current absorption due to its increased friction. We analyzed the problem and get the conclusion of that the screw nut coupling can not afford such a high axial load, especially when lifting up the system. So we decided to change the original screw and nut to the one original equipped with the TRW motor, but since the shape of the screw and nut are not the same as the one used on our system, so we modified the TRW screw and nut by cutting our original nut then press and fit the TRW nut into it, so as for the TRW screw inserted into the screw adapter, the final result is shown in Fig.4.12

Figure 4.11. The screw stripping

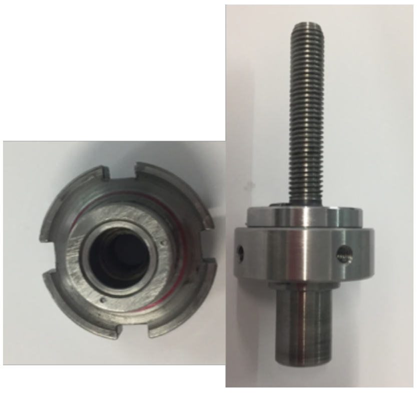

Figure 4.12. The modified TRW screw and nut

Deformation was found at the lower arm surface under the axial bearing (11 in Fig.3.2)caused by extreme high hardness of the axial bearing needles which deform the lower arm. This phenomenon losses the effectiveness of the axial bearing with negative effect of an increased friction, and therefore the current absorption. Corrective action was done by inserting an plate washer in between the axial bearing and the lower arm, as shown in Fig.4.13

Figure 4.13. Plate washer and axial bearing

Conical washer between the nut and upper arm (4 in Fig.3.2) was substituted to a rubber cushion because the conical washer didn't work well as the conical shape only contact limited area of the surfaces,but the rubber cushion can't sustain such high lifting load, so it deformed taking the shape of the nut, as shown in Fig.4.14

Figure 4.14. Deformed rubber cushion

Two disks was added, so that the rubber cushion lays in between the two disk, shown in Fig.4.15.

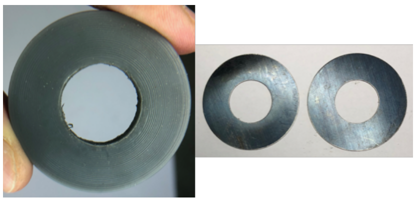

Figure 4.15. Rubber cushion and the two disks used protect it

After some hundred cycles, we have observed that the upper arm was moving while performing the test, so we assumed that this movements may cause several negative effects of the system, including the screw-nut coupling, bearing, and the current absorption. Therefore, an re-engineering was done and the profiles of the upper and lower arms were changed with an anti-rotation guide added. As shown in Fig.4.16 and Fig.4.17, the shape profiles have been changed, an anti-rotation rod is integrated with the lower arm and a hole on the upper arm used to guide the movement, in practice, a plastic tube was intruded into the upper arm bore matching the diameter of the the anti-rotation rod and the anti-rotation rod is lubricated.

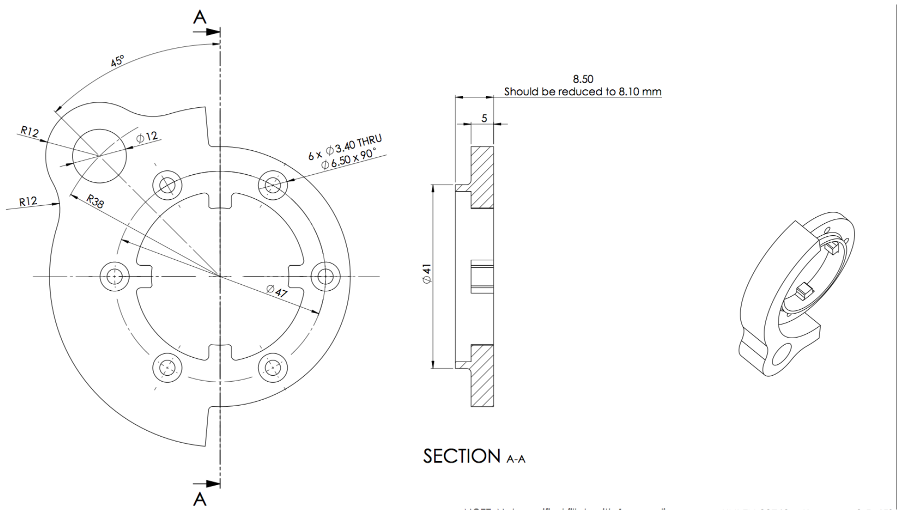

Figure 4.16. Modified upper arm profile

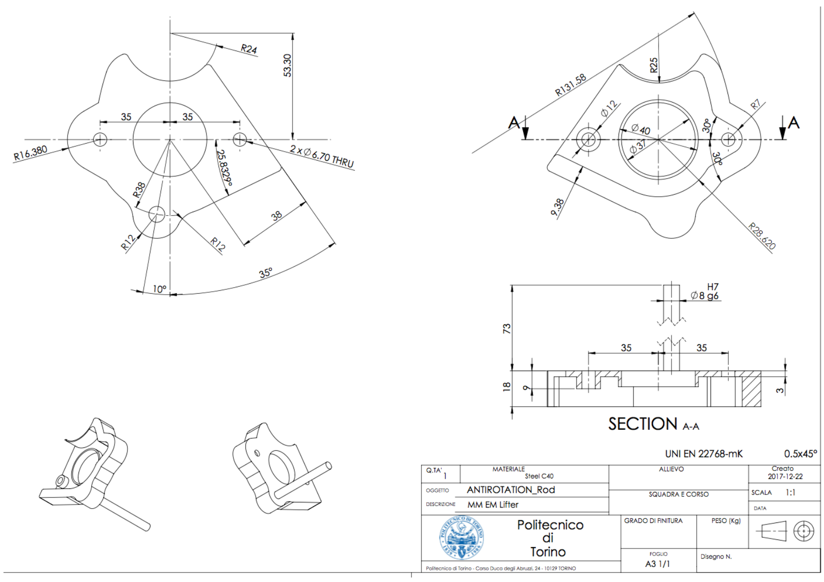

Figure 4.17. Modified lower arm profile with anti-rotation rod

After the new upper and lower arm have been produced, the test was restarted, but good time doesn't last long, after around 2500 cycles since restart, thread stripping was found again on the screw, as shown in Fig.4.18, since the screw and nut are original equipment of TRW and thread stripping is still happened, we presumed that this failure is due to the high hardness of the nut, therefore the improvement action is substituting the TRW steel nut to bronze nut, as shown in Fig.4.19, it can also be seen that the new upper and lower arm profiles, the anti-rotation rod, the cut window on the motor for installation of temperature sensor as mention in Fig.4.10

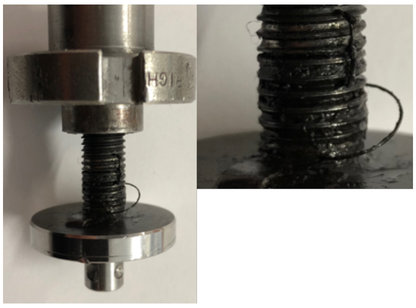

Figure 4.18. Thread stripping on TRW screw

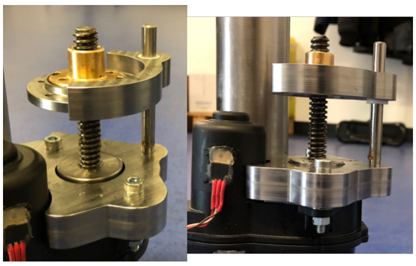

Figure 4.19. Bronze nut installed on the modified system

The test was then restarted again, about 5000 cycles after, there was a big noise during its lifting and lowering process, so we had to disassemble the suspension again, by detailed analyzing, the problem is again on the screw-nut coupling, but this time is on the bronze nut. we found thread deformation in the bronze nut, as shown in Fig.4.20. Because of this deformation the pitch of the nut became larger than the screw, in another word, the screw threads can move inside the nut groove, therefore when high axial load was applied, noise was created.

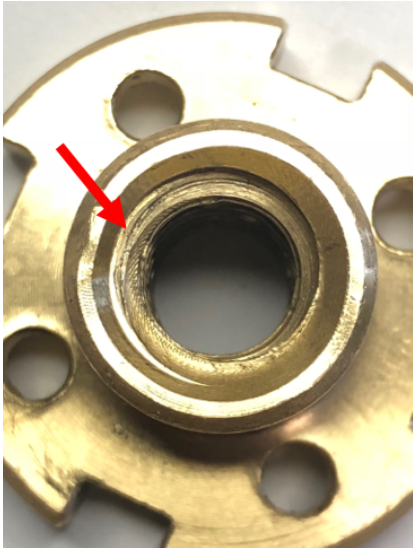

Figure 4.20. Bronze nut thread deformation

Problems on the screw and nut over and over again alarmed us that the Tr 12x3 screw and nut are not strong enough for the lifter. The afterward improvement will be discussed in the chapter of Conclusion and Suggestions for Possible Improvements of the System.

4.3 Performance comparison

The experimental results have been analyzed in different aspects, including load and unload condition, motor temperature, number of cycles, and when it closes to failure.

As shown in Fig.4.21, the current absorption of spring load only is about 8A and of 5KN added full load is about 10A. Recall the specification of the TRW motor Table.3.1 where indicates that the nominal current of TRW motor is 4.34A, therefore in any case of the load conditions, the motor is overloaded.It is the root cause of motor failure.

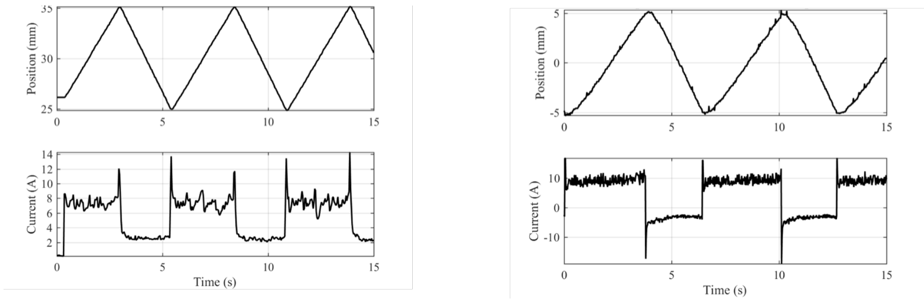

Figure 4.21. Performance comparison between spring load only on the left and full load (spring load + 5KN load) on the right

Then comparison of the current absorption in different temperature is shown in Fig.4.22. It can be observed that with increasing temperature, the current absorption increases, as the result of internal resistance increases with temperature, therefore, the motor is more overloaded.

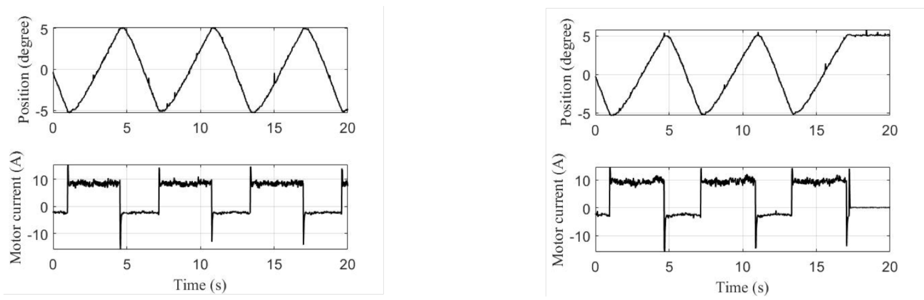

Figure 4.22. Performance comparison of different motor temperature, 17°C on the left and 53°C on the right

Current absorption is a good indicator of system's state of healthy, when current absorption increases, meaning that the resistance is increased, so means that there is something not functioning well. Fig.4.23 shows the current absorption behavior of the TRW screw and nut version at zero cycle, it can be observed that the graphical current absorption behavior is mainly below the line of 10A means that the current absorption is around 9A. Then at 1000 cycles as shown in Fig.4.24,the current absorption is increased to the level around 9.5A where the graphical behavior of current absorption emerged from line of 10A. At 2000 cycles, as shown in Fig.4.25, the current absorption is further augmented so that arrived to the level of 10A as the line of 10A is in the middle of current absorption curve from the graphical point of view.

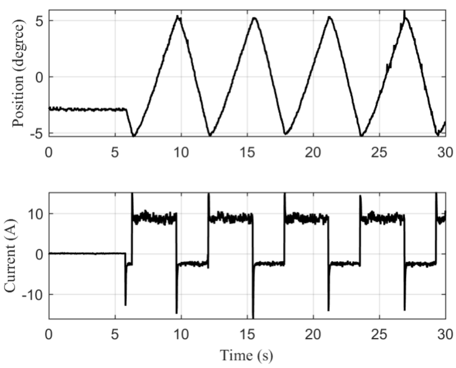

Figure 4.23. Performance of TRW screw and nut version at zero cycle

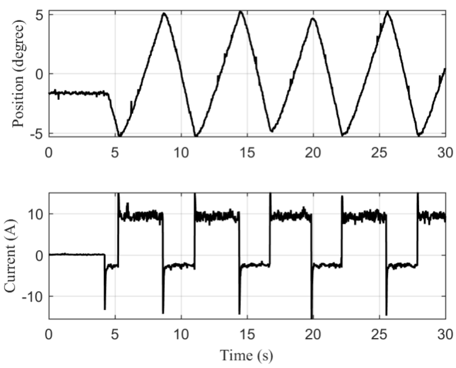

Figure 4.24. Performance of TRW screw and nut version at 1000 cycle

As aforementioned the in Fig.4.18, the thread stripping on the screw significantly increased the resistance therefore current absorption, from the Fig.4.26 it can be seen that the current absorption is continuously increasing and dramatically increased at the last point where the fault occurs.

A comparison of the performance curve between normal condition and when closed to the failre is shown in Fig.4.27 where on the right side of the figure shows the normal condition of the system and on the right side shows the condition when the screw stripping occurred. It can be observed that when the screw stripping occurs, the current absorption increases significantly until fault.

Therefore monitoring the current absorption is a good way to detect problems which is very necessary for the future failure preventive action.

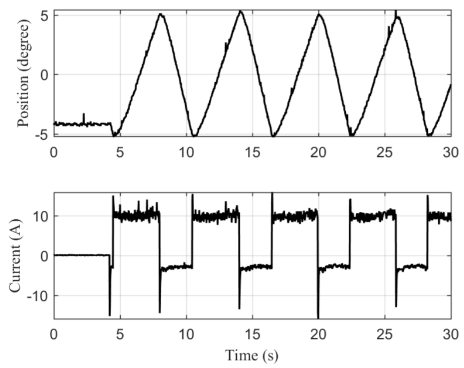

Figure 4.25. Performance of TRW screw and nut version at 2000 cycle

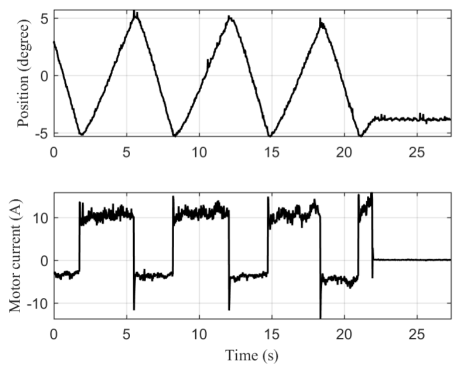

Figure 4.26. Performance of TRW screw and nut version at moment of fault

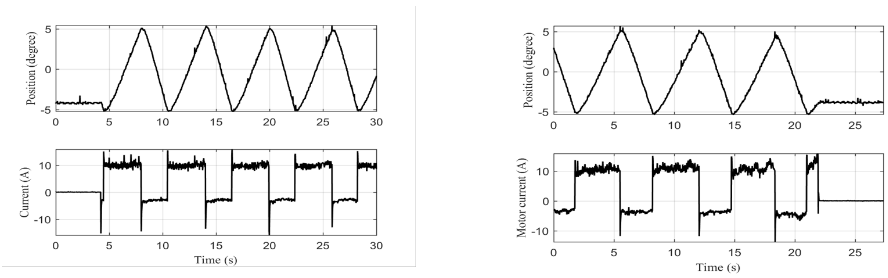

Figure 4.27. Performance comparison normal operation condition (left) and when it close to failure(right)

Chapter 5

Conclusion and Suggestions for Possible Improvements of the System

Under the tough pressure of regulation set by European Commission which requiring an emission of carbon dioxide less than 95 gCO2/km by the end of 2020[1], the importance of height adjustment systems become even more evident since it can offer a valuable benefit in reduction of fuel consumptions and emissions. The possibility of adjusting vehicle hight can save up to about 4% of carbon dioxide emission[6].

In this thesis, a literature review of different height adjustment solutions has been comprehensibly described and by doing so, the benefit of electromechanical type of height adjustment system is noticed. The attention is focused on the electromechanical version due to its coexistence of functional and economic aspects which giving high reliability, robustness, compact size together with low manufacturing cost.

The design of height adjustment system has been discussed and the prototypes have been manufactured according to the design, the reliability test has been performed and the results have been studied.

Test rig was set up according to the height adjustment system and the acceleration experiments for system components' life have been proceeded.

The reliability test is not finished yet since the target number of cycle still not reached but nevertheless many valuable informations are obtained, based on that the system has been improved.

Failure Mode and Effects Analysis (FMEA) is a design tool used to systematically analyze postulated component failures and identify the resultant effects on system operations. Process Failure Mode and Effects Analysis (PFMEA) is a methodical approach used for identifying risks on process changes. The Process FMEA initially identifies process functions, failure modes, their effects on the process. The severity ranking or danger of the effect is determined for each effect of failure. Then, causes and their mechanisms of the failure mode are identified. The assumption that the design is adequate keeps the focus on the process. A high probability of a cause drives actions to prevent or reduce the impact of the cause on the failure mode. The detection ranking determines the ability of specific tests to confirm the failure mode / causes are eliminated. The PFMEA also tracks improvements through Priority Risk Index (PRI) reductions. By comparing the before and after PRI, a history of improvement and risk mitigation can be chronicled.

A Process Failure Mode and Effects Analysis (PFMEA) of the electromechanical height adjustment system prototype has been studied and shown in Fig.5.1, the critical components are TRW motor, power screw, nut, axial bearing and rubber cushion, its class is followed right after where“c” stands for critical, “+” means important and “-” represent secondary(normal). It can be seen that the TRW motor before installing temperature sensor and adapted the duty cycle control based on it has a PRI of 180 which means high risk of failure (moderate probability of occurrence 5 points, significant functional gravity 6 points and remote level of detectability 6 points). After implemented the temperature sensor and the control algorithm accordingly, the PRI reduced to 72 which represent average risk of failure.Since the TRW motor is always overloaded, the PRI can not drop below 50 where is the boundary of low risk. On the other hand, the axial bearing and rubber cushion before having a PRI of 70 and 60 correspondingly where are in the range of average risk and after the corrective actions, the PRI become 30 and 20 accordingly where it can be seen as risk free since in the low risk range.

Power screw and nut are difficult to reduce the PRI as well, even with continuously monitoring of the current absorption but the essential problem of the screw-nut coupling is not solved.

To reach the target 10,000 cycles of life, the current power screw Tr12x3 is not enough powerful for the system, in the future improvement it can be improved to Tr16x3.

A further step is to install 4 electromechanical lifter actuators equipped with end-stroke switches and stroke sensor on the FIAT 500X, to achieve the control of both globally (4 corners) or independently for each axle (front or rear corners) by the user via user interface based on Android system. Two independent Axle Control Units (ACU) controlled through Bluetooth connectivity will be available on the car and the system will fed by the 12-V vehicle battery with fuse-protected.

PROCESS F.M.E.A.

PROCESS FMEA	MM LIFTER									SERIAL N° - DRAWING N°					PAGE
PROCESS PHASE		SUPERVISOR	PLANT		PLANT / SUPPLIER					DATA	REVISED DATE				
			ON PRODUCT		PRESENT STATUS					CORRECTIVE MEASURES		IMPROVED STATUS			
AIM OF PHASE	CLASS	FAULT METHOD	EFFECTS OF FAULT METHOD	CAUSE OF FAULT METHOD	PLANNED CONTROL MEASURES	▲ PROBABILITY OF DEFECT	* GRAVITY OF DEFECT	□ DETECTABILITY OF DEFECT	● PRIORITY RISK INDEX	RECOMMENDED IMPROVEMENTS	ACTION TAKEN	▲ PROBABILITY OF DEFECT	* GRAVITY OF DEFECT	□ DETECTABILITY OF DEFECT	● PRIORITY RISK INDEX
TRW MOTOR	+	OVERHEATING	MOTOR BURNED	SYSTEM STOP AND SAFETY CONCERN	1 MINUTE WORIKING 2 MUNUTS REST	5	6	6	180	IMPLEMTENT TEMPERATURE SENSOR	DUTY CYCLE BASED ON INFO. FROM TEMPERATRUE SENSOR	2	6	6	72
POWER SCREW	C	THREAD STRIPPING	SCREW BREAK	OVERCURRENT, SYSTEM STOP AND SAFETY CONCERN	PERIODICAL INSPECTION	6	6	6	216	CONTINUOUSLY MONITOR CURRENT ON COMPUTER	COMPUTER MONITORING AND AWARE OF NOISE	5	6	2	60
NUT	C	THREAD DEFORMATION	SLIPPAGE BETWEEN SCREW AND NUT	OVERCURRENT, SYSTEM STOP AND SAFETY CONCERN	NONE	5	6	7	210	CONTINUOUSLY MONITOR CURRENT ON COMPUTER	COMPUTER MONITORING AND AWARE OF NOISE	5	6	2	60
AXIAL BEARING	–	INCREASE ROLLING RESISTANCE	NOT FUCTIONING IF AXIAL BEARING	HIGHER CURRENT ABOSRPTION	OCCASIONAL INSPECTION	5	2	7	70	ADD PLATE WASHER	PLATE WASHER ADDED AND PERIODICAL DISASSEMBLE INSPECTION	2	3	5	30
RUBBER CUSHION	–	DEFORK AND ESCAPE FORM DISKS	REDUCE EFFECTIVENESS	REDUCE DAMPING BETWEEN NUT AND ARM	OCCASIONAL INSPECTION	5	2	6	60	ADD 2 DISKS	RUBBER CUSION LAYS IN BETWEEN 2 DISKS AND PERIODICAL INSPECTION	2	2	5	20

▲ PROBABILITY OF OCCURRENCE:		* GRAVITY (EFFECTS ON CUSTOMER)			□ DETECTABILITY		● PRIORITY RISK INDEX		PARTICIPANTS	ORGANIZATION	SIGNATURES
EVALUATION	CORRESPONDING POINTS	EVALUATION	CORRESPONDING POINTS	EQUVALENT DEMERITS	EVALUATION	CORRESPONDING POINTS	EVALUATION	RESULTING POINTS (PROB* GRAV* DETECT)			
- REMOTE	= 1	- SIGNIFICANT	= 1	1	- HIGH	= 1	- LOW	= 1 TO 50			
- LOW	= 2 TO 3	- SIGNIFICANT (FUNCTIONALLY)	= 2 TO 6	10	- MODERATE	= 2 TO 3	- AVERAGE	= 50 TO 100			
- MODERATE	= 4 TO 6	- SERIOUS (FUNCTIONALLY AESTHETIC)	= 7 TO 8	20 TO 40	- REMOTE	= 6 TO 8	- HIGH	= 100 TO 200			
- HIGH	= 7 TO 8	- VERY SERIOUS (FUNCTIONALLY)	= 9 TO 10	100	- VERY REMOTE	= 9	- VERY HIGH	= 200 TO 1000			
- VERY HIGH	= 9 TO 10				- IMPROBABLE	= 10					

Figure 5.1. Process Failure Mode and Effects Analysis of height adjustment system

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