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INNOVATIVE RUNNING GEAR SOLUTIONS FOR NEW DEPENDABLE, SUSTAINABLE, INTELLIGENT AND COMFORTABLE RAIL VEHICLES

Deliverable 3.1 – State of the art actuator technology

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EXECUTIVE SUMMARY

The target of the work package T3.1 is to analyse the technical solutions for active suspension and control actuator technology, which are available in the market of railway and non-railway applications. Based on this information new concepts should be generated in the work package T3.2 and an authorisation process of the vehicle with active suspension should be proposed to reach a manageable homologation process in work package T3.3.

This document is split in several sections which have the task to introduce the main topic, which is the requirement specification of actuators for active suspension systems.
The first section describes the design principle (primary and secondary suspension) and the dynamic characteristics of railway vehicles. The main items are vibration comfort, curving behaviour and stability. This information shows the necessity of suspensions systems in railway applications.

The second section gives an overview of the technical solutions which were developed and used in the past and today. The different technical solutions are shortly described. The main technical data and the pros and cons were listed to prepare a technical basis for the validation of the use in railway suspension systems.

The following two sections give a more detailed documentation of the suspension actuators and the system design, which were used in railway (section 3) and non-railway applications (section 4). Due to the use of these solutions in mass production the layout is industrialized for its application.

The last section (no. 5) is the basis of the work package T3.2. The technical requirements like force, relative speed, power, frequency range, stroke and stiffness of the actuators are defined for the different suspension systems, which would control the different movements of the bogie and the carbody.

ABBREVIATIONS AND ACRONYMS

MRF	Magneto-Rheological Fluid
ERF	Electro-Rheological Fluid
IRW	Independently-Rotating Wheel (or Wheelset)
DLR	German Aerospace Center
NGT	New Generation Train
TPMA	Tubular Permanent-Magnet Actuator
MR	Magneto-Rheological
ER	Electro-Rheological
DC	Direct Current
EHA	Electro-Hydraulic Actuation
EMA	Electro-Mechanical actuation

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1. INTRODUCTION

The target of the work package WP3 is to analyse the technical solutions for active suspension and control technology on the market in railway and non-railway applications. Based on this information new concepts should be generated and the authorisation process of vehicles with active suspension should be proposed to reach a manageable homologation process.

The work package T3.1 will cover the first item. In this task, the performances of different types of existing actuator technologies are reviewed. As far as possible costs for the different types in relation to their performance shall be also listed.

Further, interesting actuator technologies used in non-railway applications, especially in the automotive industry and aircraft industry, shall be investigated with their regard to the applicability in the rail vehicles.

The technical solutions, which were realized and researched in the last years, were collected and validated.

2. OVERVIEW OF RAILWAY SUSPENSION SYSTEM PRINCIPLES

2.1 VEHICLES DYNAMIC CHARACTERISTICS OF RAILWAY

Modern railway vehicles are mounted on bogies, notwithstanding that this design is more than a century old. Bogies provide a reasonable solution to the conflicting requirements associated with curve negotiation and dynamic stability of railway vehicles. As more unfavourable aspects, bogies contribute 20% of the total vehicle weight, raise the centre of mass, reduce the usable space, and increase the structure gauge and the height of the vehicle floor with respect to the platform. The generalised use of the bogies, despite the significant drawbacks involved, leads to infer the difficulties to replace them through other passive designs.

Bogies are usually characterised by two levels of suspension. The carbody is much heavier than the bogie frame, and the typical suspension designs consist of primary suspensions that are stiffer than the secondary suspension. As a result of these properties, the vibration modes of the carbody and the bogie are weakly coupled: there are modes where the carbody oscillates and the bogie remains almost at rest, and vice versa. The secondary suspension, between the bogie frame and the carbody, ensures that the rigid body modes of the carbody have their frequencies in a frequency band around 1 Hz. Generally, the vertical, lateral and pitch carbody modes have frequencies in the band of 0,8 – 1Hz, whereas the roll carbody mode has a higher frequency (~1,3 Hz). The secondary suspension filters out higher frequency vibration transmission from the bogie with a cut-off frequency of around 2 Hz (and therefore, the carbody frame must have natural frequencies sufficiently above this limit).

The bogie frame (when oscillating on the primary suspension) has natural frequencies in the range 6-10 Hz, at least two octaves above the rigid carbody modes, and at least two octaves below the natural frequencies of the unsprung masses (close to 50 Hz, in which the wheelset oscillates on the ballast bed, which is called the P2 frequency). The cut-off frequency associated with the primary suspension is approx. 15 Hz.

The primary suspension also provides some flexibility between wheelset and frame in the horizontal plane; the longitudinal stiffness of the primary suspension has to be especially high, since it has to transmit the braking and traction forces, and keep the vehicle stable.

The wheelset is a very lightly damped and stiff structure, the response of which is dominated by its resonances (which are above 80 Hz); on the contrary, the track is much more damped; it is an infinite system and is characterised by propagating waves. As a consequence of these properties, the high frequency vibrations associated with the vehicle-track dynamic interaction remain uncoupled from the vehicle behaviour and mostly only affect the unsprung masses of the vehicle.

2.2 RAILWAY VEHICLE DYNAMICS

The main problems linked to the dynamics of the vehicle/track system are:

a) Vibration comfort.
b) Curving behaviour.
c) Stability (hunting and carbody instability).
d) Vehicle/track interaction.

The first three problems are candidates to be treated through active control, and they are summarised in the following subsections.

2.2.1 Vibration comfort

The track irregularities produce vibrations in the carbody that may cause discomfort and motion sickness to the passengers. The technical regulations about railway vibration comfort are defined in the standard EN 12299 and principally affect the design of the carbody and the secondary suspension. The main role of the secondary suspension is to filter the vibration (reduce the transmissibility) that originates in the track irregularities. However, with passive suspensions there will always be some amplification up to 1.4 times the highest natural frequency of the carbody modes. As mentioned above, good suspension designs should filter frequencies higher than 2 Hz that produce discomfort, and should not amplify very low frequencies (0.5 Hz or less) since such vibration produces motion sickness. Passive suspensions are always a trade-off between damping the resonances properly and avoiding transfer of vibration above the resonance frequency. The higher the damping, the better the attenuation of resonance vibration, but this brings increased transfer of vibrations above the resonance frequency.
Fully-active secondary suspensions (i.e. incorporating controlled actuators) can potentially reduce the transmissibility in the frequency band that characterises the actuator bandwidth. Alternatively, semi-active dampers can avoid the trade-off between low and medium frequency comfort, contributing high damping at low frequency, and reducing damping at high frequency.

2.2.2 Curving behaviour

This area is partly regulated by the standard EN 14363. Conical wheels that are linked together with the axle act like a mechanism that steers the vehicle along the track. If no force is applied to the wheelset, it would adopt a radial position moving laterally outwards in the curve; this displacement is proportional to the track curvature and inversely proportional to the wheel conicity. This position of the wheelset on the track will be referred as the *no-slip position*, also called the *pure rolling line*. The steering mechanism is due to creep forces that are transmitted through the wheel-rail contact. If the wheelset lateral displacement is increased with respect to the no-slip position away from the curve centre, longitudinal creep forces appear and produce yaw torques that oversteer the wheelset. Oversteering yaw angles yield lateral creep forces toward the centre of the curve. If lateral actions act on the free wheelset outwards in the curve (i.e. ‘centrifugal forces’), they will push the wheelset beyond its no-slip position leading to longitudinal forces. In turn, these longitudinal forces will produce a torque that will lead to an oversteering yaw angle. The yaw angle (hundredths of a degree) will produce creep velocities that lead to lateral forces that compensate the external action. This balance through creep forces may not be reached, and consequently the wheelset displaces until the wheel flange contacts the rail. This results in a deficient curving behaviour, which it is more likely to occur when:

- the curve radius is small
- the conicity of the wheel profile is small
- the primary stiffness of the bogie is high
- the bogie wheelbase is large
- the longitudinal secondary suspension stiffness is high
- the saturation force of the yaw dampers is high
- the wheel diameter is large
- the track gauge is large
- the flange clearance is small.

Figure 1a shows the curving behaviour of a bogie with a rigid primary suspension. In such a case, the leading wheelset understeers, and consequently lateral creep forces push the wheelset out of the curve until the flange stops it. This case will produce high contact forces, wear, squeal noise and, in extreme circumstances, derailment. **Error! Reference source not found.** (b) shows the curving behaviour of a bogie with a flexible primary suspension that may allow travel to curve without flange contact and consequently lower contact forces, lower wear and less risk noise.

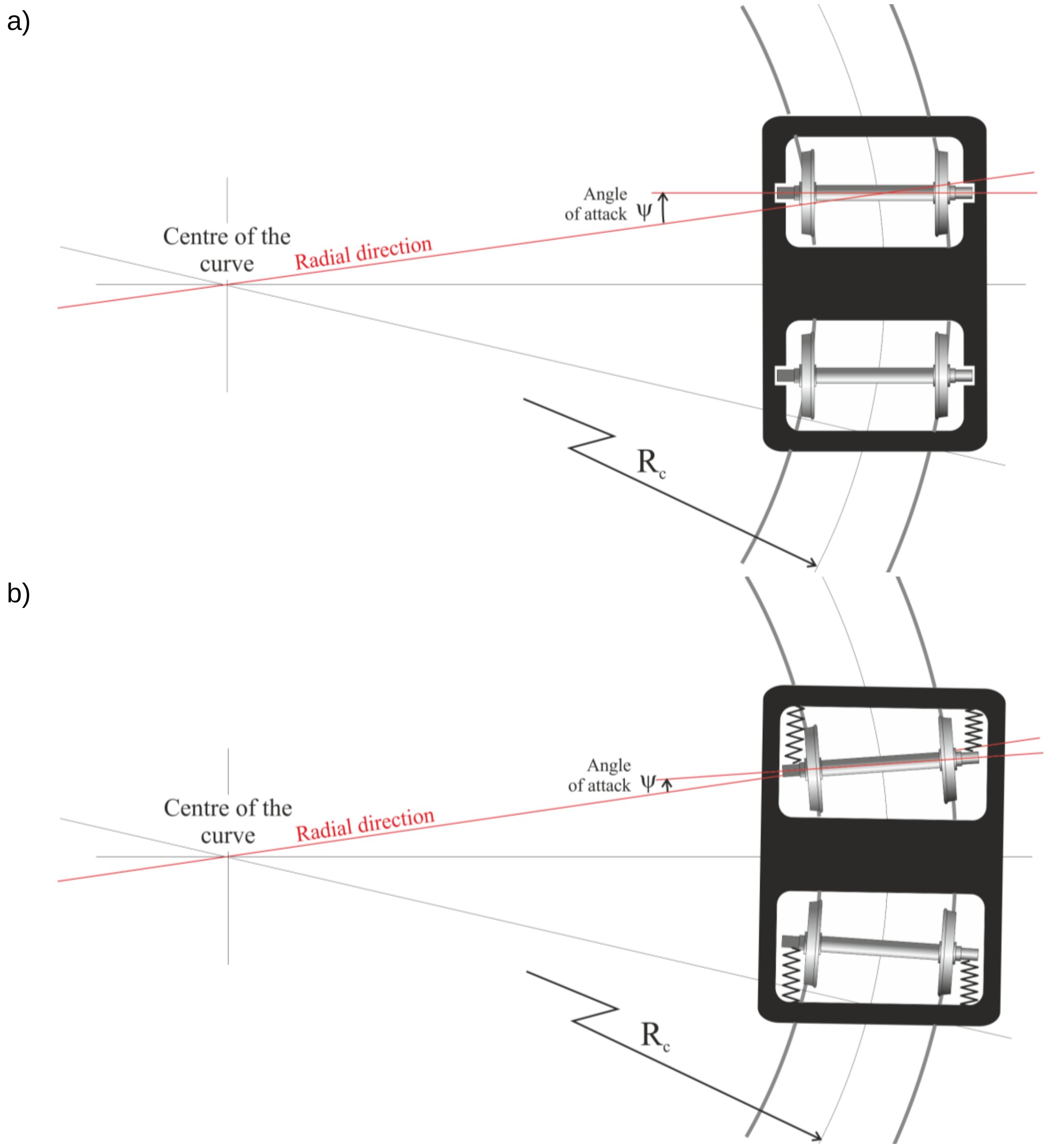

Figure 1: Stiff (a) and flexible (b) bogie negotiating a curve.

Vehicles with independently rotating wheels (IRW) lack longitudinal creep forces and hence the steering mechanism of the standard wheelset. There are examples of tramways that are equipped with IRW, but the ride quality is poor. One satisfactory technology with IRW is the Talgo system, which contains passive steering mechanisms that reduce the angle of attack. The steering forces for the Talgo technology are due to the gravitational stiffness.

IRW is a promising technology for implementation by means of active control since the longitudinal force needed to get the necessary displacements of one wheelset ends is less than 1 kN.

2.2.3 Stability

There are (at least) two different types of dynamic instability in passive railway vehicles: (1) carbody instability that is mainly due to damping deficiency; (2) and hunting. The second type is characteristic of railway vehicles. It is complex problem similar to flutter (in aero elasticity) due to the coupling between the yaw and lateral wheelset displacements through the creep wheel/rail forces. The more frequent hunting response is a harmonic lateral and yaw oscillation of the bogie at frequencies below 5Hz (the secondary suspension uncouples the unstable hunting from the carbody). The hunting frequency has a relation to the vehicle speed, the wheel diameter, the track gauge and the wheel conicity. As with flutter, hunting appears above a certain critical speed, which in some cases can be as low as 100 km/h. The testing and simulation procedures required to prove the properties of a newly designed vehicle associated with hunting are covered by the standard EN 14363.

There is a trade-off between curving behaviour and hunting. Most of the designs that would increase the critical speed worsen the curving behaviour and vice versa. The design features that can contribute to an increase in the critical speed are:

- small wheel conicity
- stiff horizontal primary suspension
- large bogie wheelbase
- stiff longitudinal secondary suspension
- soft lateral secondary suspension
- small mass and yaw moment of inertia of the bogie
- stiff yaw dampers with high saturation force
- large wheel diameter
- small track gauge
- high carbody mass and inertia

3. OVERVIEW OF ACTUATOR PRINCIPLES

3.1 HYDRAULIC ACTUATORS

3.1.1 Hydraulic actuator with hydraulic conventional fluid

Technical description

The basis of the hydraulic actuator is a hydraulic cylinder with differential effective area (Figure2).

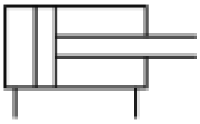

Figure 2: Schematic of hydraulic cylinder

The pressure and cylinder area difference between the rod and the bottom side of the cylinder creates the necessary force to move the cylinder (Figure 3) The acuator needs a hydraulic power supply. The hydraulic power will be produced by a hydraulic pump, which normally is driven by an electro-motor. The control logic of the hydraulic actuator could be realised by an hydraulic valve block in a so called open loop design (Figure 4), where the hydraulic pump is working only in one quadrant (oil flow only in one direction) or with a hydraulic pump which is able to work in minimum 2 quadrants (oil flow could change the direction, by changing the drive direction of the pump, Figure 5))

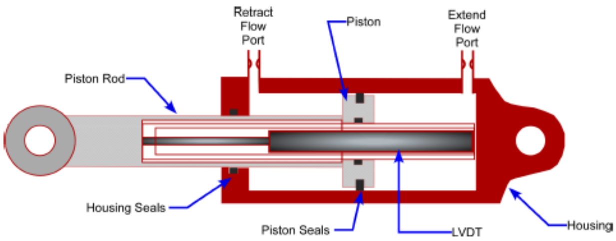

Figure 3: Schematic view of a hydraulic actuator

Figure 4: Hydraulic actuator (open loop design) [1]

Figure 5: Servo hydraulic actuator (closed loop design) (Foto: Bosch Rexroth)

Due to the differential areas of the cylinder an accumulator is requested to collect and supply the differential volume of the oil at the closed loop design, the open loop design is using a reservoir. The electrical power supply will be supported by the vehicle. The control unit should be installed somewhere in the vehicle where less environmental stress will be expected.

Technical data

Table 1: Example of technical data of hydraulic actuators

Stroke	Force	Speed	Frequency
300 mm	200 kN	200 mm/s	1-10 Hz

Pros and Cons

The combination of the high power density of the hydraulic actuator combined with the high precision and dynamic of the electrical control of the driven motor allows an effective power control and precise position control with the hydraulic actuator.

Pros:

- High forces and power density
- Fast response
- Robust system with easy overload protection
- High precision with integrated control and protection function
- Low maintenance
- Power on demand concept
- Compact unit with integrated sensors

Cons:

- The industrial design has to be adapted to cover the environmental requirements and technical regulations of the railway industry.
- Oil is required as power transfer medium

Suitability for railway application

The hydraulic actuator is well known in the railway industry. It has been used for many years as hydraulic dampers in primary and secondary passive suspension systems. The second development phase, the semi-active dampers, have been running in Europe and Asia in intercity and high-speed trains (Shinkansen) for a number of years.
Based on this released technology the third development phase was reached, the active hydraulic actuator. The use and the advantages will be show under item 4.

3.2 HYDRAULIC DAMPER WITH MAGNETORHEOLOGICAL FLUID (MRF) OR ELECTRORHEOLOGICAL FLUID (ERF)

Technical description

In general, the hydraulic actuator with MRF (Figure 6) is a hydraulic damper, which uses the MRF and has an integrated electro magnet at the piston. The electro magnet creates a magnetic field to change the viscosity of the MRF [2]. The change in viscosity influences the damping force.
The hydraulic actuator with ERF uses a controlled high voltage to change the viscosity of the ERF.

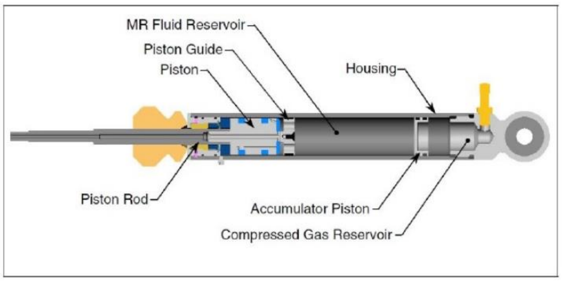

Figure 6: MRF damper

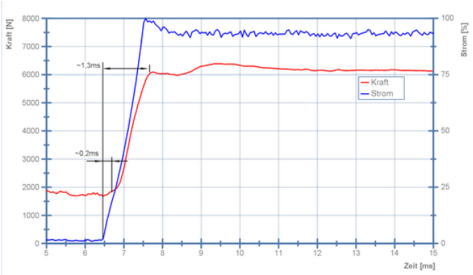

Figure 7: Response characteristic of MRF [2]

The fluids are a mixture of a special carrier fluid and soft solid particles. If an external magnetic (MRF) or electric (ERF) field is applied to the fluid, the particles form into chains,

so that the viscosity of the fluid will increase rapidly (Figure 7) and absorption of the forces will be more effective.
This effect will disappear if the magnetic / electric field is removed. The viscosity can be controlled by varying the intensity of the fields.

Technical data MRF damper

Table 2: Example of technical data of MR damper (Automotive)

Stroke	Force	Speed	Frequency
74 mm	2,5 kN	200 mm/s	> 10Hz

Pros and Cons

Pros:

- Very quick response in changing the viscosity
- Simple design of the damper
- MRF needs less control power than ERF

Cons:

- Weight increase, due to heavy particles at MRF
- Settling of the particles while not using the damper
- Fluid tends to degrade with time causing thickening ⇨change of fluid necessary
- Fluid is inherently somewhat abrasive
- High quality fluid is very expensive
- Chemical resistant of component material has to be evaluated

Suitability for railway application

Hydraulic actuators with MRF are mainly used in automotive applications for semi-active chassis for luxury cars. From the technical point it could be used also for semi- active suspension control in railway applications.
ERF applications for suspension systems are not known.

The high cost of the fluid and the higher maintenance request could limit the applicability for railway applications.

3.3 PNEUMATIC ACTUATORS

Technical description

The pneumatic actuator (figure 8) uses the same principle as the hydraulic actuator. The main differences are the medium (air) and the pressure level. Due to the higher compressibility of the air, a slower response time is inevitable compared to the hydraulic actuator.

Figure 8: Sectional view pneumatic actuator

Pneumatic suspension systems are often used in truck and off-road application, where an automatically mechanical levelling system (Figure 9) could be realized easily and economically with a mechanically-operated proportional valve and pneumatic suspension bags. An air ´power´ supply is required, which consists of an air-compressor and a pressure reservoir, to support the system with sufficient power for short periods of time.

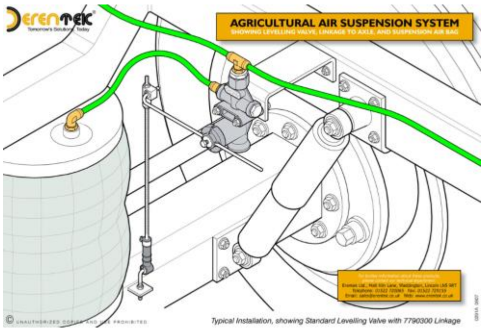

Figure 9: Mechanical levelling system for agricultural machine

This system could also be controlled electrically, where an additional sensor will be necessary (Figure 10)

Figure 10: Basis circuit of electro pneumatic suspension system

The active pneumatic suspension implemented at the Shinkansen high speed train of the Japanese Railway for the active lateral secondary suspension (Figure 11) [3].

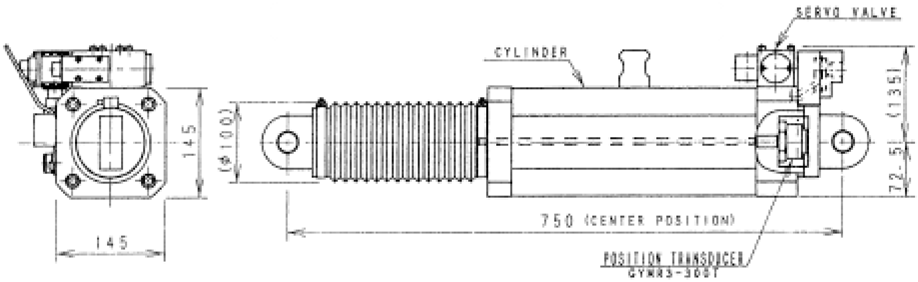

Figure 11: Active pneumatic actuator [3]

Technical data

Working stroke: ± 150mm
Air supply pressure: 5 bar

Pros and Cons

Pros:

- Only pressure piping necessary (free connection to air reservoir)
- Economic solution
- No risk of hazard to the environment
- Soft at high frequency

Cons:

- Longer response time
- Less power density
- Bigger installation envelope requested
- Humid air could create internal damages
- Difficult to find leaks
- High energy consumption due to heat of compression losses

Suitability for railway application

Pneumatic actuators are well known in railway vehicles as secondary suspensions providing excellent vibration isolation and can easy include a levelling function to compensate for load. Other applications are hold-off bump stop systems placed laterally in the secondary suspension and active damping of carbody lateral movements also placed in the secondary suspension. The use of pneumatic actuators is convenient in railway vehicles as they generally have a supply of compressed air for other purposes.

3.4 Electric actuators

3.4.1 Electromechanical actuator

Technical description

The electromechanical actuator is driven by an electromotor. The rotation movement of the electric motor is converted into a linear movement. This translation could be realized with balls-screws or rack and pinions. Depending on the requested power the electrical motor could drive the mechanics directly or via an additional gear stage. Figure 12 shows different solutions with ball-screw technique.

Figure 12: Two electro mechanical actuators with ball screw (Foto: Bosch Rexroth)

Technical data

Table 3: Example of technical data of electromechanical actuators

Stroke	Force	Speed	Frequency
Up to 1700 mm	Up to 290 kN	Up to 1000 mm/s	~ 10Hz

Pros and Cons

Pros:

- High efficiency and band-width
- High force density or high dynamics
- Compact design
- Simple control functions and positioning
- No additional medium requested ⇨No external leakage possible

Cons:

- High stiffness at high frequency

- Less self-damping compared to other solutions
- Risk of jamming
- Expensive

Suitability for railway application

The electro mechanical actuator was first applied for mobile military vehicles. Based on this experience, the industry developed technical solutions for carbody tilting system, which has about the same requirements on force and speed as the military vehicles. Laboratory studies on actuators designed for active suspensions for railway vehicles have been performed.

In the last year a couple of studies were realized with different renowned universities to analyze the possible use for railway applications.

These studies have shown, that the electro mechanical actuator could be used in railway applications, but further developments would be necessary to cover the railway specific requirements.[4]

Particularly, the risk of failure or jamming in the ball-screw mechanism needs to be properly taken into account in order to ensure the required level of reliability.

3.4.2 Electromagnetic actuator

Technical description

The electromagnetic actuator uses magnetic forces to create a movement. For example, the principle is used for an electromagnet valve to operate the control spool for a hydraulic actuator (Figure 13). The valve is positioned by a spring in one position. Actuating the solenoid, the electrical force is working against the mechanical force of the spring. By controlling the electrical forces, the piston of the valve could be positioned between 0% and 100% of the piston stroke.

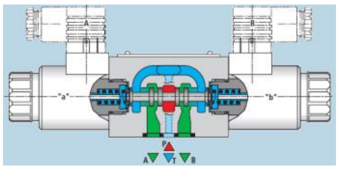

Figure 13: Electromagnetic valve operation (Foto: Bosch Rexroth)

A second principle is the linear electromagnetic motor, such as is used to drive Maglev cars (Figure 14). The principle is based on the electro motor, the only difference is the positioning of the magnets, which are mounted on a linear line. The electric motor changes the magnetic field and the drive forces. This motor type is also known as linear motor.

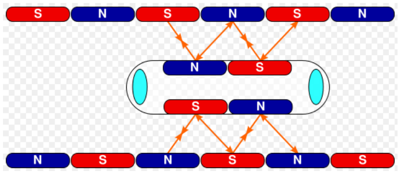

Figure 14: Principle of maglev drive system (Foto: Wikipedia)

A third principle is the electromagnetic propulsion (Figure 15). This principle uses the forces of a permanent magnet and an electro magnet. The magnetic repulsion force is used to control the position of the actuator. The damping effect could be influenced by changing the forces of the electro magnet.

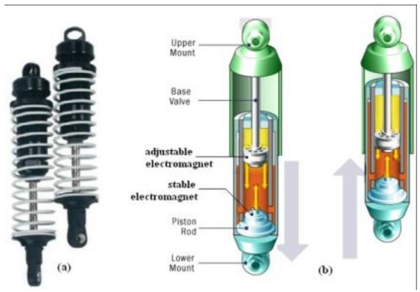

Figure 15: Principle of electromagnetic repulsion damper

Technical data

Table 4: Example of technical data of electromagnetic actuator

Stroke	Force	Speed	Frequency
150 mm	2,5 kN (peak)	Up to 2,5 m/s	~ 17Hz

Pros and Cons

Pros:

- No additional medium necessary
- Economical solution

Cons:

- Low frequency oscillation could create heating problems
- Cooling of the actuator necessary
- Low force/power density without cooling

Suitability for railway application

The electromagnetic actuator was analyzed to be used at automotive applications for active suspension systems. Various prototypes were build up together with universities to demonstrate the functionality. The latest solution was realized in 2011 at the University of Eindhoven, see also section 5.1. The results were interesting, but the industry did not pick up the technical solutions.

4. ACTUATORS IN RAILWAY APPLICATIONS

4.1 INTRODUCTION

Railway vehicles have incorporated actuators for many years as part of tilting train implementations, including hydraulic, electro-mechanical and pneumatic actuation. Since tilting technology is now relatively mature the industry has some experience of actuation for control purposes. Applications to other forms of active suspension are however very limited, and so this section provides an overview of what actuation technology has been used, mainly experimentally.

4.2 BOGIE CONFIGURATION – SECONDARY SUSPENSION

Most actuation technologies have been exploited in order to provide improved ride quality in the lateral direction. They have been employed directly using dynamic forces to give improved suspension response, for example using so-called “skyhook damping” or variations thereof. Alternatively, they have been used indirectly with quasi-static control to minimize the lateral deflection on curves, thereby enabling (for example) a softer lateral spring. A number of semi-active solutions have also been explored (hydraulic and magneto-rheological) although these cannot provide quasi-static forces and in general do not offer the same level of performance improvement as the fully-active approaches.
Modern air spring suspensions mostly offer excellent vertical ride quality and so there has been less experimentation relating to the vertical suspension

- Performance requirements: force levels of up to 10-15kN are typical, providing +/- 30-40 mm of stroke. Maximum frequencies are c. 1Hz for the quasi-static approaches and 10Hz for the full dynamic control.
- Pro and contra of the actuators
    - Pneumatic [5, 6, 7, 8]:
        - Cheap
        - Mechanically simple (but large in volume)
        - Dynamically soft
        - Limited frequency capability
        - Low efficiency
        - Not consistent with longer term trends towards more- (or all-) electric vehicles
    - Servo-hydraulic [9]:
        - Very compact at point of application
        - High frequency capability
        - Dynamically stiff
        - Low efficiency
        - High pressure hydraulic supply not generally available on modern rail vehicles
        - Relatively expensive and requiring significant maintenance

- Electro-mechanical 9, 10, 11, 12]:
  - Powered directly from vehicle electrical supply
  - High efficiency
  - Compatible with existing railway technology
  - Medium cost
  - Moderate frequency capability
  - Dynamically stiff
  - Concern about failure of ball-screw
- Electro-hydraulic [13]:
  - Powered directly from vehicle electrical supply
  - High efficiency
  - Medium cost
  - Moderate frequency capability
  - Dynamically stiff
- Electromagnetic (direct-acting) [9, 14]:
  - Powered directly from vehicle electrical supply
  - High efficiency
  - Dynamically soft
  - Low maintenance
  - Medium cost
  - Mechanically large (significant disadvantage)
  - Complex and expensive power electronics equipment required

4.3 BOGIE CONFIGURATION – PRIMARY SUSPENSION

Known applications are very limited, and none has been used in service operation, only for experimental testing. All have been focused upon wheelset yaw control: to provide steering around curves, to enhance stability or to provide guidance with respect to the track. Note that guidance control intrinsically provides steering but uses some measure of wheel-rail lateral deflection, whereas active steering uses some measure of the design alignment of the track, e.g. curvature.

- Performance: Force requirements depend upon the active control objective (guidance, steering or stability) and also upon the type of wheelset. Similarly, the bandwidth requirement is varied, relatively low frequency (<1Hz) for active steering, but up to 10Hz or higher for stability or guidance control. For heavy rail applications the maximum yaw angle will be small, typically giving +/- 25mm at the axle boxes i.e. 2-3°.
- Pro and contra of the actuators
  - Electric motor actuation [15]: the electrical motor drives via a gearbox and mechanical linkages onto the axle boxes to give independent control for the two axles. The motors were rated at about 2kW and were designed to provide both stability control and active steering.

- Powered directly from vehicle electrical supply
- High efficiency
- Compatible with existing railway technology
- Medium cost
- Moderate frequency capability
- Dynamically stiff

▪ Electric motor actuation [16] driving through a transmission to give a differential torque to a wheelset with independently-rotating wheels. The scheme was developed and demonstrated on a 1:5 scale roller rig, and forms the basis for DLR's NGT proposal [17]. (A similar reduced-scale arrangement was developed in Japan [18].)
  - ✓ Powered directly from vehicle electrical supply
  - ✓ High efficiency
  - ✓ Compatible with existing railway technology
  - Medium/high frequency capability
  - Medium cost
  - Dynamically stiff

▪ Electric motor actuation [19] using wheel motors (rated at c. 70kW) in which the motor is embedded within the structure of the wheel to provide a bogie with four motors that are controlled to give traction, braking, steering and guidance.
  - ✓ Powered directly from vehicle electrical supply
  - ✓ High efficiency
  - Compatible with existing railway technology, but a radically new configuration
  - High frequency capability
  - Medium cost?
  - Dynamically soft via high performance torque control

▪ Semi-active "creep-controlled" wheelset [20], which used an electro-magnetic coupling to vary continuously between solid-axle and independently-rotating wheels configuration.
  - ✓ Low power requirement
  - ✓ High efficiency
  - ✓ Compatible with existing railway technology
  - Medium cost
  - Moderate frequency capability

4.4 SINGLE-STAGE TWO-AXLE CONFIGURATION

Although there have been studies of primary and secondary suspensions the only application that is known was the "Dosaged torque" concept developed in Germany (RTH

Aachen) in the 1980s [21]. This used controlled induction motors to provide “single-axle running gear”, but there is very limited information available.

- Pro and contra of the actuators – similar to the ‘Electric motor actuation using wheel motors’ application described above.

5. ACTUATORS IN NON-RAILWAY APPLICATIONS

5.1 AUTOMOTIVE ACTUATORS

This section of the deliverable aims at providing an overview of actuators in use in sectors different from railways, with a focus on applications in the automotive field. According to [22], the present trend in the automotive market is to have high-end cars equipped with a pneumatic load levelling system combined with a semi-active damper. These suspension systems and others are described below in better detail.

5.1.1 Fully-active suspensions

The Magneti Marelli lifting system

This system is available on some high-end sport cars and is used to change the ground clearance of the car, based on a command defined by the driver by pressing a button [23]. At low speed the car runs at its maximum height from ground to overcome the angle of slope or the obstacles (bumps, holes) without damaging the chassis; at high speed it is requested to have the lowest height to reduce the aerodynamic drag coefficient and improve stability.
In the Magneti Marelli system actuation is realized by means of hydraulic actuators integrated in the otherwise passive suspension of the vehicle (Figure 16).
This system is only for quasi-static actuation and can be compared to levelling systems in railway vehicles.

Figure 16: The Magneti Marelli lifting system, from [23]

The Mercedes Benz "Active Body Control"

This is a proprietary system of Mercedes Benz, used to actively control the car body motion [24]. The system (Figure 17) is driven either by a set of vehicle-mounted sensors (accelerometers and level sensors) or by a device measuring the road surface geometry (this in turn can be based on lidar sensors or on a stereo camera). In the latter case, the system takes the commercial name of "Magic Body Control" (Figure 18)

Applications of this mechatronic suspension concept include:

- compensation for road roughness to improve ride comfort;
- crosswind stabilization of the car;
- active curve tilting: in the same way as in tilting trains, when the vehicle rides through a curve the car body is tilted by an angle up to 2.5 deg to partly compensate the effect of the centrifugal force.

The system uses high-pressure hydraulic servo-actuators mounted in the spring struts between the coil springs and the body. No detailed information could be retrieved regarding the maximum actuation force or power consumption, but the system is said to have a pass-band up to 5Hz, which is believed sufficient to control carbody vibration caused by uneven road surfaces or by braking and cornering.

Figure 17: Layout of actuators in the Mercedes Benz "Active Body Control"/"Magic Body Control" system, from [24].

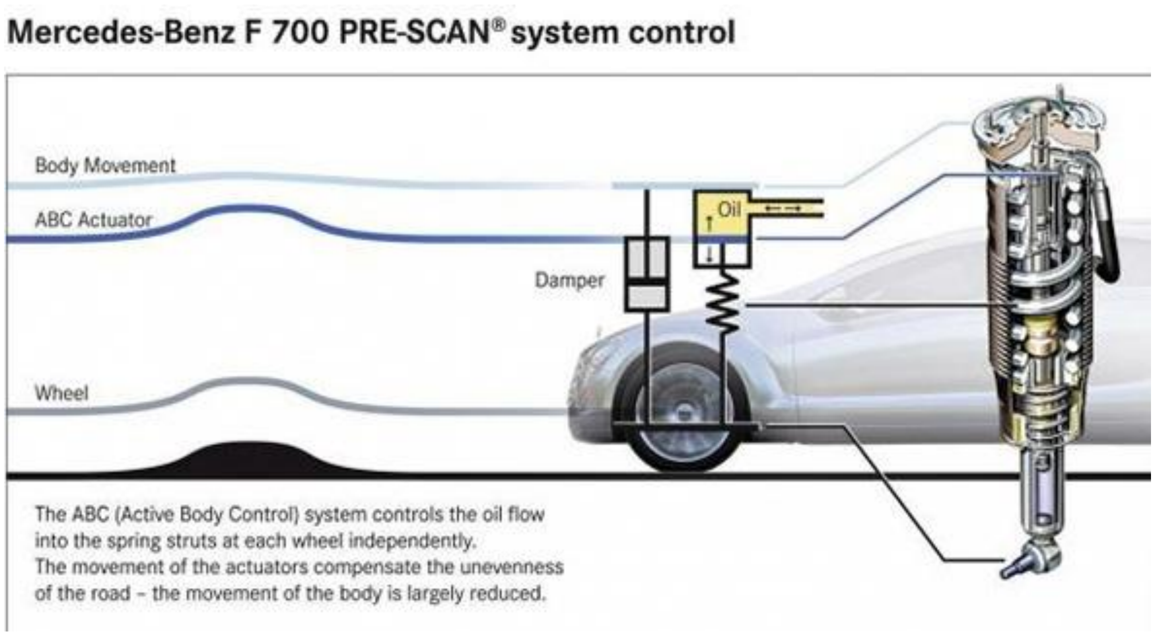

Figure 18: Active car body control based on road surface scan in the Mercedes Benz "Magic Body Control" system, from [24].

The Active Electromagnetic Suspension from TU-Eindhoven

TU-Eindhoven in the Netherlands designed an active suspension for road vehicles based on permanent magnet linear electromagnetic actuators. Prototypes of this system were laboratory tested and subjected to test drives on roads [25, 26]

This active suspension is designed to reduce car body vibration enhancing ride comfort but also to improve vehicle stability and passenger safety.

Figure 19 shows the concept of the system, in comparison to a 'classic' passive suspension (a). Two versions of the concept are shown, with interior/exterior magnet respectively (b, c). In both cases, actuation is realized using a brushless three-phase tubular permanent-magnet actuator (TPMA).

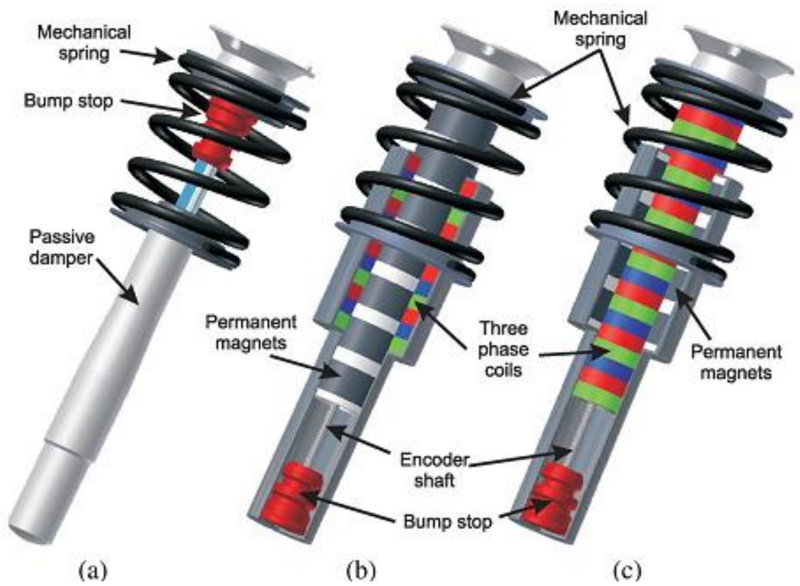

Figure 19: Concept of the Active Electromagnetic Suspension from TU-Eindhoven: (a) Passive. (b) Active interior-magnet TPMA, (c) Active exterior-magnet TPMA (from [25]).

Compared to other concepts in use for active suspensions in road vehicles, this actuator has a relatively high force density due to the tubular structure and, according to what is reported in [24], also has excellent servo characteristics with a bandwidth in excess of 50 Hz. Furthermore, electromagnetic actuators allow for bidirectional power flow, which makes it possible to design a regenerative suspension having improved energetic efficiency. Active mode can be used to apply active forces on the car body to control roll and pitch movements, whilst passive mode can be used to absorb road vibrations operating the suspension as a semi-active damper. In this latter case, the absorbed power can be fed to a battery in order to supply auxiliary loads or to be stored for future use to feed the suspension when operated in active mode.

This active suspension was designed to be capable of generating a continuous force up to 750 N at a maximum speed of dilatation/contraction equal to 1 m/s and a peak force of 4 kN but only at speeds below 0.1 m/s. The active suspension is said in [26] to require a power which is on average in the order of magnitude of hundreds of Watts when operated in active mode, although an instantaneous peak power of 2 kW is necessary in some specific maneuvers. The disadvantages of this concept are presently a relatively low power density (ratio of power vs. mass/volume), possible problems with heat dissipation, challenging design of the power electronics part of the suspension system, high cost and relatively large weight.

5.1.2 Semi-active suspensions

Compared to fully-active suspensions, semi-active suspensions are more widely used at present in road vehicles and several examples are available.

Continuously variable hydraulic semi-active dampers

Continuously-variable hydraulic semi-active dampers are based on the use of a hydraulic valve to control the pressure drop across the chambers of a hydraulic damper [27 - 29]. A scheme of the hydraulic circuit of such an adjustable hydraulic damper is shown in Figure 20, whereas a cross-section view of the same device is shown in Figure 21.
A proportional valve (P_{vl} in Figure 20) controls the pressure drop through the circuit, thereby regulating the damping coefficient. A reservoir accommodates the oil displaced by the volume of the piston rod. The reservoir is pressurized with nitrogen gas to prevent cavitation occurring.
The semi-active damper described in [30] is capable of generating a force up to 10-12 kN for a damper velocity constant at 150 mm/s.
A different concept for a continuously variable hydraulic damper is described in [27]. The advantage of this concept is that the damping characteristic of the device can be adjusted automatically depending on the vibration mode of the vehicle body.
According to [22], continuously variable hydraulic semi-active dampers have a pass-band of 50-60 Hz which is inferior to magnetorheological dampers (see Subsection below) but still fully adequate for controlling ride comfort in a road vehicle. Again according to [22], this technology presently has a 90% share of the total of semi-active dampers in the automotive market, with the remaining 10% being allocated to magnetorheological dampers.

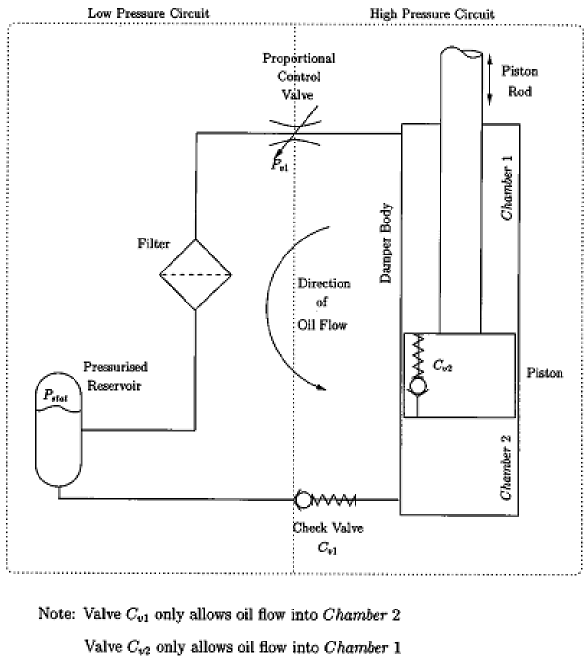

Figure 20: Scheme of the hydraulic circuit of a continuously variable hydraulic semi-active damper (from [27]).

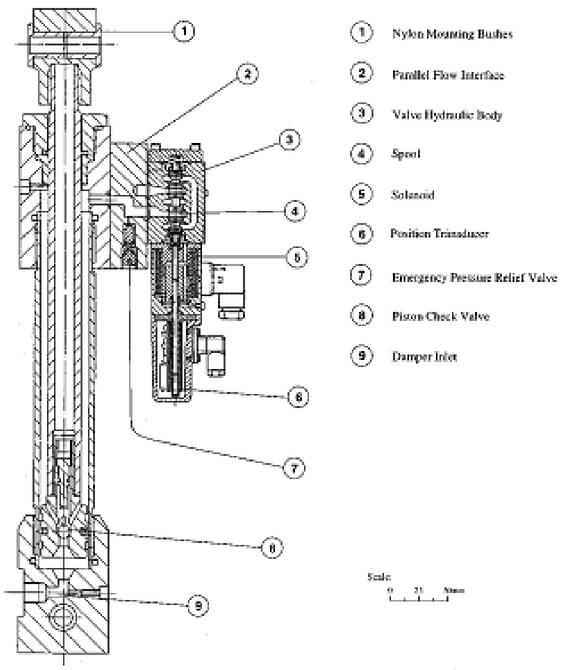

Figure 21: Cross-section view of a continuously variable hydraulic semi-active damper (from [27]).

Magnetorheological semi-active dampers

Magnetorheological (MR) dampers are semi-active dampers that exploit the property of some fluids to change their viscosity depending on the intensity of a magnetic field to which they are exposed. A similar principle is exploited in electrorheological dampers (ER), in which the change of viscosity depends on the fluid being exposed to an electric field.

ER and MR fluids are both obtained by mixing fine particles into a liquid with low viscosity. When a magnetic / electric field is applied, the particles in the fluid form chains and the fluid shows a Bingham behaviour, i.e. flow can happen only once a threshold of yield stress is reached. When the field is removed, the behaviour of the fluid is back to Newtonian with low viscosity. The process of change of fluid properties with the application of the magnetic / electric field is very fast and hence it can be easily controlled. A data sheet illustrating the properties of a MRF in use for various applications can be found in [31].

Compared to other concepts for semi-active dampers, like hydraulic dampers with variable orifice, MR dampers are characterized by a faster response, as it just take milliseconds after a change in the magnetic field applied to generate a change in the viscosity properties of the fluid, whereas the time needed to operate mechanically a change in the size of the orifice in a variable orifice damper is much greater. Moreover, MRF can operate at temperatures in the range -40 to 150° with only slight variations in their yield stress. However, MR dampers are more expensive than other semi-active dampers.

MR dampers are used in different fields, including road and rail vehicle suspensions, industrial suspension systems such as cab suspensions, seat suspensions etc., vibration control for fixed-wing and rotary-wing aircrafts. In this section an example is described concerning their application to civil engineering structures for earthquake hazard mitigation [32].

A schematic drawing of the MR damper is shown in Figure 22: the concept is similar to a standard hydraulic damper, except for a coil hosted in the piston head which is used to generate the magnetic field required to change the rheological properties of the and for the cabling of the electric wires needed to feed electric current in the coil. The current in the coil is supplied by a linear current driver which generates a 0–1 Amp current proportional to an applied DC input voltage in the range 0-3 V. The power required to operate the coil is less than 10 W. The damper is 21.5 cm long in its extended
position, and the main cylinder is 3.8 cm in diameter.

According to paper [32], this damper is capable of generating forces up to 3 kN with velocities up to some tens of cm/s. In the same paper it is said however that MR dampers for use in civil engineering structures could be designed with maximum forces up to 200 kN, but of course the size of these dampers is expected to be very large.

According to [21], MR dampers have a wider pass-band compared to electro-actuated semi-active hydraulic dampers. However, the wider pass-band only provides limited benefit in terms of improved ride control, whereas one disadvantage of MR and especially of ER dampers compared to electro-actuated semi-active hydraulic dampers is represented by the design requirements of the power-electronics part of the semi-active suspension.

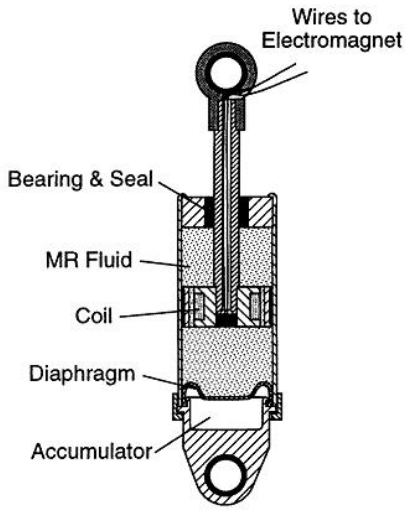

Figure 22: Schematic drawing of a MR damper (from [31]).

5.2 AIRCRAFT ACTUATORS

5.2.1 Introduction

To quote from (Jänker et al., 2008), *"Actuation technology is one of the critical technologies in aerospace. During the last years the concept of the more electric aircraft was pushed ahead by industry and scientific community. The adaptation of electric drive train technology to meet with the demanding requirement of aerospace is in the focus of the activities."* [34] More- and all-electric aircraft concepts are seen as an inevitable trend which is strongly driving the use of electrically-driven actuation, and the following sub-sections summarize the situation.

5.2.2 Summary of actuation systems

Servo-hydraulic actuation:

"Conventional" hydraulic actuators, controlled via servo-valves from a centralized hydraulic power supply, have been standard technology in aircraft for many years. Despite this the technology is seen as low efficiency, heavy and high maintenance, and generally not consistent with the more- and all-electric aircraft trends. However, flight-critical functionality such as anti-jamming and overload protection are seen as satisfactory. They are still being used for the primary flight control surfaces for the Boeing 787.

Electro-hydraulic actuation (EHA):

These have a hydrostatic transmission to avoid having a gearbox, ball screw, clutch and torque limiter, and are seen as a mature technology for aircraft. They are used for the Joint

Strike Fighter (F-35), i.e. no conventional hydraulic actuators are included even for the primary flight control surfaces. The Airbus A380 uses EHA technology, but in combination with conventional hydraulic actuation (2 electric and 2 hydraulic). However, since the oil degrades maintenance access to the hydraulic connections is still required, so the vision of a fully sealed-for-life system has not yet been realised.

Electro-mechanical actuation (EMA):

These are not yet seen as a mature technology. A key issue to be resolved is that of anti-jamming of the ball-screw/nut mechanism, although a ”jam-tolerant” actuator has been patented (Nguyen et al., 2014) [36]. However, this is mechanically complex with two ball screws and nuts supported by a “Stored Energy Device” and hence perhaps not highly appropriate for railways. Another issue is that of achieving an optimised duty cycle design to specify a minimum motor size that does not overheat under all conditions, and this remains an important target. An interesting development is a SAAB EMA design (used as a backup on the Boeing 787) [35] which provides “Internal thermal and galvanic redundancy” via the use of a high power density, split phase permanent magnet synchronous motor, although it does not incorporate anti-jamming capability.

Other:

The aerospace industry has a strong interest in piezo-electric actuation for active vibration control, and this may be relevant to active secondary vertical suspensions on rail vehicles for which flexible body vibration can be a problem.

5.2.3 Relevance to railway vehicles

It’s clear that aircraft technology is pushing in the direction of electric actuation, and in the long term railways can take advantage of the aircraft industry’s developments, just as has already happened with the Liebherr EHA device used by Bombardier. In general actuation technology development for aircraft is generally consistent with rail industry thinking, although conventional servo-hydraulic actuation has not usually been seen as very suitable for rail vehicles.

6. REQUIREMENT SPECIFICATION OF ACTUATORS IN RAILWAY APPLICATIONS

The intention of this section is to describe requirements for possible semi-active and active suspension systems on railway vehicles in a selected set of applications. Even though all the applications in this section refer to railway vehicles, the requirements could be very different depending on what the intended function is. A first differentiation is whether the actuator shall be able to produce a quasi-static force. For an intended application to attenuate vibrations of a body, a semi-active damper may be sufficient. The semi-active damper can in this case dissipate energy from the suspension system and thereby reduce the body motions. For an application where a certain position of the body is requested, i.e. keeping the carbody centered above the bogie in curves, the actuator must be able to insert energy in the system. The different types of requirements are described in the first sub-section.

The requirements given here are for the described application, although sometimes it is possible to combine two applications in one hardware. One example is a quasi-static secondary hold-off bump stop function which can be combined with dynamic secondary damping for improved ride comfort. In this case the actuator must both be able to give a quasi-static force and to attenuate vibrations, hence the two set of requirements must be combined.

6.1 TYPES OF REQUIREMENTS

Type of actuator

There are two types of actuators, those that can produce a quasi-static force and those that cannot. The first type requires that power is inserted in the system and they are called active (or fully active) actuators. The other type can only dissipate energy from the suspension system and they are called semi-active actuators. Normally the active actuators are more complex, are heavier and are more expensive.

Force

The amount of force the actuator needs to handle depends on the application. Some applications require force in both directions, whereas other applications can accept that force is created in only one direction.

Relative speed

The relative speed is the speed between the two mechanical connection points of the actuator, describing how fast the actuator can displace its movable part relative to its fixed part. The term relative speed is used for actuators intended to displace a body like a wheelset or a carbody. For actuators intended to attenuate motion it is more relevant to set a requirement on frequency range (see below). An actuator intended to give the wheelset a radial position in curve needs to relocate the axle longitudinally a few mm and may take a few seconds to achieve this. Compare this with an application, where the actuator must be able tilt the carbody, which could require relative speeds up to 100 mm/s.

Power
Power is the product of relative speed and force. Power is therefore seldom used as a requirement and is excluded here. However, power may be the limiting factor for the actuator as the power source could be limited, or there may be a limited capability of dissipating waste energy.

Frequency range
The term frequency range is used for actuators intended to attenuate motion. A common definition of the cut-off frequencies (the limits of the range) are when the actuator has lost 50% of its force capability. The actuator must be able cover the frequency range of the application.

Stroke
The actuator stroke must be large enough to ensure that the actuator is not limiting the motion in service. However, a bump stop could be included in the actuator to allow the actuator to act as a stop. The stroke is here defined as ± from a central position, this means that the stroke from one extreme to the other is twice as large.

Stiffness
The stiffness has in some applications a very large influence on the performance. Damping bogie hunting motions is one example where high stiffness is needed. If the actuator is soft, then it will start act like a spring above a certain frequency and can therefore not dissipate energy from the bogie hunting motions. On the other hand, an actuator placed in parallel with the suspension must be soft so as not to transfer vibrations from one body to another. This is a very important parameter, often decisive for the choice of actuator type.

6.2 QUASI-STATIC WHEELSET YAW FOR RADIAL STEERING

6.2.1 Intention of the application

The intention of quasi-static wheelset yaw is to achieve optimal radial alignment of the wheelset to reduce the angle of attack between wheel and rail. Wheelsets which take a radial position will both reduce the running resistance and the wear on wheel and rail. The application is not limited to the conventional solid-axle wheelset configuration, but can also be applied for independently rotating wheels without a common axle. The intention is to displace the wheelset in a longitudinal direction on each side and thereby turn the wheelset in the yaw direction, see Figure 23. The actuator is assumed to be set in parallel to the conventional primary longitudinal suspension. An alternative solution is to set the actuator in series with the conventional primary longitudinal suspension, but this setup leads to safety issues and is therefore not considered here. One simplification could be to just have an actuator on one side of the wheelset. The applied force should be quasi-static and with relation to the curve radius, the smaller radius the larger the force. The actuator must be of the active type as the force requested is quasi-static. The actuator will typically be located longitudinally between the running gear frame and the axle box. Quasi-static wheelset yaw for radial steering is also applicable to single axle running gears.

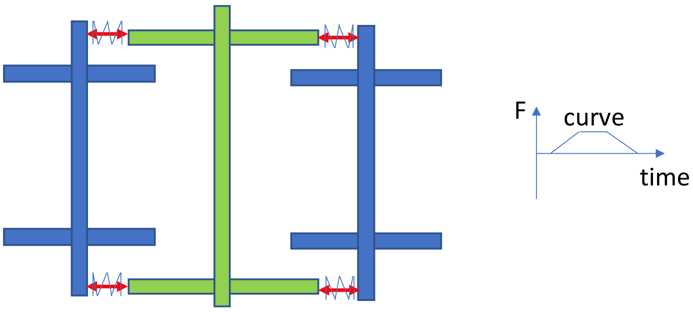

Figure 23: Quasi-static wheelset yaw for radial steering

6.2.2 Requirements

The actuator must be able to create a quasi-static force large enough to displace the axle box to get a radial position of the wheelset (Table 5). The actuator will be counteracted by the primary longitudinal suspension if placed in parallel. The stiffness of the primary longitudinal suspension is often set by the requirement of stable running on straight track, hence it cannot be softened. The stroke must be large enough to allow the wheelset to take a radial position, which is just a few mm. The speed of the actuator must be large enough to allow the wheelset to take the correct position within a certain time. Most transitions from one track element to another are equipped with a dedicated transition element, i.e. a curve transition that connects straight track to curve. Such elements are normally a few seconds' long allowing vehicle and passengers to adopt to the new condition.

Table 5: Quasi-static wheelset yaw for radial steering

Property	Requirement
Type	Active
Force	5 – 25 kN
Speed	1 – 4 mm/s
Stroke	± 10 mm
Stiffness	No requirement

6.3 Dynamic wheelset yaw control to suppress hunting

6.3.1 Intention of the application

The intention of dynamic wheelset yaw control is to extend the speed range for the running gear to higher speeds, without risk for hunting, than would otherwise have been possible. Optionally the running gear can be made more track friendly by selecting a softer primary longitudinal stiffness. The application is intended for the conventional wheelset configuration as the running gears with free rotating wheels are not prone to hunting in the same way. The Intention is to control the wheelset in longitudinal direction and thereby attenuate the wheelset tendency to rotate in yaw direction, see Figure 24. The applied force should be dynamic. One simplification could be to just have an actuator on one side of the wheelset. The arrangement could also be applied on single step suspensions, the actuator will then be connection the wheelset with the carbody. The actuator could be either of the semi-active or the active type. The actuator will typically be located longitudinally between the running gear frame and the axle box. Dynamic wheelset yaw control to suppress hunting is also applicable to single axle running gears.

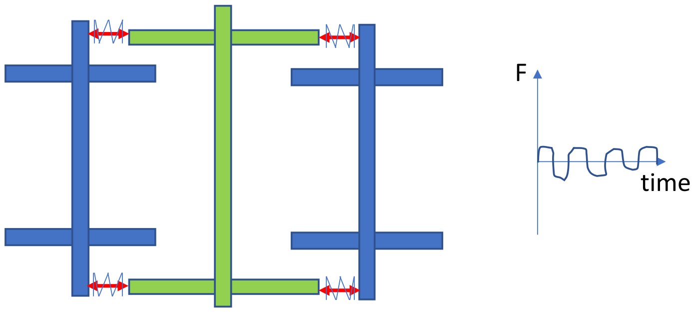

Figure 24: Dynamic wheelset yaw control to suppress hunting

6.3.2 Requirements

The actuator must be able to create a dynamic force large enough to control the hunting tendency of the wheelset. The stroke must be large enough to allow the wheelset to take a radial position, which is a few mm. The frequency range of the actuator must be large enough to cover all likely hunting frequencies (Table 6).

Table 6: Dynamic wheelset yaw control to suppress hunting

Property	Requirement
Type	Semi-active or active
Force	5 – 10 kN
Frequency	1 – 10 Hz

Stroke	± 10 mm
Stiffness	High

6.4 Dynamic wheel speed control for radial steering

6.4.1 Intention of the application

The intention of dynamic wheel speed control to allow a wheelset with free rotating wheels to take a radial position in curves. A wheelset with free rotating wheel is not prone of hunting in the same way as conventional wheelsets and the speed range can thereby be extended, but such a wheelset does not have the same built-in ability to take a radial position in curves as conventional wheelsets. For driven running gears there is a possibility to provide differential wheel speed and thereby force the wheelset to a radial position. The intention is to control the wheelset yaw relative the running gear frame, see Figure 25. The applied force should be dynamic as the motor at the same time may provide traction and braking. Dynamic wheel speed control for radial steering can be applied on single axle running gears and running gears with more axles.

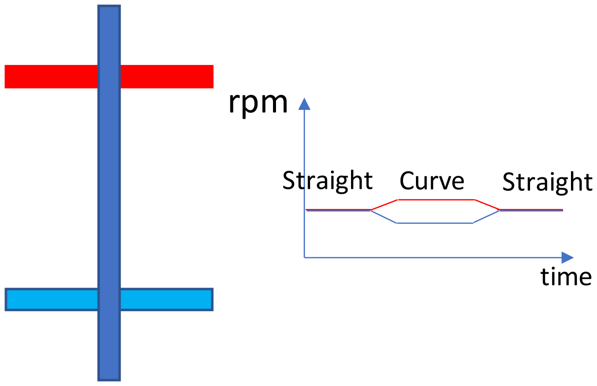

Figure 255: Dynamic wheel speed control for radial steering

6.4.2 Requirements

The actuator (the traction motor) must be able to create a dynamic force large enough to position the wheelset radially, which depends upon the chosen value of primary yaw stiffness.

6.5 DYNAMIC PRIMARY DAMPING FOR IMPROVED RIDE COMFORT

6.5.1 Intention of the application

The intention of dynamic primary damping is to attenuate suspension resonances and carbody eigen modes. Dynamic primary damping can preferably be applied on single step suspensions, allowing simpler and cheaper running gears to be used for passenger services. The application could be either in lateral or vertical direction, see Figure 26. A conventional damper in this application must be selected to ensure that bogie frame resonances are attenuated under all conditions, and the semi-active or active damper must be soft when there are no motions detected in order to transfer less vibration from the wheelset to the running gear frame.

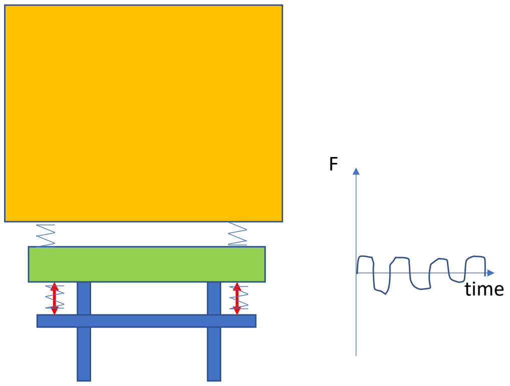

Figure 26: Dynamic primary damping for improved ride comfort

6.5.2 Requirements

The actuator must be able to create a dynamic force large enough to control the rigid body motion of the frame and/or the carbody. The stroke must be large enough to allow the full suspension travel, this is many mm. The frequency range of the actuator must be large enough to cover all likely rigid body motion frequencies (Table 7).

Table 7: Dynamic primary damping for improved ride comfort

Property	Requirement
Type	Semi-active or active
Force	5 – 10 kN
Frequency	1 – 10 Hz
Stroke	± 40 mm
Stiffness	Soft an advantage

6.6 DYNAMIC RUNNING GEAR YAW CONTROL TO SUPPRESS HUNTING

6.6.1 Intention of the application

The intention of dynamic running gear yaw control is to extend the speed range for the running gear to higher speeds, without risk for hunting, than it otherwise would have been possible. Optionally the running gear can be made more track friendly by selecting a softer primary longitudinal stiffness. The application is intended for conventional wheelset configuration as the running gears with free rotating wheels are not prone to hunting in the same way. The intention is to control the running gear in a longitudinal direction on each side and thereby attenuate the running gear tendency to turn in yaw direction, see Figure 27. The applied force should be dynamic. The actuator could be either of the semi-active or the active type. The actuator will typically be located longitudinally between the carbody and running gear frame.

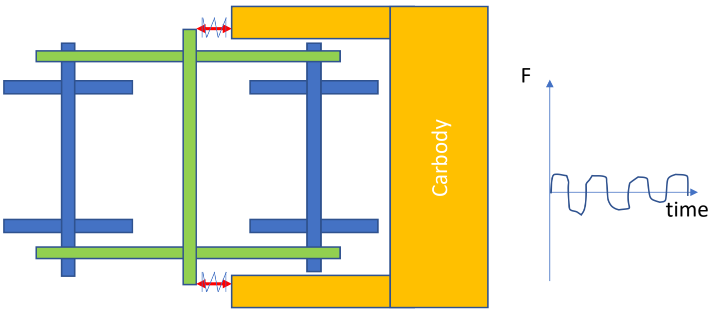

Figure 27: Dynamic running gear yaw control to suppress hunting

6.6.2 Requirements

The actuator must be able to create a dynamic force large enough to control the hunting tendency of the running gear. The stroke must be large enough to allow the running to properly run though horizontal curves, which is many mm. The frequency range of the actuator must be large enough to cover all likely hunting frequencies. The actuator must have high stiffness as it otherwise will act as a spring above a certain frequency and thereby lose its possibility to attenuate the hunting (Table 8)

Table 8: Dynamic running gear yaw control to suppress hunting

Property	Requirement
Type	Semi-active or active
Force	5 – 20 kN
Frequency	1 – 10 Hz
Stroke	± 100 mm

Stiffness	High

6.7 Quasi-static running gear yaw control to improve curving

6.7.1 Intention of the application

The intention of quasi-static running gear yaw control is to improve the curving performance in curves. The intention is to displace each side of the running gear in longitudinal direction and thereby rotate (yaw) with reference to the carbody, see Figure 28. This will equalize the lateral track forces between the two wheelsets and thereby reduce the higher force and reduce the wheel and track wear. The actuator will typically be located longitudinally between the carbody and running gear frame. The applied force should be quasi-static with relation to the curve radius, the smaller the radius the larger the force. The actuator must be of the active type as the force requested is quasi-static.

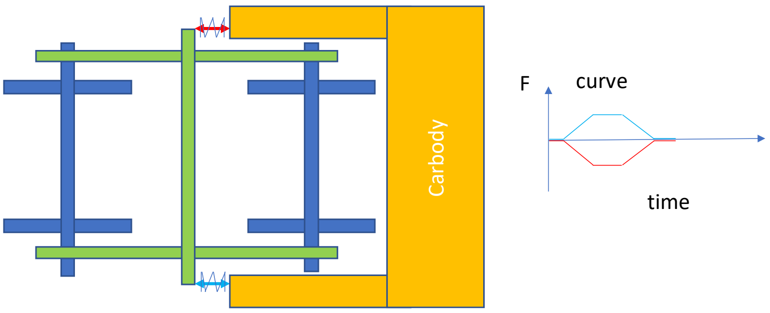

Figure 288: Quasi-static running gear yaw control to improve curving

6.7.2 Requirements

The actuator must be able to create a quasi-static force large enough to displace the running gear to get a radial position on the running gear (Table 9). The actuator will be counteracted by the secondary longitudinal suspension. The stroke must be large enough to allow the running gear to take a radial position. The speed of the actuator must be large enough to allow the running gear to take the correct position within a certain time. Most transitions from one track element to another are equipped with a transition element, i.e. a curve transition that connects straight track to curve. Such elements are normally a few seconds long allowing vehicle and passengers to adopt to the new condition.

Table 9: Quasi-static running gear yaw control to improve curving

Property	Requirement
Type	Semi-active or active
Force	5 – 20 kN

Speed	20 – 50 mm/s
Stroke	± 100 mm
Stiffness	High

6.8 Dynamic secondary damping for improved ride comfort

6.8.1 Intention of the application

The intention of dynamic secondary damping is to attenuate suspension resonances. The application could be either in lateral or vertical direction, see Figure29. A conventional damper in this application must be selected to ensure that the suspension resonances are attenuated under all conditions, the semi-active or active damper can be soft when there are no rigid body motions detected. A soft damper transfers less vibration from the running gear frame to the carbody. Dynamic secondary damping for improved ride comfort is a mature application today.

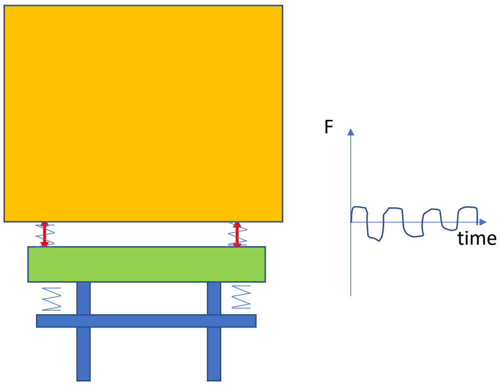

Figure 29: Dynamic secondary damping for improved ride comfort

6.8.2 Requirements

The actuator must be able to create a dynamic force large enough to control the rigid body motion of the carbody. The stroke must be large enough to allow the full suspension travel, this is many mm. The frequency range of the actuator must be large enough to cover all likely rigid body motion frequencies (Table 10)

Table 10: Dynamic secondary damping for improved ride comfort

Property	Requirement
Type	Semi-active or active
Force	5 – 10 kN

Frequency	0.5 – 10 Hz
Stroke	± 60 mm
Stiffness	Soft an advantage

6.9 QUASI-STATIC SECONDARY FORCE TO HOLD-OFF BUMP STOPS

6.9.1 Intention of the application

The Intention of the application is to keep the carbody away from the lateral bump-stop. Any contact with the stiff bump-stop will short-circuit the secondary suspension and thereby dramatically deteriorate the ride comfort. The application will allow selecting a softer secondary lateral suspension than otherwise possible, which provides improved ride quality. This application is particularly important for vehicles running at high speeds in curves. Quasi-static secondary force to hold-off bump stop is a mature application today.

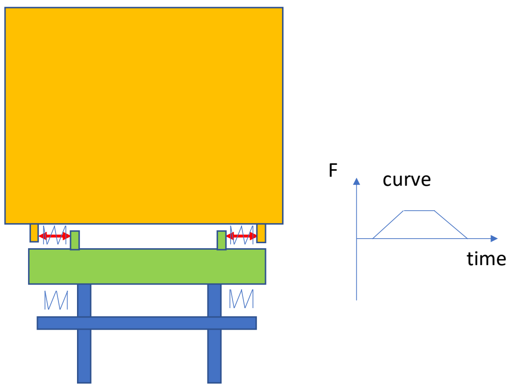

Figure 30: Quasi-static secondary force to hold-off bump stops

6.9.2 Requirements

The actuator must be able to create a quasi-static force large enough to keep the carbody away from the bump-stop while curving. The stroke must be large enough to allow the suspension travel, which will be tens of mm (Table 11) The actuator must be soft for superposed displacements as the actuator is placed in parallel with conventional secondary suspension.

Table 11: Quasi-static secondary force to hold-off bump stops

Property	Requirement
Type	Active
Force	10 – 25 kN

Frequency	– 0.5 Hz
Stroke	± 60 mm
Stiffness	Soft

6.10 QUASI-STATIC VERTICAL LOAD COMPENSATION

6.10.1 Intention of the application

The intention of the application is to keep the carbody (floor) at the same level independent of passenger load to allow easy entering. The functionality is important for vehicles with large load to weight ratio, such as metro vehicles. Quasi-static secondary load compensation is a mature application today, and the standard solution involves air suspension and levelling valves to control the height. Load compensation may also be installed in vehicles with single step suspensions. Separate systems for load compensation allow use of suspension technologies other than air suspensions.

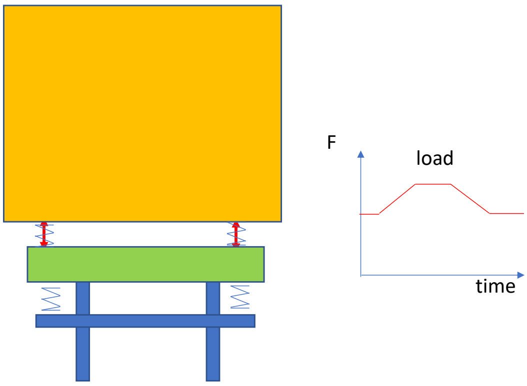

Figure 291: Quasi-static vertical load compensation

6.10.2 Requirements

A load compensation system can be setup in three principal different ways. 1. The actuator isolates vibration and compensates for the passenger load, 2. The actuator is in series with the conventional suspension and compensates the suspension compression at passenger load, 3. The actuator is in parallel to the suspension system and compensates for the passenger load. The requirements depend on the system setup. For the combined and in-series setup, the actuator must be able to carry the carbody weight plus the passenger load, whereas in the parallel setup the actuator must only carry the passenger load. The actuator should be soft for the combined and in parallel setups, but can be stiff for the in-series setup (Table 12)

Table 12: Quasi-static vertical load compensation

Property	Requirement		
System setup	Combined	In-series	In-parallel
Type	Active	Active	Active
Force	100 kN	100 kN	50 kN
Frequency	– 0.5 Hz	– 0.5 Hz	– 0.5 Hz
Stroke	± 60 mm	± 10 mm	± 60 mm
Stiffness	Soft	High	Soft

6.11 ACTIVATION OF NON-LOAD CARRYING CARBODY TILT

6.11.1 Intention of the application

The intention of the application is to activate carbody tilting. The tilting mechanism is assumed to carry the carbody load allowing the actuator to be specialized to control the tilt motion. Tilting is used to reduce the quasi-static lateral acceleration perceived by the passenger and thereby allowing higher speeds in curves. Activation of no-load carrying tilting is a mature application today.

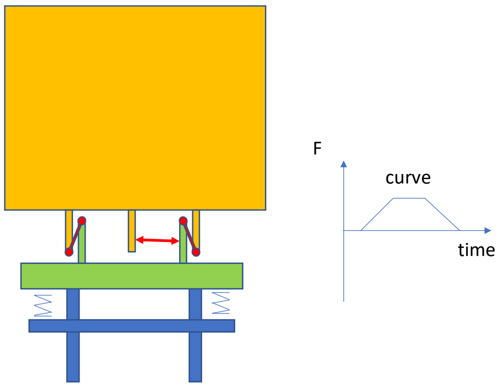

Figure 302: Activation of non-load carrying carbody tilt

6.11.2 Requirements

The actuator must be able to create a quasi-static force large enough to tilt the carbody while curving. The speed must be large enough to bring the carbody to a new tilting position in a certain time. Most transitions from one track element to another involves a curve transition that connects a straight track to curve. Such elements are normally a few seconds long allowing vehicle and passengers to adopt the new condition. The stroke must be large enough to allow the full tilt travel, which is many mm. The actuator should be stiff as it otherwise loses its motion controllability (Table 13)

Table 13: Activation of non-load carrying carbody tilt

Property	Requirement
Type	Active
Force	> 25 kN
Speed	50 – 100 mm/s
Stroke	± 200 mm
Stiffness	Stiff

6.12 LOAD CARRYING TILT

6.12.1 Intention of the application

The intention of the application is to let the suspension also provide carbody tilting. This implies that the actuator both carries the load and activates tilting. Tilting is used to reduce the quasi-static lateral acceleration perceived by the passenger and thereby allowing higher speeds in curves. A system setup this way will be power consuming as the carbody must be raised and lowered in each curve. Therefore, the application is limited to small tilt angles. Load carrying tilting is a mature application today.

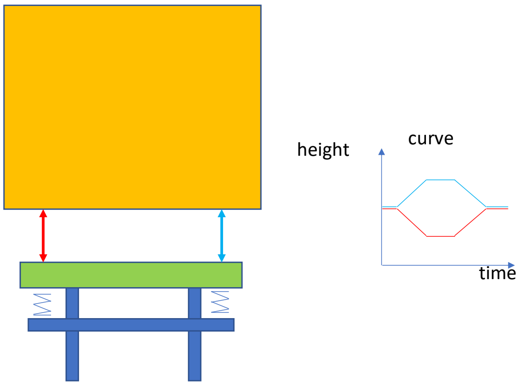

Figure 313: Load carrying tilt

6.12.2 Requirements

The actuator must be able to create a quasi-static force large enough to lift the carbody. The speed must be large enough to bring the carbody to a new tilting position in a certain time. Most transitions from one track element to another include a curve transition that connects a straight track to curve. Such elements are normally a few seconds long allowing vehicle and passengers to the adapt the new condition. The stroke must be large enough to allow the full tilt travel, which is many mm. The actuator should be soft as it otherwise will transfer vibrations from bogie frame to carbody (Table 14)

Table 14: Load carrying tilt

Property	Requirement
Type	Active
Force	100 kN
Speed	10 – 20 mm/s
Stroke	± 40 mm
Stiffness	Soft

7. ACTUATOR TECHNOLOGY VALIDATION

A lot of actuators are used in semi-active and active suspension system in different applications in the market. The market research brought out 39 different actuator types which are used in the industry or were analyzed by universities. The information was documented in an actuator technology information form for each actuator which collects the main technical data, a short functional description and the pros and cons. This documentation is not part of this deliverable, but is stored at the CT4 Tool of the project.

Due to the difference of the actuator technologies, which you have seen in the sections before, it was necessary to validate them. The following validation criteria were defined:

- Level of maturity
- Technical use for railway application
- Cost estimation

The validation was done form the team members of the work package 3.1, which are specialists on suspension technology. They have collected their experience by working for railway OEM´s and universities, which are specialized in this area.

The values show the average of the validation from the team members.

The validation matrices are shown in Tables 15-17.

7.1 LEVEL OF MATURITY MATRIX

Table 15: Level of maturity matrix

Actuator technology								
Type	Full active					Semi-active		
	1	2	3	4	5	6	7	8
Description	Central hydraulic power pack	Compact hydraulic actuator	Electro-mechanical	Pneumatic	Electro-magnetic	Electro-hydraulic damper	MR rheological	ER rheological
slow acting	5	5	5	5	2			
fast acting	5	5	4	2	4	5	5	1

Legend	
1	in development
2	
3	medium
4	
5	series product

7.2 TECHNICAL VALIDATION MATRIX

Table 16: Technical validation matrix

Actuator technology											
Type				Full active					Semi-active		
				1	2	3	4	5	6	7	8
Description				hydraulic central power pack	hydraulic compact actuator	electro-mechanical	pneumatic use of existing supply	electro-magnetic	hydraulic	magneto rheological	electro rheological
Level of maturity of technology			slow acting	5	5	5	5	2			
			fast acting	5	5	4	2	4	5	5	1
Active suspension application	vehicle with 2 suspension stages	Secondary	lateral (centering)	3,33	4,17	3,50	4,00	2,17	2,00	1,67	1,17
			lateral (dynamic)	3,33	4,33	3,00	2,17	4,00	4,50	3,00	1,83
			vertical (levelling)	3,17	4,00	3,33	4,83	2,17	2,00	1,67	1,17
			vertical (dynamic)	3,17	4,00	2,83	2,00	3,50	4,50	3,50	2,17
			yaw (stability)	3,33	3,83	3,17	1,33	2,33	3,17	2,33	1,67
			yaw (steering)	3,00	3,67	4,00	2,00	2,33	2,00	2,00	1,50
		Primary	lateral (dynamic)	1,33	1,50	1,17	1,00	1,17	1,83	1,17	1,00
			vertical (dynamic)	1,83	2,17	2,00	1,83	2,00	3,17	2,33	1,33
			yaw (stability)	2,50	3,00	3,50	1,50	1,83	2,00	1,00	1,00
			yaw (steering)	3,00	3,83	4,83	2,33	1,67	1,00	0,83	0,83
	Innovative vehicle with one suspension stage		lateral (centring)	2,50	3,17	2,50	3,33	2,00	1,83	1,50	1,00
			lateral (dynamic)	2,83	3,33	2,50	1,83	3,67	3,83	2,83	1,83
			vertical (levelling)	2,67	3,50	2,50	3,83	2,17	2,00	1,67	1,17
			vertical (dynamic)	2,83	3,50	2,33	2,33	3,33	4,17	3,50	2,33
			yaw stability	3,00	3,33	3,33	1,50	1,67	2,50	1,67	1,33
			yaw (steering)	3,17	3,67	4,67	2,00	1,83	1,00	0,83	0,83

Legend	
1	not possible to use at railway
2	
3	medium
4	
5	easy to use at railway

7.3 COST ESTIMATION MATRIX

Table 17: Cost estimation matrix

				Actuator technology							
Type				Full active					Semi-active		
				1	2	3	4	5	6	7	8
Description				hydraulic central power pack	hydraulic compact actuator	electro-mechanical	pneumatic use of existing supply	electro-magnetic	hydraulic	magneto rheological	electro rheological
Level of maturity of technology			slow acting	5	5	5	5	2			
			fast acting	5	5	4	2	4	5	5	1
Active suspension application	vehicle with 2 suspension stages	Secondary	lateral (centering)	2,67	3,00	4,00	4,67	1,33	0,00	0,00	0,00
			lateral (dynamic)	2,67	3,00	4,00	4,00	3,00	5,00	3,67	2,00
			vertical (levelling)	2,67	3,00	4,00	4,67	1,33	0,00	0,00	0,00
			vertical (dynamic)	2,67	3,00	4,00	4,00	3,00	5,00	3,67	2,33
			yaw (stability)	2,67	3,00	3,67	2,67	1,67	4,67	3,33	2,00
			yaw (steering)	2,67	2,67	4,00	4,67	1,67	0,00	0,00	0,00
		Primary	lateral (dynamic)	2,00	2,00	1,67	2,33	1,33	2,33	2,00	1,67
			vertical (dynamic)	1,67	2,00	3,00	3,67	3,00	3,67	3,00	1,67
			yaw (stability)	2,00	3,00	3,67	2,33	1,67	2,67	1,67	1,33
			yaw (steering)	2,67	3,00	4,00	4,33	1,67	1,00	0,33	0,33
	Innovative vehicle with one suspension stage		lateral (centring)	2,00	2,33	2,33	3,00	1,33	0,00	0,00	0,00
			lateral (dynamic)	1,67	2,33	2,33	4,00	3,00	3,33	3,67	2,00
			vertical (levelling)	2,00	2,33	2,33	3,00	1,67	0,00	0,00	0,00
			vertical (dynamic)	1,67	2,33	2,33	4,00	3,00	4,67	3,33	2,00
			yaw stability	2,67	3,00	3,67	2,00	1,67	3,67	3,00	1,67
			yaw (steering)	3,00	3,33	4,00	2,33	1,67	0,33	0,33	0,33

Legend	
1	high cost impact
2	
3	medium
4	
5	low cost impact

The requirement specification of the actuators in railway application (see section 6) and the validation matrices provide the basis for the next work package T3.2, which has the task to look at the implementation of active technology on conventional bogie vehicles and a two-axle vehicle.

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